HEATING VENTILATING AIRCONDITIONING GUIDE

1942

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AMERICAN SOCIETY of HEAT
and VENTILATING ENGINEE

HEATING VENTILATING AIR CONDITIONING GUIDE 1942

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1942

AN INSTRUMENT OF SERVICE PREPARED FOR THE PROFESSION—CONTAINING A

Technical Data Section

OF REFERENCE MATERIAL ON THE DESIGN AND SPECIFICATION OF HEATING, VENTILATING AND AIR CONDITIONING SYSTEMS BASED ON—THE TRANSACTIONS—THE INVESTIGATIONS OF THE RESEARCH LABORATORY AND CO-OPERATING INSTITUTIONS—AND THE PRACTICE OF THE MEMBERS AND FRIENDS OF THE SOCIETY

TOGETHER WITH A

Manufacturers' Catalog Data Section

CONTAINING ESSENTIAL AND RELIABLE INFORMATION CONCERNING MODERN EQUIPMENT

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PREFACE TO THE 20th EDITION

IN the preparation of this 20th edition, the HEATING, VENTILATING, AIR CONDITIONING GUIDE 1942, the objective of the Guide Publication Committee has been to present the latest available authoritative information on the various subjects covered in the 47 Chapters. In accordance with the policy established by its founders, approximately one-half of the GUIDE chapters have been reviewed. Of these nine were completely rewritten and minor revisions were made to nine others.

A new chapter entitled Fundamentals of Heat Transfer has been added which includes the basic equations for conduction, convection and radiation. A detailed solution is given for a problem involving all three mechanisms of heat transfer. This new Chapter 3 in Section I Principles, should prove very helpful to many GUIDE readers. Except for the insertion of this new chapter, there has been no change in the order of subject presentation.

Some revisions have been made in the text of the chapter on Thermodynamics of Air and Water Mixtures. The Mollier Diagram for Moist Air, which first appeared in the GUIDE 1941 has been redrawn, and a new Volume Diagram for Moist Air has been added. Both will be found in the pocket on the inside of the back cover.

Material in the chapter on Physiological Principles has been revised to include current knowledge on the thermal interchanges taking place between the body and the environment, which has been prepared from results observed at the A.S.H.V.E. Research Laboratory and in cooperative institutions.

The chapter on Central Systems for Comfort Air Conditioning has been completely rewritten and the revision of the chapters on Air Distribution, Air Duct Design, Sound Control and Fans provides new data on the design of air handling systems. The new air friction chart developed at the A.S.H.V.E. Research Laboratory is included in the Air Duct Design chapter. The material on Sound Control contains additional information on machinery isolation and detailed information has been included for calculating the attenuation in a duct system with various sound absorbing treatments.

The chapter on Radiant Heating has undergone major revisions, and now contains the essential data necessary for the design of radiant heating systems. A new chart has been added to the chapter on Pipe and Duct Heat Losses for determining the heat gain or loss from an insulated duct. A group of refrigerating graphical symbols for drawings have been added to the chapter on Terminology.

In addition to the chapters specifically mentioned, the following were also reviewed and revised: Combustion and Fuels, Automatic Fuel Burning Equipment, Radiators and Convectors, Pipe, Fittings, Welding, Heat Transfer Surface Coils, Spray Equipment, Air Pollution, Air Cleaning Devices, Natural Ventilation, and Water Supply Piping and Water Heating.

The greater part of the text of the GUIDE results from the contributions of many individuals submitted over a considerable period of years. The Guide Publication Committee acknowledges its indebtedness to all those who have participated in the compilation of earlier editions. The Committee also wishes to thank the many GUIDE users for their helpful suggestions, many of which were used in the preparation of this GUIDE 1942.

Special thanks are due to the following men who have assisted the Committee and have contributed the new material which appears in this edition:

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Still others who have assisted in the preparation of this GUIDE and who merit the thanks of the Committee are the members of the Committee on Research, authors of papers, and the many men who willingly gave their time in the review and editing of the material.

The reader's attention is also directed to the Catalog Data Section and careful inspection of this section is recommended. Concise information on many types of equipment is included in this section, and it has been carefully arranged in logical sub-divisions to facilitate its use.

This 1942 edition of the GUIDE is released with the sincere hope that its users will find that it is a fitting tribute to those whose 20 years of unselfish service have contributed so much to the advancement of heating, ventilating and air conditioning. May this GUIDE be as useful as its predecessors.

GUIDE PUBLICATION COMMITTEE

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CODE *of* ETHICS *for* ENGINEERS

ENGINEERING work has become an increasingly important factor in the progress of civilization and in the welfare of the community. The engineering profession is held responsible for the planning, construction and operation of such work and is entitled to the position and authority which will enable it to discharge this responsibility and to render effective service to humanity.

That the dignity of their chosen profession may be maintained, it is the duty of all engineers to conduct themselves according to the principles of the following Code of Ethics:

1—The engineer will carry on his professional work in a spirit of fairness to employees and contractors, fidelity to clients and employers, loyalty to his country and devotion to high ideals of courtesy and personal honor.

2—He will refrain from associating himself with or allowing the use of his name by an enterprise of questionable character.

3—He will advertise only in a dignified manner, being careful to avoid misleading statements.

4—He will regard as confidential any information obtained by him as to the business affairs and technical methods or processes of a client or employer.

5—He will inform a client or employer of any business connections, interests or affiliations which might influence his judgment or impair the disinterested quality of his services.

6—He will refrain from using any improper or questionable methods of soliciting professional work and will decline to pay or to accept commissions for securing such work.

7—He will accept compensation, financial or otherwise, for a particular service, from one source only, except with the full knowledge and consent of all interested parties.

8—He will not use unfair means to win professional advancement or to injure the chances of another engineer to secure and hold employment.

9—He will cooperate in upbuilding the engineering profession by exchanging general information and experience with his fellow engineers and students of engineering and also by contributing to work of engineering societies, schools of applied science and the technical press.

10—He will interest himself in the public welfare in behalf of which he will be ready to apply his special knowledge, skill and training for the use and benefit of mankind.

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Chapter I

THERMODYNAMICS OF AIR AND WATER MIXTURES

Dry Air, Specific Enthalpy, Water Vapor, Moist Air, Dalton's Law, Humidity Ratio, Relative Humidity, Dew-point, Enthalpy, Thermodynamic Wet-bulb Temperature, Mollier Diagram, Typical Air Conditioning Processes, Adiabatic Saturation, Psychrometric Chart, Steady Flow Energy Equation

THE working substance of the air conditioning engineer may be regarded, for the purpose of analysis, as a mixture of only two constitutents, dry air and water. The mixture may consist of two, and possibly three distinct phases, solid, liquid and vapor. The vapor phase is conveniently referred to as *moist air* and is regarded as a mixture of *dry air* and *water vapor.*

DRY AIR

Composition. Dry air is itself a mixture of several gases, but its composition is subject to such slight variation that it may be regarded as fixed. According to *International Critical Tables*, the mol-fraction composition of dry air is given by the first column of figures in Table 1. Molecular weights are given in the second column; the last figure in the third column is the *apparent* molecular weight of the mixture; the fourth column of figures gives the ordinary weight-fraction composition.

It is well known that dry air contains other gases besides those listed in Table 1; but these are present in such minute amounts that they can be grouped together as argon. Values in the lower section of Table 1 give the approximate mol-fraction composition of what is called argon in the upper portion of the table.

In physical and chemical thermodynamics, there is a distinct advantage in using a different unit of weight, the mol, for each different substance involved. A mol of oxygen weighs 32.000 lb as a matter of definition; a mol of any other substance is a weight, in pounds, equal to its molecular weight.

Specific Volume and Density

The ratio of total volume to total weight is called *specific volume, v.* In the English system, volume is expressed in cubic feet and weight in pounds; hence, specific volume is expressed in cubic feet per pound. The reciprocal of specific volume, that is, weight per unit volume is called *weight density, d.* The unit of density is the pound per cubic foot.

The earliest investigation into the relation between pressure, specific volume, and temperature for gases was made by Boyle (1661) who was able to confirm the hypothesis that the volume of a given weight of gas should vary inversely as the absolute pressure if temperature is maintained constant. Thus, within the limits of his experimental error Boyle found that at constant temperature the product pv, pressure times specific volume, has a constant value over a considerable range of pressures. These results are best visualized by plotting values of the product pv as ordinate against values of pressure p itself as abscissa. According to Boyle's experimental findings lines of constant temperature (isotherms) of a gas are *straight* and *horizontal* on this pv, p-plane.

The first rough experiments of Charles (1787) and the subsequent more refined experiments of Gay-Lussac (1802) suggested the possibility

TABLE 1. COMPOSITION OF DRY AIR

GAS	MOL PER MOL DRY AIR		LB PER MOL		LB PER MOL DRY AIR	LB PER LB DRY AIR
Nitrogen	0.7803	×	28.016	=	21.861	0.7547
Oxygen	0.2099	×	32.000	=	6.717	0.2319
Carbon Dioxide	0.0003	×	44.003	=	0.013	0.0004
Hydrogen	0.0001	×	2.016	=	0.000	0.0000
Argon	0.0094	×	39.944	=	0.376	0.0130
	1.0000				28.967	1.0000

COMPOSITION OF ARGON	MOL PER MOL DRY AIR
Argon	0.00933
Neon	0.000018
Helium	0.000005
Krypton	0.000001
Xenon	....
	0.00935

of establishing a universal temperature scale such that the product pv *for any gas* is simply proportional to temperature measured on this scale in accordance with Equation 1.

$$pv = BT \tag{1}$$

where B is a constant characteristic of the given gas. Referring to the graphical representation previously described in which the product pv is plotted as ordinate against pressure p as abscissa, the vertical spacing of the isotherms should be such that the ordinates to any two isotherms are in the ratio of corresponding absolute temperatures and therefore in the same ratio for any gas.

Precise measurements by modern methods have shown that the experimental findings of Boyle, Charles and Gay-Lussac are only approximately correct. In the range of sufficiently low pressures the isotherms of gases are indeed *straight* on the pv, p-plane; but they are *not horizontal* in accordance with Boyle's Law, being inclined downward to the right at

relatively low temperatures, upward to the right at higher temperatures. Extrapolation of each isotherm to zero pressure has revealed the remarkable fact that the limiting value of the product pv thus obtained is strictly proportional to absolute temperature as suggested by Equation 1, this strict proportionality providing an accurate basis for the establishment of the absolute temperature scale.

The experimental facts of the preceding paragraph are expressed mathematically by Equation 2.

$$pv = BT - A(T)\,p \tag{2}$$

where

p = absolute pressure, pounds per square foot.
v = specific volume, cubic feet per pound.
B = a constant depending on the molecular weight of the gas.
T = absolute temperature, degrees Fahrenheit.
$A(T)$ = a temperature function called *second virial coefficient*, cubic feet per pound [2].* The name undoubtedly originated from consideration of Clausius' Virial Theorem according to which the mean kinetic energy of a molecular aggregate is equal to the mean value of a quantity, which Clausius called the *virial* of the system, depending solely on the forces acting upon the molecules and not upon the motion of the molecules. This name is used extensively. For some gases, the magnitude of the second virial coefficient can be predicted from theory; but at present, direct experimental measurements are more reliable.

It will appear in what follows that the error committed in computing values of specific volume from Equation 1 instead of Equation 2 is extremely small. Thermodynamically, however, the former would deny the effect of pressure on the thermal properties of a gas which experiment shows to be appreciable. Therefore Equation 1 cannot be made the basis of an accurate analysis.

The numerical value of the constant B in Equation 2 is different for every different gas, but can be calculated if the molecular weight m, pounds per mol, is known; for the product mB is a universal gas constant R, namely,

$$R = 1545.4$$

Example 1. Find the value of B for dry air and water vapor.
Solution. $B_a = 1545.4 \div 28.967 = 53.351$
$B_w = 1545.4 \div 18.0154 = 85.782$

The temperature function $A(T)$, the so-called *second virial coefficient*, expresses the effect of intermolecular forces. It is positive at low temperatures where these forces are predominantly attractive, negative at higher temperatures where they are predominantly repulsive. It is known with satisfactory accuracy for both dry air and water vapor. Values of specific volume are listed in Table 2 for dry air at standard atmospheric pressure (29.921 in. Hg) as computed from Equation 2.

The fact that $A(T)$ is multiplied by pressure in Equation 2 means that intermolecular forces vanish at zero pressure and infinite volume where infinite distances separate the molecules. The finite value of the product pv at zero pressure is due entirely to the translational kinetic

*Bracketed numbers refer to references at end of chapter.

TABLE 2. SPECIFIC VOLUME OF DRY AIR[a] at 29.921 IN. HG

TEMP F t	CU FT PER LB v_a	TEMP F t	CU FT PER LB v_a	TEMP F t	CU FT PER LB v_a
−96	9.1488	32	12.3888	160	15.6229
−64	9.9597	64	13.1977	192	16.4310
−32	10.7699	96	14.0063	224	17.2389
0	11.5796	128	14.8147	256	19.0467

[a]Prepared by John A. Goff.

energy of the molecules. In ordinary calculations not requiring too great accuracy, the effect of intermolecular forces may be ignored and Equation 2 simplified to

$$pv = BT \tag{1}$$

Example 2. Calculate an approximate value for the specific volume of dry air at 64 F, 29.921 in. Hg.

Solution. $v = \dfrac{53.351 \times 523.70}{29.921 \times 0.49115 \times 144} = 13.200$ cu ft per pound.

Note: This answer may be compared with the value in Table 2. The difference is due to intermolecular forces. It should not be concluded, however, that because the effect of intermolecular forces on the volume is so small these forces can be ignored entirely.

The relationship of Equation 1 expresses certain familiar laws approximately true for gases at not too high pressures. Thus, *with temperature constant, the volume of a given weight of gas is inversely proportional to its absolute pressure* is a statement of Boyle's Law. If v_1 denotes the specific volume at absolute pressure p_1, then *at the same temperature*, the specific volume v_2 at absolute pressure p_2 is approximately,

$$v_2 = v_1 \left(\frac{p_1}{p_2}\right)$$

Also, *with pressure constant the volume of a given weight of gas is directly proportional to its absolute temperature* is a statement of Charles' Law. If v_1 denotes the specific volume at absolute temperature T_1, then *at the same pressure*, the specific volume v_2 at absolute temperature T_2 is approximately,

$$v_2 = v_1 \left(\frac{T_2}{T_1}\right)$$

Absolute Temperature

For the range 0 to 660 C, the standard temperature scale is the International Centigrade Scale, namely, the readings of a platinum resistance thermometer standardized at the ice-point (0 C), the steam-point (100 C) and the sulphur-point (444.60 C). The corresponding Fahrenheit scale t used in scientific work is derived from the International Centigrade Scale by means of the relation,

$$t = 1.8\,(\text{Int. Cent. Temp.}) + 32 \tag{3}$$

Temperatures on the absolute Fahrenheit scale are then obtained by adding 459.70 according to the equation

$$T = t + 459.70 \tag{4}$$

Absolute temperatures computed from Equations 3 and 4 are *practically* identical with the fundamental thermodynamic temperatures to which the zero-pressure values of the product pv for gases are proportional in accordance with Equation 2.

Specific Enthalpy

Most air conditioning processes are of the steady-flow type. In steady flow the energy convected with the fluid crossing a given section is the sum of (a) kinetic energy due to velocity, (b) gravitational energy due to elevation, (c) enthalpy due to the condition of temperature, pressure and composition at a given section. It is clear, therefore, that in order to apply the Law of Conservation of Energy to steady-flow processes, information regarding the enthalpy is needed.

Recent developments in quantum mechanics have made it possible to calculate the zero-pressure specific enthalpy of a gas from spectroscopic measurements, and with a degree of accuracy exceeding that with which this property can be inferred from direct calorimetric measurements. Available data for each gas listed in Table 1 have been assembled and critically examined; and from them have been calculated best values for the specific enthalpy of dry air at zero pressure. These are listed in Table 3. The unit of energy is the Btu which is related to the foot-pound as follows:

$$1 \text{ Btu} = 778.18 \text{ ft-lb}^{1} \tag{5}$$

In Table 3 are also listed values of *mean zero-pressure specific heat* for the range 0 to t F. This is simply the increase of specific enthalpy from 0 to t F, divided by the increase of temperature or, with 0 F as the reference point, by the temperature itself. The numerical values indicate that a rounded figure of 0.24 Btu per pound can be used in ordinary calculations.

Applying well known identical relations of thermodynamics to Equation 2, the following expression for specific enthalpy, valid at not too high pressures, is obtained

$$h = h^0 + \left[T^2 \frac{d\,(A/T)}{dT}\right] p \tag{6}$$

where h^0 denotes specific enthalpy at zero pressure. This equation emphasizes that the effect of pressure on specific enthalpy is not so much due to the second virial coefficient A itself as to its variation with tempera-

TABLE 3. SPECIFIC ENTHALPY OF DRY AIR AT ZERO PRESSURE[a]

TEMP F t	SPECIFIC ENTHALPY BTU PER LB h_a^o	MEAN SPECIFIC HEAT $\left[c_p^o\right]_o^t$	TEMP F t	SPECIFIC ENTHALPY BTU PER LB h_a^o	MEAN SPECIFIC HEAT $\left[c_p^o\right]_o^t$	TEMP F t	SPECIFIC ENTHALPY BTU PER LB h_a^o	MEAN SPECIFIC HEAT $\left[c_p^o\right]_o^t$
−96	−22.839	0.2393	32	7.796	0.2395	160	38.529	0.2400
−64	−15.186	0.2393	64	15.466	0.2396	192	46.238	0.2401
−32	−7.529	0.2394	96	23.145	0.2397	224	53.962	0.2403
0	+0.131	0.2394	128	30.831	0.2398	256	61.702	0.2405

[a]Prepared by John A Goff from published data computed from spectroscopic measurements.

[1]This conversion factor is not exact by definition, but involves an experimental determination of the relation between the absolute and the standard electrical units of energy. The value 1 int. joule = 1.00019 abs. joule, recommended by Osborne, Stimson and Ginnings [8] was used.

Table 4. Specific Enthalpy of Dry Air[a] at 29.921 In. Hg

Temp F t	Specific Enthalpy Btu per Lb h_a	Specific Heat $[c_p]_0^t$	Temp F t	Specific Enthalpy Btu per Lb h_a	Specific Heat $[c_p]_0^t$	Temp F t	Specific Enthalpy Btu per Lb h_a	Specific Heat $[c_p]_0^t$
−96	−23.035	0.2399	32	7.680	0.2400	160	38.454	0.2403
−64	−15.356	0.2399	64	15.363	0.2400	192	46.172	0.2405
−32	−7.678	0.2399	96	23.053	0.2401	224	53.903	0.2406
0	0.000	0.2400	128	30.749	0.2402	256	61.649	0.2408

[a]Prepared by John A. Goff.

ture. In other words, the pressure effect may be much more important than the corresponding effect on specific volume. Values of the specific enthalpy of dry air at standard atmospheric pressure (29.921 in. Hg) as computed from Equation 6 are given in Table 4.

Reference Point. It is desired to give some prominence to the choice of reference point. As energy, and therefore enthalpy, is purely relative, any convenient state can be selected at which to assign the value *zero* to specific enthalpy. The state chosen is 0 F, 29.921 in. Hg. Perhaps the only really valid argument for this particular choice is that, for ordinary calculations at, or near, atmospheric pressure, a very simple equation can be used, namely,

$$h_a = 0.24t \qquad (7)$$

WATER VAPOR

Saturation Pressure. It is common knowledge that a substance like water can exist in at least three distinct phases, solid (ordinary ice), liquid and vapor; and that under certain conditions two or more phases can *co-exist* in stable equilibrium. For example, steam having a quality of 98 per cent is a mixture of two co-existing phases, vapor and liquid, 98 per cent by weight being vapor and 2 per cent by weight, liquid. When two phases can co-exist in stable equilibrium, each is said to be *saturated* with respect to the other.

One of the important problems of thermodynamics is to formulate the conditions for *saturation* in mathematical terms. The answer to the problem can be stated quite generally as equality, between the several co-existing phases, of (a) pressure, (b) temperature, (c) each component *chemical potential.*

In the case of a pure substance like water, containing a single component, there is only one component chemical potential; and this becomes identical with a thermodynamic property called *specific free enthalpy* denoted by the letter g (Btu per pound) and defined by the equation:

$$g = h - Ts$$

where

h = specific enthalpy, Btu per pound.
T = absolute temperature, degrees Fahrenheit.
s = specific entropy, Btu per pound per degree Fahrenheit.

To illustrate, *liquid* water at 212 F, 14.696 lb per square inch has a specific free enthalpy of $180.07 - 671.70 \times 0.3120 = -25.90$ Btu per pound. At the same temperature and pressure, water *vapor* has a specific

free enthalpy of 1150.4 − 671.70 × 1.7566 = −25.90 Btu per pound. The numerical data used in these calculations are to be found in the steam tables[2]. Since the two specific free enthalpies are equal *at the same temperature and the same pressure*, the two phases can co-exist in stable equilibrium to form a saturated mixture and are therefore saturated with respect to each other.

But suppose that a different pressure had been assumed, the temperature being 212 F as before; for example, assume a pressure of 14 lb per square inch. The specific free enthalpy of the liquid phase will be practically the same as before, but that of the vapor phase will change from −25.90 to −32.84 Btu per pound, most of this change being due to change of entropy which, in the case of a vapor, depends markedly upon the pressure. Since the specific free enthalpies of the two phases are no longer equal, they cannot co-exist in stable equilibrium and neither is saturated. As a matter of fact the vapor is superheated while the liquid is supersaturated.

From this analysis it will be seen that *to a given temperature T there corresponds a definite saturation pressure p_s.* This is also called the *vapor pressure* of the liquid or solid as the case may be. It will also be seen that a working definition of saturation can only be arrived at by application of the fundamental laws of thermodynamics.

Referring specifically to the vapor phase, if the actual pressure is *less* than the saturation pressure corresponding to the actual temperature, the vapor is said to be *superheated*; if it is *greater*, as it may well be under proper circumstances, the vapor is said to be *supersaturated.* Values of the saturation pressure of pure water are given in Table 6[3].

Specific Volume

Accurate values of the specific volume of water vapor at pressures equal or near the saturation pressure (for the given temperature) can be computed from Equation 1 since the second virial coefficient $A(T)$ is known with satisfactory accuracy. Usually, however, the desired information can be read directly from the steam tables. Values for the specific volume of the *saturated* vapor, v_g, are also listed in Table 8.

Specific Enthalpy

The zero-pressure specific enthalpy, as calculated by A. R. Gordon from spectroscopic measurements, has recently been corrected for distortion of the water molecules due to centrifugal forces. Best values at present available are listed in Table 5.

From the numerical values of mean specific heat, it is clear that for ordinary calculations the following simple relation may be used:

$$h_w^\circ = 0.444t + 1061 \tag{8}$$

Reference Point. The reference point for water has been chosen as *saturated liquid at 32 F* in conformity with usual steam table practice. In order to refer the zero-pressure values of specific enthalpy to this

[2]Thermodynamic Properties of Steam, by J. H. Keenan and F. G. Keyes, published by John Wiley & Sons, Inc., 1936, of which Table 8 is an abridgment.

[3]Strictly speaking the values listed in Table 6 are not values of p_s as labeled, but of p_s' (Equation 13b) with the Dalton Factor (DF) taken to be unity.

TABLE 5. SPECIFIC ENTHALPY OF WATER VAPOR AT ZERO PRESSURE[a]

TEMP F t	SPECIFIC ENTHALPY BTU PER LB h_w^o	MEAN SPECIFIC HEAT $\left[c_p^o\right]_o^t$	TEMP F t	SPECIFIC ENTHALPY BTU PER LB h_w^o	MEAN SPECIFIC HEAT $\left[c_p^o\right]_o^t$	TEMP F t	SPECIFIC ENTHALPY BTU PER LB h_w^o	MEAN SPECIFIC HEAT $\left[c_p^o\right]_o^t$
−96	1018.61	0.4425	32	1075.28	0.4435	160	1132.38	0.4455
−64	1032.76	0.4427	64	1089.51	0.4440	192	1146.76	0.4462
−32	1046.92	0.4429	96	1103.76	0.4444	224	1161.20	0.4469
0	1061.09	0.4431	128	1118.05	0.4450	256	1175.70	0.4477

[a]Prepared by John A. Goff from published data computed from spectroscopic measurements.

datum, best available information regarding latent heat, saturation pressure and second virial coefficient at 32 F has been used. The values in Table 5 do not agree exactly with those in the steam tables, but do agree with later information from the National Bureau of Standards [8].

MOIST AIR

Dalton's Law. Having accurate information regarding the thermodynamic properties of dry air and water vapor *separately*, it is desired to predict the properties of moist air which is regarded as a mixture of these two constitutents. Statistical mechanics furnishes a starting point in the form of a prediction that, at not too high pressures,

$$Pv = RT - \left[A_{aa}x^2 + 2A_{aw}\,x\,(1 - x) + A_{ww}\,(1 - x)^2\right] P \qquad (9)$$

where

P = observed pressure, pounds per square foot.
v = specific volume, cubic feet per *mol.*
A_{aa} = second virial coefficient for the dry air expressing the effect of forces between air—air molecules, cubic feet per *mol.*
A_{ww} = second virial coefficient for the water vapor, expressing the effect of forces between water—water molecules, cubic feet per *mol.*
A_{aw} = *interaction constant* expressing the effect of forces between air—water molecules, cubic feet per *mol.*
x = mol-fraction of dry air in the mixture, mols dry air per mol mixture.

Equation 9 will be recognized as a generalization of Equation 2. Both A_{aa} and A_{ww} are known; but until recently no reliable information on the interaction constant A_{aw} has been available. Preliminary results of a cooperative investigation between the A.S.H.V.E. and the Towne Scientific School, University of Pennsylvania, have indicated that the ratio $2A_{aw}/(A_{aa} + A_{ww})$ has an approximately constant value λ = 0.075 [10]. However, before attempting to make use of this information it is advisable, in the interest of simplicity, to first ignore the complications arising from intermolecular forces.

Now, in the absence of intermolecular forces, each constituent gas in a mixture such as moist air would behave exactly as if it alone occupied the volume V at the temperature T of the mixture and: (1) the observed pressure P would be the sum of individual *partial* pressures p; (2) the total enthalpy H would be the sum of the individual enthalpies. This is the essence of Dalton's Law of Partial Pressures.

Referring to dry air by the subscript *a* and, to water vapor by the subscript *w*, Dalton's Law would predict

$$V = \frac{n_a RT}{p_a} = \frac{n_w RT}{p_w} = \frac{(n_a + n_w)\, RT}{P} \tag{10a}$$

where

$$P = p_a + p_w \tag{10b}$$

From these equations are easily obtained,

$$\frac{n_w}{n_a} = \frac{p_w}{P - p_w} \quad \text{or} \quad \frac{p_w}{P} = \frac{n_w/n_a}{1 + n_w/n_a} \tag{10c}$$

in which,

p_a = partial pressure of the *dry air*.
p_w = partial pressure of the *water vapor*.
P = observed pressure of the mixture.
n_a = weight of *dry air* (mols).
n_w = weight of *water vapor* (mols).

Humidity Ratio

In Equation 10c the ratio by weight of water vapor to dry air, n_w/n_a, is expressed in mols per mol. Most engineers prefer to express it in pounds per pound which can easily be done, since the molecular weights of both water vapor (18.0154 lb per mol) and of dry air (28.967 lb per mol) are known. Thus Equation 10c becomes

$$W = 0.62193\, \frac{p_w}{P - p_w} \quad \text{or} \quad \frac{p_w}{P} = \frac{W}{0.62193 + W} \tag{11}$$

There is little doubt but that the weight ratio W is the most convenient parameter in terms of which to express the composition of moist air; but to choose a suitable name and one that would have general acceptance has always been a perplexing problem. In previous issues of the GUIDE, *specific humidity* was adopted even though it was recognized that the adjective *specific* should properly refer to weight of water vapor per pound of *mixture*, and not per pound of dry air. Various other names have been proposed from time to time including: mixing ratio, proportionate humidity, density ratio, absolute humidity. It is believed that the name *humidity ratio* is most suggestive of the meaning which it is desired to express, that it violates no well established usage as does the name specific humidity and that its adoption will avoid much confusion.

To repeat: in the case of moist air, the ratio by weight (pounds) of water vapor to dry air is called *humidity ratio* and denoted by the letter W.

Saturation

It is often stated that moist air is *saturated* when the water vapor in it is itself in the *dry saturated condition* at the given temperature. This statement would imply that the humidity ratio of saturated moist air is, in accordance with Equation 11,

$$W_s = 0.62193\, \frac{p_s}{P - p_s} \tag{12}$$

where p_s is the saturation pressure of pure water vapor.

Table 6. Thermodynamic Properties of Moist Air[a], 29.921 In. Hg

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air		Volume Cu Ft per Lb Dry Air			Enthalpy Btu per Lb Dry Air			Specific Enthalpy of Solid Water Btu per Lb	Saturation Pressure $p_s \times 10^5$		Temp Deg F
	Pounds $\times 10^5$	Grains	Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	h'_w	In of Hg	Lb per Sq In.	
−60	2.108	0.14756	10.07	0.00	10.07	−14.48	0.02	−14.46	−185.4	101.4	49.808	−60
−59	2.262	.15834	10.09	.00	10.09	−14.23	.02	−14.21	−185.0	108.8	53.443	−59
−58	2.418	.16926	10.12	.00	10.12	−13.99	.02	−13.97	−184.6	116.3	57.127	−58
−57	2.595	.18165	10.14	.00	10.14	−13.75	.03	−13.72	−184.2	124.8	61.302	−57
−56	2.773	.19411	10.17	.00	10.17	−13.50	.03	−13.47	−183.8	133.4	65.526	−56
−55	2.973	0.20811	10.19	0.00	10.19	−13.26	0.03	−13.23	−183.4	143.0	70.242	−55
−54	3.181	.22267	10.22	.00	10.22	−13.02	.03	−12.99	−182.9	153.0	75.154	−54
−53	3.399	.23793	10.24	.00	10.24	−12.78	.04	−12.74	−182.5	163.5	80.311	−53
−52	3.636	.25452	10.27	.00	10.27	−12.53	.04	−12.49	−182.1	174.9	85.911	−52
−51	3.888	.27216	10.29	.00	10.29	−12.29	.04	−12.25	−181.7	187.0	91.854	−51
−50	4.156	0.29092	10.32	0.00	10.32	−12.05	0.04	−12.01	−181.3	199.9	98.191	−50
−49	4.428	.30996	10.34	.00	10.34	−11.81	.05	−11.76	−180.9	213.0	104.63	−49
−48	4.738	.33166	10.37	.00	10.37	−11.57	.05	−11.52	−180.4	227.9	111.94	−48
−47	5.054	.35378	10.40	.00	10.40	−11.32	.05	−11.27	−180.0	243.1	119.41	−47
−46	5.395	.37765	10.42	.00	10.42	−11.08	.06	−11.02	−179.6	259.5	127.47	−46
−45	5.753	0.40271	10.45	0.00	10.45	−10.84	0.06	−10.78	−179.2	276.7	135.92	−45
−44	6.133	.42931	10.47	.00	10.47	−10.60	.06	−10.54	−178.7	295.0	144.90	−44
−43	6.543	.45801	10.50	.00	10.50	−10.35	.07	−10.28	−178.3	314.7	154.58	−43
−42	6.971	.48797	10.52	.00	10.52	−10.11	.07	−10.04	−177.9	335.3	164.70	−42
−41	7.435	.52045	10.55	.00	10.55	−9.872	.077	−9.795	−177.4	357.6	175.65	−41
−40	7.907	0.55349	10.57	0.00	10.57	−9.629	0.082	−9.547	−177.0	380.3	186.80	−40
−39	8.431	.59017	10.60	.00	10.60	−9.388	.088	−9.300	−176.6	405.5	199.18	−39
−38	8.965	.62755	10.62	.00	10.62	−9.146	.093	−9.053	−176.1	431.2	211.81	−38
−37	9.548	.66836	10.65	.00	10.65	−8.905	.100	−8.805	−175.7	459.2	225.56	−37
−36	10.16	.71120	10.67	.00	10.67	−8.663	.106	−8.557	−174.8	488.4	239.90	−36
−35	10.80	0.75600	10.69	0.00	10.69	−8.422	0.113	−8.309	−174.4	519.5	255.18	−35
−34	11.49	.80430	10.72	.00	10.72	−8.180	.120	−8.060	−174.0	552.4	271.34	−34
−33	12.20	.85400	10.75	.00	10.75	−7.939	.127	−7.812	−173.5	586.5	288.09	−33
−32	12.97	.90790	10.77	.00	10.77	−7.698	.136	−7.562	−173.1	623.7	306.36	−32
−31	13.76	.96320	10.80	.00	10.80	−7.457	.144	−7.313	−172.6	661.8	325.08	−31
−30	14.58	1.0206	10.82	0.00	10.82	−7.216	0.152	−7.064	−172.2	701.0	344.33	−30
−29	15.43	1.0801	10.85	.00	10.85	−6.975	.161	−6.814	−171.7	742.2	364.57	−29
−28	16.45	1.1515	10.87	.00	10.87	−6.734	.172	−6.562	−171.3	791.2	388.64	−28
−27	17.49	1.2243	10.90	.00	10.90	−6.493	.183	−6.310	−170.9	841.0	413.10	−27
−26	18.55	1.2985	10.92	.00	10.92	−6.251	.194	−6.057	−170.4	892.1	438.20	−26

[a]Compiled by W. M. Sawdon and extended by John A. Goff.

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (CONTINUED)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air		Volume Cu Ft per Lb Dry Air			Enthalpy Btu per Lb Dry Air			Specific Enthalpy of Solid Water Btu per Lb	Saturation Pressure $p_s \times 10^5$		Temp Deg F
	Pounds $\times 10^5$	Grains	Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	h'_w	In. of Hg	Lb per Sq In.	
−25	19.68	1.3776	10.95	0.00	10 95	−6.011	0.206	−5.805	−170.4	946.4	464.87	−25
−24	20.86	1.4602	10 97	00	10.97	−5.770	.219	−5.551	−170.0	1003.	492.67	−24
−23	22.13	1.5491	11.00	.00	11.00	−5.529	.232	−5.297	−169.5	1064.	522.64	−23
−22	23.42	1.6394	11.02	.00	11.02	−5.288	.246	−5.042	−169.0	1126.	553.09	−22
−21	24.79	1.7353	11.05	.00	11.05	−5.047	.260	−4.787	−168.6	1192.	585.51	−21
−20	26.25	1.8375	11.07	0.00	11.07	−4.807	0.276	−4.531	−168 2	1262.0	619.89	−20
−19	27.81	1.9467	11.10	.00	11.10	−4.566	.292	−4.274	−167.7	1337.	656.73	−19
−18	29.45	2.0615	11.13	.00	11.13	−4.325	.310	−4.015	−167.2	1416.	695.54	−18
−17	31.12	2.1784	11.15	.00	11.15	−4.085	.327	−3.758	−166.8	1496.	734.84	−17
−16	32.95	2.3065	11.18	.00	11.18	−3 844	.347	−3.497	−166.3	1584.	778.06	−16
−15	34.84	2.4388	11.20	0.01	11.21	−3.604	0.367	−3.237	−165 9	1675.0	822.76	−15
−14	36.86	2.5802	11.23	.01	11.24	−3.363	.388	−2.975	−165.4	1772.	870.41	−14
−13	38.98	2.7286	11.25	.01	11.26	−3.123	.411	−2.712	−165.0	1874.	920.51	−13
−12	41.19	2.8833	11.28	.01	11.29	−2.883	.434	−2.449	−164 5	1980.	972.58	−12
−11	43.54	3.0478	11.30	.01	11.31	−2.642	.459	−2.183	−164.0	2093.	1028.1	−11
−10	45.98	3.2186	11.33	0.01	11.34	−2.402	0.485	−1.917	−163.6	2210.0	1085.6	−10
−9	48.58	3.4006	11.35	.01	11.36	−2.162	.513	−1.649	−163.1	2335.	1147.0	−9
−8	51.25	3.5875	11.38	.01	11.39	−1.921	.541	−1.380	−162.7	2463.	1209.8	−8
−7	52.06	3.6442	11.40	.01	11.41	−1.681	.570	−1.111	−162.2	2502.	1229.0	−7
−6	57.12	3.9984	11.43	.01	11.44	−1.441	.603	−0.838	−161.7	2745.	1348.3	−6
−5	60.30	4.2210	11.45	0.01	11.46	−1.201	0.637	−0.564	−161.3	2898.0	1423.5	−5
−4	63.57	4.4499	11.48	.01	11.49	−0.960	.672	−0.288	−160.8	3055.	1500.6	−4
−3	67.05	4.6935	11.50	.01	11.51	−0.720	.709	−0.011	−160.3	3222.	1582.6	−3
−2	70.69	4.9483	11.53	.01	11.54	−0.480	.748	+0.268	−159.9	3397.	1668.6	−2
−1	74.50	5.2150	11.55	.01	11.57	−0.240	.789	+0.549	−159.4	3580.	1758.5	−1
0	78.52	5.5000	11.58	0.01	11.59	0	0.832	+0.832	−158.9	3773.0	1853.3	0

[a]Compiled by W. M. Sawdon and extended by John A. Goff.

Table 6. Thermodynamic Properties of Moist Air[a], 29.921 In. Hg (Continued)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Solid Water Btu per Lb h'_w	Saturation Pressure p_s: In. of Hg	Lb per Sq In.	Temp Deg F
0	0.0007852	5.50	11.58	0.01	11.59	0.000	0.832	0.832	−158.9	0.03773	0.01853	0
1	.0008275	5.79	11.60	.02	11.62	.240	.877	1.117	−158 5	.03975	.01963	1
2	.0008714	6.10	11.63	.01	11.64	.480	.924	1.404	−158 0	.04186	.02056	2
3	.0009179	6.43	11.65	.02	11.67	.720	.974	1.694	−157.5	.04409	.02166	3
4	.0009671	6.77	11.68	.02	11.70	.960	1.026	1.986	−157.0	.04645	.02282	4
5	0.001017	7.12	11.70	0.02	11.72	1.200	1.080	2.280	−156.6	0.04886	0.02400	5
6	.001071	7.50	11.73	.02	11.75	1.440	1.137	2.577	−156.1	.05144	.02527	6
7	.001127	7.89	11 75	.02	11.77	1.680	1.197	2 877	−155.6	.05412	.02658	7
8	.001186	8.30	11.78	.02	11.80	1.920	1 260	3.180	−155.1	.05692	.02796	8
9	.001247	8.73	11.80	.03	11.83	2.160	1.336	3 486	−154.7	05988	.02941	9
10	0.001311	9.18	11.83	0.02	11.85	2.400	1.395	3.795	−154 2	0 06295	0 03092	10
11	.001379	9.65	11.86	.02	11.83	2.640	1.468	4.108	−153 7	.06618	.03251	11
12	.001450	10.15	11.88	.03	11.91	2.880	1.544	4 424	−153.2	.06958	.03418	12
13	.001523	10.66	11.91	.02	11.93	3.120	1.622	4 742	−152.7	.07309	03590	13
14	.001600	11.20	11.93	.03	11.96	3.359	1.705	5.064	−152.2	07677	.03771	14
15	0.001682	11.77	11.96	0 03	11 99	3.599	1 793	5 392	−151 8	0.08067	0.03963	15
16	.001766	12.36	11.98	.03	12.01	3.839	1 883	5 722	−151.3	.08469	.04160	16
17	.001855	12 99	12.00	.04	12.04	4 079	1 979	6 058	−150.8	.08895	.04369	17
18	.001947	13.63	12.03	04	12.07	4.319	2 078	6 397	−150 3	.09337	.04586	18
19	.002043	14.30	12.06	.03	12 09	4 559	2 182	6 741	−149.8	.09797	.04812	19
20	0.002144	15.01	12 08	0.04	12.12	4 798	2 290	7 088	−149 3	0.1028	0.05050	20
21	.002250	15.75	12.11	.04	12.15	5.038	2 405	7 443	−148 8	.1078	05295	21
22	.002361	16.53	12.13	.05	12 18	5 278	2 524	7 802	−148 3	.1132	.05560	22
23	.002476	17.33	12.16	.04	12 20	5.518	2 648	8.166	−147 8	.1186	.05826	23
24	.002596	18.17	12.18	.05	12 23	5.758	2 778	8.536	−147.3	.1244	.06111	24
25	0.002722	19.05	12.21	0 05	12 26	5 998	2.914	8.912	−146 8	0.1304	0.06405	25
26	.002853	19.97	12 23	.06	12.29	6.237	3.055	9.292	−146 4	.1366	.06710	26
27	.002991	20.94	12 26	06	12 32	6.477	3.205	9.682	−145 9	1432	.07034	27
28	.003133	21.93	12 28	.06	12.34	6.717	3 358	10.075	−145 4	.1500	.07368	28
29	.003283	22.99	12 31	.06	12.37	6.957	3 520	10.477	−144 9	.1571	.07717	29
30	0.003439	24.07	12 33	0.07	12.40	7.197	3 689	10.886	−144 4	0 1645	0.08080	30
31	.003601	25.21	12.36	.07	12.43	7 437	3 865	11.302	−143 9	.1722	.08458	31
32	.003771	26.40	12.38	.08	12.46	7.677	4 049	11.726	−143 4	.1803	08856	32
33	.003931	27.52	12.41	.08	12.49	7.917	4 222	12.139	+ 1 0	.1879	.09230	33
34	.004094	28.66	12.43	.08	12 51	8 157	4.399	12 556	+ 2 0	1957	09610	34

[a]Compiled by W. M. Sawdon and extended by John A. Goff.

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (CONTINUED)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Liquid Water Btu per Lb h'_w	Saturation Pressure p_s: In. of Hg	Lb per Sq In.	Temp Deg F
35	0.004262	29.83	12.46	0.08	12.54	8.397	4.582	12.979	3.0	0.20360	0.1000	35
36	.004438	31.07	12.48	.09	12.57	8.636	4.773	13.409	4.0	.21195	.1041	36
37	.004618	32.33	12.51	.09	12.60	8.876	4.969	13.845	5.0	.22050	.1083	37
38	.004803	33.62	12.53	.10	12.63	9.116	5.169	14.285	6.0	.22925	.1126	38
39	.004996	34.97	12.56	.10	12.66	9.356	5.380	14.736	7.0	.23842	.1171	39
40	0.005194	36.36	12.59	0.10	12.69	9.596	5.595	15.191	8.0	0.24778	0.1217	40
41	.005401	37.80	12.61	.11	12.72	9.836	5.821	15.657	9.1	.25755	.1265	41
42	.005616	39.31	12.64	.11	12.75	10.08	6.05	16.13	10.1	.26773	.1315	42
43	.005840	40.88	12.66	.12	12.78	10.32	6.30	16.62	11.1	.27832	.1367	43
44	.006069	42.48	12.69	.12	12.81	10.56	6.55	17.11	12.1	.28911	.1420	44
45	0.006306	44.14	12.71	0.13	12.84	10.80	6.81	17.61	13.1	0.30031	0.1475	45
46	.006553	45.87	12.74	.13	12.87	11.04	7.08	18.12	14.1	.31191	.1532	46
47	.006808	47.66	12.76	.14	12.90	11.28	7.36	18.64	15.1	.32393	.1591	47
48	.007072	49.50	12.79	.14	12.93	11.52	7.64	19.16	16.1	.33635	.1652	48
49	.007345	51.42	12.81	.15	12.96	11.76	7.94	19.70	17.1	.34917	.1715	49
50	0.007626	53.38	12.84	0.15	12.99	12.00	8.25	20.25	18.1	0.36241	0.1780	50
51	.007921	55.45	12.86	.16	13.02	12.23	8.57	20.80	19.1	.37625	.1848	51
52	.008226	57.58	12.89	.17	13.06	12.47	8.91	21.38	20.1	.39051	.1918	52
53	.008534	59.74	12.91	.18	13.09	12.71	9.24	21.95	21.1	.40496	.1989	53
54	.008856	61.99	12.94	.18	13.12	12.95	9.60	22.55	22.1	.42003	.2063	54
55	0.009192	64.34	12.96	0.19	13.15	13.19	9.96	23.15	23.1	0.43570	0.2140	55
56	.009536	66.75	12.99	.20	13.19	13.43	10.34	23.77	24.1	.45179	.2219	56
57	.009890	69.23	13.01	.21	13.22	13.67	10.73	24.40	25.1	.46828	.2300	57
58	.01026	71.82	13.04	.21	13.25	13.91	11.14	25.05	26.1	.48538	.2384	58
59	.01064	74.48	13.06	.23	13.29	14.15	11.55	25.70	27.1	.50310	.2471	59
60	0.01103	77.21	13.09	0.23	13.32	14.39	11.98	26.37	28.1	0.52142	0.2561	60
61	.01144	80.08	13.11	.24	13.35	14.63	12.43	27.06	29.1	.54035	.2654	61
62	.01186	83.02	13.14	.25	13.39	14.87	12.89	27.76	30.1	.55970	.2749	62
63	.01229	86.03	13.16	.26	13.42	15.11	13.37	28.48	31.1	.57985	.2848	63
64	.01274	89.18	13.19	.27	13.46	15.35	13.86	29.21	32.1	.60042	.2949	64
65	0.01320	92.40	13.21	0.28	13.49	15.59	14.37	29.96	33.1	0.62179	0.3054	65
66	.01368	95.76	13.24	.29	13.53	15.83	14.90	30.73	34.1	.64378	.3162	66
67	.01417	99.19	13.26	.31	13.57	16.07	15.44	31.51	35.1	.66638	.3273	67
68	.01468	102.8	13.29	.31	13.60	16.31	16.00	32.31	36.0	.68980	.3388	68
69	.01520	106.4	13.31	.33	13.64	16.55	16.57	33.12	37.0	.71382	.3506	69

[a]Compiled by W. M. Sawdon and extended by John A. Goff.

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (CONTINUED)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air		Volume Cu Ft per Lb Dry Air			Enthalpy Btu per Lb Dry Air			Specific Enthalpy of Liquid Water Btu per Lb h'_w	Saturation Pressure p_s		Temp Deg F
	Pounds	Grains	Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s		In of Hg	Lb per Sq In.	
70	0.01574	110.2	13.34	0 34	13.68	16 79	17.17	33.96	38.0	0.73866	0 3628	**70**
71	.01631	114.2	13 37	.34	13 71	17.03	17.80	34 83	39 0	.76431	3754	71
72	01688	118 2	13 40	35	13 75	17.27	18 43	35.70	40.0	.79058	3883	72
73	.01748	122.4	13 42	37	13 79	17.51	19 09	36 60	41.0	81766	.4016	73
74	.01809	126.6	13.44	.39	13.83	17.75	19.76	37.51	42.0	.84555	.4153	74
75	0.01873	131.1	13.47	0.40	13 87	17 99	20 47	38.46	43.0	0 87448	0.4295	**75**
76	.01938	135.7	13.49	.42	13 91	18 23	21.19	39.42	44 0	.90398	.4440	76
77	.02005	140.4	13.52	43	13.95	18 47	21.93	40 40	45.0	.93452	.4590	77
78	.02075	145 3	13 54	45	13.99	18 71	22 71	41.42	46 0	96588	.4744	78
79	.02147	150 3	13 57	46	14.03	18 95	23 51	42 46	47.0	.99825	.4903	79
80	0 02221	155.5	13.59	0.49	14.08	19.19	24.32	43.51	48.0	1.0316	0.5067	**80**
81	.02298	160.9	13.62	50	14.12	19 43	25.18	44.61	49.0	1.0661	.5236	81
82	.02377	166.4	13.64	.52	14.16	19.67	26.05	45.72	50 0	1.1013	.5409	82
83	.02459	172.1	13.67	.54	14.21	19.91	26.97	46.88	51 0	1.1377	.5588	83
84	.02543	178.0	13 69	.57	14.26	20.15	27.90	48.05	52.0	1.1752	.5772	84
85	0.02629	184.0	13.72	0.58	14.30	20.39	28 85	49 24	53.0	1.2135	0.5960	**85**
86	.02718	190.3	13.74	.60	14.34	20.63	29 84	50.47	54 0	1.2527	.6153	86
87	.02810	196.7	13.77	62	14.39	20.87	30.87	51 74	55 0	1 2933	.6352	87
88	02904	203.3	13 79	65	14.44	21.11	31.91	53 02	56 0	1.3346	.6555	88
89	.03002	210.1	13 82	66	14.48	21.35	33.00	54.35	57.0	1.3774	.6765	89
90	0 03102	217.1	13.84	0.69	14.53	21.59	34.11	55.70	58.0	1.4211	0.6980	**90**
91	.03205	224.4	13 87	71	14.58	21.83	35 26	57.09	59.0	1.4661	7201	91
92	.03312	231.8	13.89	.74	14.63	22.07	36.45	58 52	60.0	1.5125	.7429	92
93	.03421	239.5	13.92	77	14.69	22.32	37 67	59 99	61 0	1.5600	.7662	93
94	.03535	247.5	13.94	.79	14.73	22.56	38.94	61.50	62 0	1.6088	.7902	94
95	0.03652	255.6	13 97	0 82	14.79	22.80	40.25	63.05	63 0	1.6591	0.8149	**95**
96	.03772	264.0	13.99	.85	14.84	23 04	41.58	64 62	64.0	1.7108	.8403	96
97	.03896	272.7	14 02	88	14.90	23.28	42 97	66.25	65.0	1.7638	.8663	97
98	.04024	281.7	14.04	91	14.95	23.52	44.40	67.92	66.0	1.8181	.8930	98
99	.04156	290.9	14.07	.94	15.01	23.76	45.87	69.63	67.0	1.8741	.9205	99
100	0 04293	300.5	14.10	0.97	15 07	24 00	47.40	71.40	68.0	1.9316	0.9487	**100**
101	.04433	310.3	14.12	1.00	15.12	24 24	48 97	73.21	69.0	1.9904	.9776	101
102	.04577	320.4	14.15	1.03	15.18	24.48	50.58	75 06	70.0	2 0507	1.0072	102
103	.04726	330.8	14.17	1.08	15.25	24.72	52 25	76.97	71 0	2.1128	1.0377	103
104	.04879	341.5	14.20	1.11	15.31	24 96	53.96	78 92	72.0	2.1763	1 0689	104

[a]Compiled by W. M. Sawdon and extended by John A. Goff.

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (CONTINUED)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air		Volume Cu Ft per Lb Dry Air			Enthalpy Btu per Lb Dry Air			Specific Enthalpy of Liquid Water Btu per Lb h'_w	Saturation Pressure p_s		Temp Deg F
	Pounds	Grains	Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s		In. of Hg	Lb per Sq In	
105	0.05037	352.6	14.22	1.15	15.37	25 20	55.73	80.93	73.0	2.2414	1.1009	**105**
106	.05200	364.0	14.25	1 19	15.44	25.44	57.56	83 00	74 0	2.3084	1 1338	106
107	.05368	375.8	14 27	1.23	15.50	25 68	59.45	85.13	75.0	2.3770	1 1675	107
108	.05541	387.9	14.30	1.27	15.57	25 92	61 38	87.30	76.0	2.4473	1.2020	108
109	.05719	400.3	14.32	1.32	15.64	26.16	63.38	89.54	76 9	2.5196	1.2375	109
110	0.05904	413 3	14.35	1.36	15.71	26.40	65.46	91.86	77.9	2.5939	1.274	**110**
111	.06092	426.4	14.37	1.41	15.78	26.64	67.57	94.21	78.9	2.6692	1 311	111
112	.06292	440.4	14.39	1.46	15.85	26.88	69 82	96 70	79.9	2.7486	1.350	112
113	.06493	454.5	14.42	1.51	15.93	27.12	72.08	99.20	80.9	2.8280	1 389	113
114	.06700	469.0	14.45	1.55	16.00	27 36	74.40	101.76	81.9	2.9094	1.429	114
115	0.06913	483.9	14.47	1.61	16.08	27 60	76.80	104.40	82.9	2.9929	1.470	**115**
116	.07134	499.4	14.50	1.66	16.16	27 84	79.29	107.13	83.9	3.0784	1.512	116
117	.07361	515.3	14.52	1.72	16.24	28 08	81.84	109.92	84.9	3.1660	1.555	117
118	.07600	532 0	14.55	1.77	16.32	28 32	84.53	112.85	85.9	3.2576	1.600	118
119	.07840	548.8	14.57	1.84	16.41	28.56	87 24	115.80	86.9	3.3492	1.645	119
120	0.08093	566.5	14.60	1.90	16.50	28.80	90.09	118.89	87.9	3.4449	1.692	**120**
121	.08348	584.4	14 62	1.96	16.58	29.04	92.97	122.01	88.9	3.5406	1.739	121
122	.08616	603.1	14 65	2.03	16.68	29.28	95.99	125.27	89.9	3.6404	1.788	122
123	.08892	622.4	14.67	2 10	16.77	29.52	99.11	128.63	90.9	3 7422	1.838	123
124	.09175	642.3	14.70	2.17	16.87	29 76	102.30	132.06	91.9	3.8460	1 889	124
125	0 09466	662.6	14.72	2 24	16.96	30 00	105.59	135.59	92 9	3.9519	1.941	**125**
126	.09770	683.9	14.75	2.31	17.06	30 24	109.02	139.26	93.9	4 0618	1.995	126
127	.1008	705.6	14.77	2.40	17.17	30.48	112.53	143.01	94.9	4.1718	2 049	127
128	.1040	728.0	14.80	2.47	17.27	30.72	116.15	146.87	95.9	4 2858	2.105	128
129	.1074	751.8	14.83	2 55	17.38	30 96	120.00	150.96	96.9	4.4039	2.163	129
130	0.1107	774.9	14.85	2.64	17.49	31.20	123.73	154.93	97.9	4.5220	2.221	**130**
131	.1143	800.1	14.88	2.73	17.61	31.45	128.81	159.26	98.9	4 6441	2 281	131
132	.1180	826.0	14.90	2 83	17.73	31.69	131.99	163.68	99.9	4 7703	2.343	132
133	.1218	852.6	14.93	2.92	17 85	31 93	136 31	168.24	100 9	4 8986	2.406	133
134	.1257	879.9	14 95	3 02	17.97	32.17	140.72	172.89	101.9	5.0289	2.470	134
135	0.1297	907.9	14.98	3 12	18 10	32 41	145.26	177.67	102.9	5.1633	2.536	**135**
136	.1339	937.3	15.00	3.23	18 23	32 65	150.02	182.67	103.9	5.2997	2.603	136
137	.1382	967.4	15.03	3 33	18.36	32.89	154.91	187 80	104.9	5 4402	2.672	137
138	.1427	998.9	15 05	3.45	18.50	33.13	160.01	193.14	105.9	5.5827	2.742	138
139	.1473	1031.1	15.08	3.57	18.65	33.37	165.24	198.61	106.9	5.7293	2.814	139

[a]Compiled by W. M. Sawdon and extended by John A. Goff.

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (CONTINUED)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Liquid Water Btu per Lb h'_w	Saturation Pressure p_s: In. of Hg	Lb per Sq In.	Temp Deg F
140	0.1521	1064.7	15 10	3.69	18.79	33 61	170 69	204.30	107 9	5.8779	2 887	**140**
141	.1570	1099.0	15.13	3.81	18 94	33 85	176 26	210.11	108 9	6.0306	2.962	141
142	.1622	1135.4	15.15	3.95	19.10	34 09	182.17	216.26	109.9	6.1874	3.039	142
143	.1675	1172 5	15.18	4 08	19 26	34 33	188.20	222.53	110.9	6.3482	3.118	143
144	.1730	1211.0	15.20	4.23	19.43	34.57	194.45	229.02	111.9	6.5111	3.198	144
145	0.1787	1250 9	15.23	4 37	19 60	34 81	200.95	235.76	112 9	6 6781	3.280	**145**
146	.1846	1292.2	15.25	4.53	19 78	35.05	207.66	242.71	113.9	6.8471	3.363	146
147	.1908	1335.6	15.28	4 68	19.96	35 29	214.73	250.02	114 9	7.0222	3.449	147
148	.1971	1379.7	15.30	4.85	20.15	35.53	221.90	257.43	115.9	7 1993	3.536	148
149	.2037	1425.9	15.33	5.02	20.35	35 77	229 43	265.20	116 9	7.3805	3.625	149
150	0.2105	1473.5	15.35	5 20	20.55	36.02	237 17	273.19	117.9	7 5658	3 716	**150**
151	.2176	1523.2	15.38	5.38	20.76	36.26	245 28	281.54	118.9	7 7551	3.809	151
152	.2250	1575.0	15.40	5 57	20 97	36.50	253.71	290.21	119 9	7 9485	3 904	152
153	.2327	1628.9	15.43	5 77	21.20	36.74	262 51	299 25	120 9	8.1460	4 001	153
154	.2407	1684 9	15.45	5 98	21.43	36.98	271 63	308.61	121.9	8.3476	4.100	154
155	0.2490	1743.0	15.48	6.19	21 67	37.22	281.12	318 34	122 9	8 5532	4 201	**155**
156	.2577	1803.9	15.50	6.43	21.93	37 46	291 05	328 51	123 9	8.7650	4.305	156
157	.2667	1866.9	15 53	6.66	22.19	37.70	301.24	339 04	124.9	8.9788	4 410	157
158	2761	1932.7	15.56	6.90	22.46	37.94	312.08	350 02	125.9	9.1986	4 518	158
159	.2858	2000.6	15.58	7.16	22.74	38.18	323 18	361.36	126 9	9.4206	4.627	159
160	0.2961	2072.7	15.61	7.42	23.03	38.43	334.95	373.38	127.9	9.6486	4 739	**160**
161	.3067	2146 9	15 63	7.70	23.33	38 67	347.09	385 76	128.9	9.8807	4 853	161
162	.3179	2225.3	15 66	7.99	23.65	38.91	359.89	398.80	129.9	10.119	4 970	162
163	.3295	2306.5	15.68	8.30	23.98	39.15	373.19	412.34	130.9	10.361	5.089	163
164	.3416	2391.2	15.71	8.62	24.33	39 39	387 03	426.42	131.9	10.608	5.210	164
165	0.3544	2480.8	15.73	8.96	24.69	39.63	401 71	441.34	132 9	10.860	5.334	**165**
166	.3677	2573.9	15.76	9.31	25.07	39.87	416 94	456.81	133 9	11.117	5 460	166
167	.3817	2671.9	15.78	9 68	25.46	40.11	433 00	473.11	134.9	11.379	5 589	167
168	.3964	2774.8	15.81	10.07	25.88	40.35	449 83	490.18	135.9	11.646	5.720	168
169	.4118	2882.6	15 83	10.48	26.31	40.59	467.52	508.11	136.9	11.919	5.854	169
170	0.4280	2996.0	15.86	10.91	26.77	40.83	486 08	526.91	137.9	12.196	5.990	**170**
171	.4451	3115.7	15.88	11.36	27.24	41 07	505.72	546.79	138.9	12 480	6 130	171
172	.4631	3241.7	15.91	11.83	27.74	41 32	526.36	567.68	139 9	12 770	6.272	172
173	.4821	3374.7	15.93	12.35	28.28	41.56	548.20	589.76	140 9	13 065	6.417	173
174	.5022	3515.4	15.96	12.88	28.84	41.80	571.25	613 05	141.9	13 366	6.565	174

[a]Compiled by W. M Sawdon and extended by John A. Goff.

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (CONCLUDED)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Liquid Water Btu per Lb h'_w	Saturation Pressure p_s: In. of Hg	Lb per Sq In.	Temp Deg F
175	0.5235	3664.5	15.98	13.45	29.43	42 04	595 74	637.78	142 9	13.674	6.716	**175**
176	.5459	3821 3	16 01	14.04	30.05	42 28	621.45	663.73	143.9	13 985	6 869	176
177	.5697	3987.9	16.03	14.68	30.71	42.52	648 83	691.35	144.9	14.303	7.025	177
178	.5949	4164.3	16 06	15.35	31.41	42.76	677 77	720.53	145 9	14 627	7.184	178
179	.6215	4350.5	16 08	16.07	32.15	43 00	708 39	751 39	146 9	14 954	7.345	179
180	0.6501	4550.7	16.11	16.83	32.94	43 24	741.24	784.48	147 9	15.290	7.510	**180**
181	.6805	4763.5	16.13	17.65	33.78	43.49	776.25	819.74	148.9	15 632	7.678	181
182	.7131	4991.7	16.16	18.52	34.68	43.73	813.72	857.45	149.9	15.981	7.849	182
183	.7481	5236.7	16.18	19.47	35.65	43.97	854.03	898 00	150 9	16.337	8.024	183
184	.7854	5497 8	16.21	20.46	36.67	44.21	896.93	941.14	151.9	16.697	8.201	184
185	0.8258	5780.6	16 23	21.55	37.78	44.45	943.48	987.93	152.9	17.066	8.382	**185**
186	.8693	6085.1	16 26	22.72	38.98	44 69	993.52	1038.21	153.9	17.440	8.566	186
187	.9162	6413.4	16.28	23.99	40.27	44.93	1047.58	1092.51	154.9	17.821	8.753	187
188	.9673	6771.1	16 31	25.36	41.67	45.18	1106.40	1151.58	155.9	18.210	8.944	188
189	1.0227	7158.9	16.34	26.70	43.04	45.42	1170.62	1216.04	157.0	18.605	9.138	189
190	1.083	7581.0	16.36	28.49	44.85	45.66	1239.71	1285.37	158 0	19.008	9.336	**190**
191	1.150	8050.0	16.39	30.29	46.68	45.90	1316.98	1362.88	159.0	19.419	9.538	191
192	1.224	8568.0	16.41	32.29	48 70	46.14	1402.21	1448.35	160 0	19.839	9.744	192
193	1.306	9142.0	16.44	34.49	50.93	46.38	1496.81	1543.19	161.0	20.266	9.954	193
194	1.397	9779.0	16.46	36.96	53 42	46.62	1601.66	1648.28	162 0	20.702	10.168	194
195	1.499	10493.0	16.49	39.71	56.20	46.86	1719.35	1766.21	163.0	21.144	10 385	**195**
196	1.613	11291.0	16.51	42.80	59 31	47.10	1850.76	1897.86	164.0	21.592	10.605	196
197	1.742	12194.0	16.54	46.31	62.85	47.34	1999.64	2046.98	165 0	22.048	10.829	197
198	1.890	13230.0	16.56	50.32	66.88	47.59	2170.29	2217.88	166.0	22.512	11.057	198
199	2.061	14427.0	16.59	54.95	71.54	47.83	2367.68	2415.51	167.0	22.984	11.289	199
200	2.261	15827.0	16.61	60.38	76.99	48 07	2598.34	2646.41	168 0	23.465	11 525	**200**

[a]Compiled by W. M. Sawdon and extended by John A. Goff.

This statement lacks thermodynamic soundness due to actual departures from Dalton's Law, but has real practical merit as an approximation.

Example 3. Calculate the humidity ratio of saturated moist air at 68 F, 30 in. Hg.

Solution. The saturation pressure of pure water at 68 F from Table 6 is 0.68980 in. Hg; hence,

$$W_s = \frac{0.62193 \times 0.68980}{29.3102} = 0.01464 \text{ (pound per pound of dry air).}$$

It is also frequently stated that moist air is saturated when the space (volume) occupied by it contains the *maximum weight of water vapor* at the given temperature. This means that any additional water would have to be in the liquid or solid phase. But under proper circumstances the water vapor can be *supersaturated*, in which case the space occupied by the mixture can contain more than the *maximum possible* water vapor. The statement is therefore meaningless as a definition of saturation.

A precise definition must necessarily refer to the co-existence of at least two distinct phases, say, liquid and vapor. These can only co-exist in stable equilibrium if evaporation of the liquid or condensation of the vapor under conditions of constant total volume and constant total internal energy would have to involve a decrease of total entropy. This would be the situation if, and only if, the pressure, the temperature, and each component chemical potential has the same value in each phase.

In the case of moist air, the general conditions for saturation previously stated can be deduced from Equation 9 together with available data on the solubility of air in the liquid. They can be reduced to the form,

$$W_s = 0.62193 \frac{p_s'}{P - p_s'} \tag{13a}$$

where

$$p_s' = \frac{(PF)\ (DF)}{(RF)}\, p_s \tag{13b}$$

The liquid (or solid) phase will contain a small amount of dissolved air and the Raoult factor (*RF*) expresses the effect of this dissolved air in lowering the vapor pressure in accordance with Raoult's Law. The Poynting factor (*PF*) accounts for the fact that the very presence of dry air requires the liquid (or solid) to support a higher pressure at saturation than it would if no dry air were present. The Dalton factor (*DF*) expresses the effect of intermolecular forces in the vapor phase. All three factors depend more or less on pressure as well as on temperature.

The Raoult and Poynting factors are calculable. The order of magnitude of the Dalton factor can now be determined by computing its value at one temperature and pressure using the information previously referred to, namely, $2A_{aw} = 0.075\ (A_{aa} + A_{ww})$. At 68 F, 29.921 in. Hg, for example,

$$p_s' = \frac{1.00073 \times 1.0052}{1.00002}\, p_s$$

This indicates departures from Dalton's Law of the order of 0.5 per cent. The data in Table 6 which are based on an assumed value of unity for the

Dalton factor have not been revised pending final results on the measurement of the interaction constant [10].

Relative Humidity

The ratio of actual humidity ratio W to the saturation humidity ratio W_s corresponding to the actual temperature and the observed pressure is denoted by the symbol μ and may be called alternatively *degree of saturation* or *percent saturation;* thus,

$$W = \mu \quad W_s \tag{14}$$

Example 4. Air is to be maintained at 70 F, 40 per cent saturation when outside air is at 0 F, 70 per cent. The observed pressure may be taken to be 29.921 in. Hg. Find the weight of water to be added to each pound of dry air using Table 6.

Solution. The desired humidity ratio is $0.40 \times 0.01574 = 0.006296$ while that of outside air is $0.70 \times 0.0007852 = 0.000550$. Hence the weight of water to be added is $0.006296 - 0.000550 = 0.005746$ lb per pound dry air.

Under Dalton's Law the water vapor exerts a partial pressure p_w which may be calculated from the given humidity ratio W and the observed pressure P by means of Equation 11. The ratio of this partial pressure p_w to the saturation pressure of pure water p_s corresponding to the actual temperature is called *relative humidity* and may be denoted by the symbol Φ; thus,

$$\Phi = \frac{p_w}{p_s} \tag{15}$$

The relation between μ and Φ is obtained directly from Equations 11 and 12 and is

$$\mu = \left(\frac{P - p_s}{P - p_w}\right) \Phi \tag{15a}$$

whence it is clear that for ordinary temperatures where p_s and therefore p_w are small compared with P, the two are approximately equal.

As an aid in quickly translating degree of saturation μ into relative humidity Φ, the following empirical equation may be substituted for Equation 15a:

$$\Phi = \mu + \alpha \cdot \mu (1 - \mu) \tag{15b}$$

where α depends upon temperature *for standard atmospheric pressure,* as shown by the values in Table 7.

Within the limits of accuracy of (15b) this may also be written

$$\mu = \Phi - \alpha \cdot \Phi (1 - \Phi) \tag{15c}$$

and used to translate relative humidity Φ into degree of saturation μ.

TABLE 7. PERCENTAGE DIFFERENCES CORRESPONDING TO TEMPERATURE FOR EQUATION 15b

TEMP F t	PER CENT α	TEMP F t	PER CENT α	TEMP F t	PER CENT α	TEMP F t	PER CENT α
5	0.16	30	0.55	55	1.47	80	3.51
10	0.21	35	0.68	60	1.76	85	4.14
15	0.27	40	0.83	65	2.10	90	4.86
20	0.34	45	1.01	70	2.50	95	5.70
25	0.44	50	1.22	75	2.97	100	6.67

For example, corresponding to 40 per cent saturation at 100 F, the relative humidity is $0.40 + 0.0667 \times 0.40 \times 0.60 = 0.416$ or 41.6 per cent (15b). Conversely, corresponding to a relative humidity of 41.6 per cent, the degree of saturation is $0.416 - 0.0667 \times 0.416 \times 0.584 = 0.400$ or 40 per cent (15c).

Dew-point

If moist air is cooled at constant humidity ratio W and constant observed pressure P, a temperature will be reached at which the air just becomes saturated and formation of a liquid (or solid) phase just commences. This temperature is called the *dew-point* corresponding to the given humidity ratio and observed pressure.

Example 5. Find the dew-point of the humidified air of Example 4.

Solution. The given humidity ratio is 0.006296 which is the saturation value at 44.96 F (Table 6, assuming the total pressure to be 29.921 in. Hg). This is therefore the dew-point of the humidified air.

Example 6. Find the degree of saturation of air having a temperature of 90 F, a dew-point of 60 F.

Solution. Assuming the total pressure to be 29.921 in. Hg, the humidity ratio is given in Table 6 as 0.01103 lb per pound dry air. The saturation humidity ratio at 90 F is 0.03102 lb per pound dry air; hence the degree of saturation is $0.01103 \div 0.03102 = 0.355$ or 35.5 per cent.

Volume

The volume of moist air *per pound of dry air* contained in it is a very useful quantity. It should not be called specific volume; for the adjective *specific* should properly refer to volume per pound of *mixture.* Using Equations 10a and 14 an expression for the volume per pound of dry air is obtained, namely,

$$v = \frac{B_a T}{P} + \mu \left(\frac{W_s B_w T}{P} \right) \qquad (16)$$

Example 7. Find the volume (per pound of dry air) of the humidified air of Example 4.

Solution. $v = \left(\frac{53.35 \times 529.7}{29.92 \times 0.49115 \times 144} \right) + 0.40 \left(\frac{0.01574 \times 85.78 \times 529.7}{29.92 \times 0.49115 \times 144} \right)$
$= 13.354 + 0.40 \times 0.338 = 13.489$ cu ft per pound dry air.

Equation 16 is linear in degree of saturation μ and of the form

$$v = v_a + \mu \, v_{as} \qquad (17)$$

where v_a denotes specific volume of dry air at temperature T and pressure P; and v_{as} denotes the difference between this and the volume of the saturated mixture per pound of dry air v_s. Strict linearity is, of course, a result of the use of Dalton's Law; but it is expected that it can be retained as a very close approximation even when the abandonment of Dalton's Law becomes possible.

Example 8. Work Example 7 using Table 6.

Solution. $v = 13.34 + (0.40 \times 0.34) = 13.48$ cu ft per pound dry air.

By putting $\mu = 1$ (100 per cent saturation) in Equation 16 an expression for v_s, the volume of saturated air per pound of dry air, is obtained. Values for standard atmospheric pressure (29.921 in. Hg) are listed in Table 6.

Often it is preferred to express this information in terms of *density*, that is, weight of saturated air per unit volume. This can easily be done by dividing v_s (volume of saturated air per pound of dry air) *into* $(1 + w_s)$ (*weight* of saturated air per pound of dry air). Thus, at 100 F, 29.921 in. Hg, the density of saturated air is, from Table 6, $1.04293 \div 15.07 = 0.06921$ lb per cubic foot.

Values in Table 9 are intended to aid in determining the density of saturated air at different pressures. Values for temperatures and pressures other than those listed can be obtained by linear interpolation which is aided by the next to last column of figures. Thus, at 100 F, 29.921 in. Hg, the density of saturated air is, from Table 9, $0.06818 + (4.21 \times 0.00024) = 0.06919$ lb per cubic foot, in approximate agreement with Table 6.

A column of figures is included in Table 9 giving the approximate average increase in density per degree wet-bulb depression. This makes it easy to calculate a value for the density of moist air taking into account its moisture content as well as its temperature and pressure.

Volume Chart

A volume chart drawn for a total pressure of 29.921 in. Hg will be found in the envelope attached to the inside back cover of this book. On this chart values of volume per pound of dry air v are plotted as abscissa against values of humidity ratio W as ordinate. The chart is self-explanatory.

Enthalpy

Thermodynamically, Equation 10a implies that the specific enthalpies of dry air and water vapor are independent of pressure and that the enthalpy of moist air (per pound of dry air) is the sum of separate contributions from the dry air and water vapor according to the simple equation

$$h = h_a + \mu\,(W_s h_w) \tag{18}$$

Equation 18 is also linear in degree of saturation μ and of the form

$$h = h_a + \mu\, h_{as} \tag{19}$$

where h_a denotes the specific enthalpy of dry air at the given temperature and total pressure; and h_{as} denotes the difference between this and the enthalpy of the saturated mixture per pound of dry air h_s. Provisional values are listed in Table 6.

Example 9. Find the enthalpy (per pound of dry air) of air at 96 F, 60 per cent saturation and 29.921 in. Hg.

Solution. Using Table 6, $h = 23.04 + (0.60 \times 41.58) = 47.99$ Btu per pound dry air.

Thermodynamic Wet-bulb Temperature

If liquid (or solid) water be injected into an air stream it will evaporate and thus increase the humidity ratio of the air. Enough water may be injected to saturate the air. If the process is one of steady flow with *observed pressure constant*; if it is *adiabatic*; and if the temperature at which the air reaches saturation *coincides* with the temperature of the

liquid (or solid) as added; then the common temperature is called *thermodynamic wet-bulb temperature.* This lengthy definition is easily visualized by referring to Fig. 1 in which h'_w denotes the specific enthalpy of the liquid (or solid) as injected.

The process being adiabatic, weight and energy accountings give

$$h_1 + (W_s - W_1)\, h'_w = h_s \tag{20}$$

If the temperature of the saturated air at the leaving section *coincides* with that of the injected liquid (or solid), then W_s, h'_w and h_s are functions of a single temperature t' which can therefore be determined by solving (20). This is the thermodynamic wet-bulb temperature corresponding to conditions at the entering section.

Example 10. Find the thermodynamic wet-bulb temperature of dry air at 80 F and 29.921 in. Hg.

Solution. Using Table 6, the equation to be solved is $19.19 + (W_s - 0)\, h'_w = h_s$.

A trial value is obtained by ignoring the small quantity $(W_s - 0)\, h'_w$; it is 48 F corresponding to $h_s = 19.19$ Btu per pound dry air. A final value of 48.26 F is then obtained from $h_s = 19.19 + (0.007072 \times 16.1) = 19.30$ Btu per pound dry air.

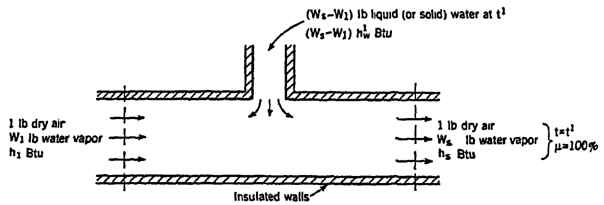

FIG 1. DIAGRAM ILLUSTRATING THERMODYNAMIC WET-BULB TEMPERATURE

Example 11. Find the degree of saturation of moist air at 90 F dry-bulb, 70 F wet-bulb and 29.921 in. Hg.

Solution. Using Table 6, the equation to be solved is $(21.59 + 34.11\,\mu) + (0.01574 - 0.03102\mu) \times 38.0 = 33.96$ from which

$$\mu = 11.77 \div 32.93 = 0.357 \text{ or } 35.7 \text{ per cent.}$$

It is important to note in connection with Equation 20 that the enthalpy per pound dry air is not constant along a line of constant thermodynamic wet-bulb temperature on account of the term $(W_s - W_1)\, h'_w$. In rough calculations, however, it is usually legitimate to ignore this term.

Thermodynamic wet-bulb is an important property of moist air because it is approximately the temperature indicated by the wet-bulb psychrometer. This instrument consists of a thermometer with its bulb covered with gauze moistened with clean liquid water. It is whirled through the air until the thermometer reads a steady temperature. At this point, the temperature of the liquid evaporating from the wetted surface has adjusted itself so that the air immediately in contact with the liquid is brought to saturation at the same temperature. Unfortunately, the mixing taking place beyond the liquid surface is not adiabatic; for one reason because the wet-bulb *sees* objects at dry-bulb temperature

and considerable heat is transferred by radiation. Also there are other reasons why the readings of the psychrometer depend upon the design of the instrument, the velocity of the air stream in which it is placed, and other factors. Therefore wet-bulb temperature as indicated by the psychrometer cannot be regarded as a thermodynamic property; in fact, the approximate agreement with thermodynamic wet-bulb temperature in the case of moist air has been shown to be largely fortuitous [4].

Mollier Diagram

A thermodynamic analysis of any air conditioning process consists in writing: (1) a weight balance for the dry air; (2) a weight balance for the water; (3) an energy balance. The first is reduced to its simplest form by basing all quantities on one pound of dry air. The second is the most simply expressed in terms of humidity ratio, or weight of water per pound of dry air. Since most air conditioning processes are of the steady flow type in which the thermal energy convected with the fluid is its enthalpy, the third is most simply expressed in terms of enthalpy per pound of dry air. It is clear, therefore, that humidity ratio W and enthalpy per pound of dry air h are fundamental coordinates. Their use for the purpose of graphical representation is due to Mollier [3]. A convenient modification of the Mollier diagram devised by Goff is obtained by taking humidity ratio W as ordinate and *reduced enthalpy* $(h - 1000W)$ as abscissa, as shown in the chart enclosed in the envelope attached to the inside back cover of this book.

The reasons for the use of the difference $(h - 1000W)$ as abscissa instead of h itself in the Mollier Diagram for Moist Air are the following: (1) it amounts to plotting on oblique coordinates and thus reduces to convenient proportions a diagram which would otherwise take the form of a scroll; (2) by the choice of the factor 1000 the necessary multiplication reduces to shifting the decimal point; (3) the ease with which the ordinate W can be multiplied by 1000 and added to the abscissa to obtain the enthalpy h makes it unnecessary to complicate the chart by a family of isenthalpic lines.

In the Mollier diagram, the lines inclined upward and slightly to the right are lines of constant (dry-bulb) temperature. They are straight under Dalton's Law but actually have slight curvature. The lines inclined upward to the left are lines of constant thermodynamic wet-bulb and are straight by definition. The dry-bulb and wet-bulb lines meet at the saturation curve and coincide in the region to the left of this curve. This region is divided into three sub-regions by the narrow *wedge* with apex at the junction of the 32 F wet-bulb and dry-bulb lines. Above the wedge, the mixture consists of two distinct phases, saturated vapor and saturated liquid. At point A, for example, the temperature is 60 F and the vapor phase contains 0.01103 pounds of water vapor per pound of dry air from Table 6. From the Mollier diagram, $W = 0.016$ lb, leaving 0.00497 lb per pound dry air in the liquid phase. The total enthalpy of the mixture is $10.51 + (1000 \times 0.016) = 26.51$ Btu per pound dry air of which $15.34 + (1000 \times 0.01103) = 26.37$ Btu per pound dry air is contributed by the vapor phase.

Within the wedge, the mixture consists of three distinct phases, saturated vapor, saturated solid and saturated liquid. The temperature is

Table 8. Properties of Saturated Steam: Pressure Table[a]

Abs. Press. In. Hg. p	Temp F t	Specific Volume: Sat. Liquid V_f	Specific Volume: Sat. Vapor V_g	Enthalpy: Sat. Liquid h_f	Enthalpy: Evap. h_{fg}	Enthalpy: Sat Vapor h_g	Entropy: Sat. Liquid S_f	Entropy: Evap. S_{fg}	Entropy: Sat. Vapor S_g	Abs. Press. In. Hg p
0.25	40.23	0 01602	2423.7	8.28	1071.1	1079.4	0.0166	2.1423	2.1589	**0.25**
0.50	58 80	0.01604	1256.4	26.86	1060.6	1087.5	0.0532	2.0453	2.0985	**0.50**
0.75	70.43	0.01606	856.1	38.47	1054.0	1092.5	0.0754	1.9881	2.0635	**0.75**
1.00	79.03	0.01608	652.3	47.05	1049.2	1096.3	0.0914	1.9473	2.0387	**1.00**
1.5	91.72	0.01611	444.9	59.71	1042.0	1101.7	0.1147	1 8894	2.0041	**1.5**
2	101.14	0.01614	339.2	69.10	1036.6	1105.7	0.1316	1.8481	1.9797	**2**
4	125.43	0.01622	176.7	93.34	1022.7	1116.0	0.1738	1.7476	1.9214	**4**
6	140.78	0.01630	120.72	108.67	1013.6	1122 3	0 1996	1.6881	1.8877	**6**
8	152.24	0.01635	92.16	120.13	1006.9	1127.0	0.2186	1.6454	1.8640	**8**
10	161.49	0.01640	74.76	129.38	1001.4	1130.8	0.2335	1.6121	1.8456	**10**
12	169.28	0.01644	63 03	137 18	996 7	1133.9	0.2460	1.5847	1.8307	**12**
14	176.05	0.01648	54 55	143.96	992 6	1136.6	0.2568	1.5613	1.8181	**14**
16	182.05	0.01652	48 14	149.98	988 9	1138.9	0.2662	1.5410	1.8072	**16**
18	187.45	0.01655	43.11	155.39	985.7	1141 1	0.2746	1.5231	1.7977	**18**
20	192.37	0.01658	39.07	160.33	982.7	1143.0	0.2822	1.5069	1.7891	**20**
22	196.90	0 01661	35.73	164.87	979.8	1144.7	0.2891	1.4923	1 7814	**22**
24	201.09	0.01664	32.94	169.09	977.2	1146.3	0.2955	1.4789	1.7744	**24**
26	205.00	0.01667	30.56	173.02	974.8	1147.8	0 3014	1.4665	1.7679	**26**
28	208.67	0.01669	28.52	176.72	972.5	1149.2	0.3069	1.4550	1.7619	**28**
30	212.13	0.01672	26.74	180.19	970.3	1150.5	0.3122	1.4442	1.7564	**30**
Lb/Sq In.										Lb/Sq In.
14.696	212.00	0.01672	26 80	180.07	970.3	1150 4	0.3120	1.4446	1.7566	**14.696**
16	216.32	0.01674	24 75	184.42	967.6	1152.0	0.3184	1.4313	1.7497	**16**
18	222.41	0.01679	22.17	190.56	963.6	1154.2	0.3275	1.4128	1.7403	**18**
20	227.96	0.01683	20.089	196.16	960 1	1156.3	0.3356	1.3962	1.7319	**20**
22	233.07	0.01687	18.375	201.33	956 8	1158.1	0.3431	1.3811	1.7242	**22**
24	237.82	0.01691	16.938	206.14	953 7	1159.8	0.3500	1.3672	1.7172	**24**
26	242.25	0.01694	15.715	210.62	950.7	1161.3	0.3564	1.3544	1.7108	**26**
28	246.41	0.01698	14.663	214.83	947.9	1162.7	0.3623	1.3425	1.7048	**28**
30	250.33	0.01701	13 746	218 82	945.3	1164 1	0.3680	1.3313	1 6993	**30**
32	254.05	0.01704	12.940	222.59	942 8	1165 4	0.3733	1.3209	1.6941	**32**
34	257.58	0.01707	12.226	226.18	940 3	1166 5	0 3783	1.3110	1.6893	**34**
36	260.95	0.01709	11 588	229.60	938 0	1167.6	0.3831	1.3017	1.6848	**36**
38	264.16	0.01712	11.015	232 89	935.8	1168 7	0.3876	1.2929	1.6805	**38**
40	267.25	0.01715	10 498	236.03	933.7	1169.7	0 3919	1.2844	1.6763	**40**
42	270.21	0.01717	10.029	239.04	931.6	1170.7	0.3960	1.2764	1.6724	**42**
44	273.05	0.01720	9.601	241.95	929.6	1171.6	0.4000	1.2687	1.6687	**44**
46	275.80	0.01722	9.209	244 75	927.7	1172 4	0.4038	1.2613	1.6652	**46**
48	278.45	0.01725	8 848	247 47	925.8	1173.3	0.4075	1.2542	1.6617	**48**
50	281.01	0.01727	8.515	250.09	924.0	1174 1	0.4110	1.2474	1.6585	**50**
52	283.49	0.01729	8 208	252.63	922.2	1174.8	0.4144	1.2409	1.6553	**52**
54	285.90	0.01731	7.922	255 09	920 5	1175.6	0.4177	1.2346	1.6523	**54**
56	288.23	0.01733	7.656	257.50	918.8	1176.3	0.4209	1.2285	1 6494	**56**
58	290.50	0.01736	7.407	259.82	917.1	1176.9	0.4240	1.2226	1.6466	**58**
60	292.71	0.01738	7 175	262.09	915 5	1177.6	0.4270	1.2168	1 6438	**60**
62	294.85	0.01740	6.957	264 30	913.9	1178 2	0.4300	1.2112	1.6412	**62**
64	296.94	0.01742	6.752	266.45	912.3	1178.8	0.4328	1.2059	1.6387	**64**
66	298.99	0.01744	6 560	268.55	910 8	1179.4	0.4356	1.2006	1.6362	**66**
68	300 98	0.01746	6.378	270.60	909.4	1180.0	0.4383	1.1955	1.6338	**68**
70	302.92	0.01748	6.206	272.61	907.9	1180.6	0.4409	1.1906	1.6315	**70**
72	304.83	0.01750	6.044	274.57	906.5	1181.1	0.4435	1.1857	1.6292	**72**
74	306.68	0.01752	5 890	276.49	905.1	1181.6	0.4460	1.1810	1 6270	**74**
76	308.50	0.01754	5 743	278.37	903.7	1182.1	0.4484	1.1764	1 6248	**76**
78	310.29	0.01755	5.604	280.21	902.4	1182.6	0.4508	1.1720	1.6228	**78**
80	312.03	0.01757	5.472	282.02	901.1	1183 1	0.4531	1.1676	1.6207	**80**
82	313.74	0.01759	5.346	283.79	899.7	1183.5	0.4554	1.1633	1.6187	**82**
84	315.42	0.01761	5.226	285.53	898.5	1184.0	0.4576	1.1592	1.6168	**84**
86	317.07	0.01762	5.111	287.24	897.2	1184.4	0.4598	1.1551	1.6149	**86**
88	318.68	0.01764	5.001	288.91	895.9	1184.8	0.4620	1.1510	1.6130	**88**
90	320.27	0.01766	4.896	290.56	894.7	1185.3	0 4641	1.1471	1.6112	**90**
92	321.83	0.01768	4 796	292.18	893.5	1185.7	0.4661	1.1433	1.6094	**92**
94	323 36	0.01769	4.699	293.78	892.3	1186.1	0.4682	1.1394	1.6076	**94**
96	324.87	0.01771	4.606	295.34	891.1	1186.4	0.4702	1.1358	1.6060	**96**
98	326.35	0.01772	4.517	296.89	889 9	1186.8	0.4721	1.1322	1.6043	**98**
100	327.81	0.01774	4.432	298.40	888.8	1187.2	0.4740	1.1286	1.6026	**100**
150	358.42	0.01809	3.015	330.51	863.6	1194.1	0.5138	0.0556	1.5694	**150**
200	381.79	0.01839	2.288	355.36	843.0	1198.4	0.5435	1.0018	1.5453	**200**
300	417.33	0.01890	1.5433	393.84	809.0	1202 8	0 5879	0.9225	1.5104	**300**
400	444.59	0.0193	1.1613	424.0	780.5	1204 5	0.6214	0.8630	1.4844	**400**
500	467.01	0.0197	0.9278	449.4	755.0	1204.4	0.6487	0.8147	1.4634	**500**

[a]Reprinted by permission from *Thermodynamic Properties of Steam*, by J. H. Keenan and F G. Keyes, published by John Wiley and Sons, Inc , 1936 edition.

TABLE 8. PROPERTIES OF SATURATED STEAM: TEMPERATURE TABLE[a]

Temp F t	Abs. Pressure: Lb per Sq In. p	Abs. Pressure: In. Hg p	Specific Volume: Sat. Liquid v_f	Specific Volume: Evap. v_{fg}	Specific Volume: Sat Vapor v_g	Enthalpy: Sat. Liquid h_f	Enthalpy: Evap. h_{fg}	Enthalpy: Sat. Vapor h_g	Entropy: Sat. Liquid s_f	Entropy: Evap. s_{fg}	Entropy: Sat. Vapor s_g	Temp F t
32	0.08854	0.1803	0 01602	3306	3306	0.00	1075.8	1075.8	0.0000	2.1877	2.1877	32
33	0.09223	0.1878	0.01602	3180	3180	1.01	1075.2	1076.2	0.0020	2.1821	2.1841	33
34	0.09603	0.1955	0 01602	3061	3061	2.02	1074.7	1076.7	0.0041	2.1764	2.1805	34
35	0.09995	0.2035	0 01602	2947	2947	3.02	1074.1	1077.1	0.0061	2.1709	2.1770	35
36	0.10401	0.2118	0.01602	2837	2837	4 03	1073.6	1077.6	0.0081	2.1654	2.1735	36
37	0.10821	0.2203	0 01602	2732	2732	5.04	1073.0	1078.0	0.0102	2.1598	2.1700	37
38	0.11256	0.2292	0.01602	2632	2632	6.04	1072.4	1078.4	0.0122	2.1544	2.1666	38
39	0.11705	0 2383	0.01602	2536	2536	7.04	1071.9	1078.9	0.0142	2.1489	2.1631	39
40	0.12170	0.2478	0.01602	2444	2444	8.05	1071.3	1079.3	0.0162	2.1435	2.1597	40
41	0.12652	0.2576	0.01602	2356	2356	9 05	1070.7	1079.7	0.0182	2 1381	2.1563	41
42	0.13150	0.2677	0.01602	2271	2271	10.05	1070.1	1080.2	0.0202	2.1327	2.1529	42
43	0.13665	0.2782	0.01602	2190	2190	11.06	1069.5	1080.6	0.0222	2.1274	2.1496	43
44	0.14199	0.2891	0.01602	2112	2112	12 06	1068.9	1081.0	0.0242	2.1220	2.1462	44
45	0.14752	0.3004	0.01602	2036.4	2036.4	13.06	1068.4	1081.5	0.0262	2.1167	2.1429	45
46	0.15323	0.3120	0.01602	1964.3	1964.3	14.06	1067.8	1081.9	0.0282	2.1113	2.1395	46
47	0.15914	0 3240	0.01603	1895.1	1895.1	15.07	1067.3	1082.4	0.0302	2.1060	2.1362	47
48	0.16525	0.3364	0.01603	1828 6	1828 6	16.07	1066.7	1082.8	0.0321	2.1008	2.1329	48
49	0.17157	0.3493	0.01603	1764.7	1764.7	17.07	1066.1	1083.2	0.0341	2.0956	2.1297	49
50	0.17811	0.3626	0.01603	1703.2	1703.2	18.07	1065.6	1083.7	0.0361	2.0903	2.1264	50
51	0.18486	0.3764	0.01603	1644 2	1644.2	19.07	1065.0	1084.1	0.0380	2.0852	2.1232	51
52	0.19182	0.3906	0.01603	1587.6	1587.6	20.07	1064.4	1084.5	0.0400	2 0799	2.1199	52
53	0.19900	0.4052	0 01603	1533 3	1533.3	21.07	1063.9	1085.0	0.0420	2.0747	2.1167	53
54	0.20642	0.4203	0 01603	1481.0	1481.0	22.07	1063.3	1085.4	0.0439	2.0697	2.1136	54
55	0.2141	0.4359	0.01603	1430.7	1430.7	23.07	1062 7	1085.8	0.0459	2.0645	2.1104	55
56	0.2220	0.4520	0.01603	1382.4	1382.4	24.06	1062.2	1086.3	0.0478	2.0594	2 1072	56
57	0.2302	0.4686	0 01603	1335.9	1335.9	25 06	1061 6	1086.7	2.0497	2.0544	2.1041	57
58	0.2386	0 4858	0 01604	1291 1	1291.1	26.06	1061.0	1087.1	0.0517	2.0493	2.1010	58
59	0.2473	0 5035	0.01604	1248.1	1248.1	27.06	1060.5	1087.6	0.0536	2 0443	2.0979	59
60	0.2563	0 5218	0.01604	1206.6	1206.7	28.06	1059 9	1088.0	0.0555	2.0393	2.0948	60
61	0.2655	0.5407	0.01604	1166.8	1166.8	29.06	1059.3	1088.4	0.0574	2.0343	2.0917	61
62	0.2751	0.5601	0 01604	1128.4	1128.4	30.05	1058 8	1088.9	0.0593	2.0293	2.0886	62
63	0.2850	0.5802	0.01604	1091.4	1091.4	31.05	1058.2	1089.3	0.0613	2.0243	2.0856	63
64	0.2951	0.6009	0.01605	1055.7	1055.7	32.05	1057.6	1089.7	0.0632	2.0194	2.0826	64
65	0.3056	0.6222	0.01605	1021.4	1021.4	33.05	1057.1	1090.2	0.0651	2.0145	2.0796	65
66	0.3164	0.6442	0 01605	988 4	988 4	34.05	1056.5	1090.6	0.0670	2.0096	2.0766	66
67	0.3276	0.6669	0 01605	956.6	956.6	35.05	1056.0	1091.0	0.0689	2 0047	2.0736	67
68	0 3390	0.6903	0 01605	925.9	925.9	36.04	1055.5	1091.5	0.0708	1.9998	2.0706	68
69	0.3509	0.7144	0.01605	896 3	896.3	37.04	1054.9	1091.9	0.0726	1.9950	2 0676	69
70	0.3631	0.7392	0 01606	867.8	867.9	38.04	1054.3	1092.3	0.0745	1.9902	2.0647	70
71	0.3756	0.7648	0 01606	840 4	840.4	39.04	1053 8	1092.8	0.0764	1.9854	2.0618	71
72	0 3886	0.7912	0.01606	813.9	813.9	40.04	1053.2	1093.2	0.0783	1.9805	2.0588	72
73	0.4019	0 8183	0.01606	788.3	788.4	41.03	1052.6	1093.6	0.0802	1.9757	2.0559	73
74	0.4156	0.8462	0.01606	763 7	763.8	42.03	1052 1	1094.1	0.0820	1.9710	2.0530	74
75	0.4298	0.8750	0 01607	740.0	740 0	43.03	1051.5	1094.5	0.0839	1 9663	2.0502	75
76	0.4443	0.9046	0.01607	717.1	717.1	44.03	1050.9	1094.9	0.0858	1.9615	2.0473	76
77	0 4593	0.9352	0.01607	694.9	694.9	45.02	1050.4	1095.4	0.0876	1 9569	2.0445	77
78	0.4747	0 9666	0 01607	673 6	673.6	46 02	1049.8	1095.8	0.0895	1.9521	2.0416	78
79	0.4906	0.9989	0 01608	653.0	653.0	47.02	1049.2	1096.2	0.0913	1.9475	2.0388	79
80	0.5069	1.0321	0.01608	633.1	633.1	48.02	1048.6	1096.6	0.0932	1.9428	2.0360	80
81	0.5237	1.0664	0.01608	613.9	613.9	49.02	1048.1	1097.1	0.0950	1.9382	2 0332	81
82	0.5410	1.1016	0.01608	595.3	595 3	50.01	1047.5	1097.5	0.0969	1.9335	2 0304	82
83	0.5588	1.1378	0.01609	577.4	577 4	51.01	1046.9	1097.9	0.0987	1.9290	2.0277	83
84	0.5771	1.1750	0.01609	560.1	560.2	52.01	1046.4	1098.4	0.1005	1.9244	2 0249	84
85	0.5959	1.2133	0.01609	543 4	543.5	53.00	1045.8	1098.8	0.1024	1.9198	2.0222	85
86	0.6152	1.2527	0.01609	527 3	527.3	54 00	1045 2	1099.2	0.1042	1.9153	2.0195	86
87	0.6351	1.2931	0.01610	511.7	511.7	55.00	1044.7	1099.7	0.1060	1.9108	2.0168	87
88	0.6556	1.3347	0.01610	496 6	496.7	56.00	1044.1	1100.1	0.1079	1.9062	2.0141	88
89	0.6766	1.3775	0.01610	482.1	482.1	56.99	1043.5	1100.5	0.1097	1.9017	2.0114	89
90	0.6982	1.4215	0.01610	468.0	468.0	57.99	1042.9	1100 9	0.1115	1.8972	2.0087	90
91	0.7204	1.4667	0 01611	454.4	454.4	58.99	1042.4	1101.4	0.1133	1 8927	2 0060	91
92	0.7432	1.5131	0.01611	441.2	441.3	59 99	1041.8	1101 8	0.1151	1.8883	2.0034	92
93	0.7666	1.5608	0 01611	428.5	428.5	60.98	1041.2	1102.2	0.1169	1.8838	2.0007	93
94	0.7906	1.6097	0.01612	416.2	416.2	61.98	1040.7	1102 6	0.1187	1.8794	1.9981	94
95	0.8153	1.6600	0.01612	404 3	404.3	62 98	1040.1	1103 1	0.1205	1 8750	1.9955	95
96	0.8407	1.7117	0.01612	392 8	392 8	63 98	1039.5	1103.5	0.1223	1.8706	1 9929	96
97	0.8668	1 7647	0 01612	381.7	381.7	64.97	1038.9	1103 9	0.1241	1.8662	1.9903	97
98	0.8935	1.8192	0.01613	370.9	370.9	65.97	1038.4	1104.4	0.1259	1.8618	1 9877	98
99	0.9210	1.8751	0.01613	360.4	360.5	66.97	1037.8	1104 8	0.1277	1.8575	1.9852	99
100	0.9492	1.9325	0.01613	350.3	350 4	67.97	1037.2	1105 2	0.1295	1.8531	1.9826	100
101	0.9781	1.9915	0.01614	340.6	340.6	68.96	1036.6	1105.6	0.1313	1.8488	1.9801	101

[a]Reprinted by permission from *Thermodynamic Properties of Steam*, by J. H. Keenan and F. G. Keyes, published by John Wiley and Sons, Inc , 1936 edition.

TABLE 9. WEIGHT OF SATURATED AND PARTLY SATURATED AIR[a]

Dry-Bulb Temp Deg F	Weight of Saturated Air for Various Barometric and Hygrometric Conditions—Pounds per Cubic Foot							Approx. Average Increase in Weight per Deg Wet-Bulb Depression
	Barometric Pressure Inches of Mercury						Increase In Weight Per 0 1 in. Rise in Barometer	
	28 5	29 0	29.5	30 0	30 5	31 0		
30	0.07703	0.07839	0.07974	0.08110	0.08245	0.08381	0.00027	0.000017
32	0.07671	0.07806	0.07940	0.08075	0.08210	0.08345	0.00027	0.000017
34	0.07638	0.07772	0.07907	0.08041	0.08175	0.08310	0.00027	0.000018
36	0.07605	0.07739	0.07873	0.08007	0.08141	0.08274	0.00027	0.000018
38	0.07573	0.07706	0.07840	0.07973	0.08106	0.08239	0.00027	0.000019
40	0.07541	0.07674	0.07806	0.07939	0.08072	0.08205	0.00027	0.000019
42	0.07509	0.07641	0.07773	0.07905	0.08038	0.08170	0.00026	0.000020
44	0.07477	0.07609	0.07740	0.07872	0.08004	0.08135	0.00026	0.000020
46	0.07445	0.07576	0.07707	0.07838	0.07970	0.08101	0.00026	0.000021
48	0.07413	0.07544	0.07674	0.07805	0.07936	0.08066	0.00026	0.000021
50	0.07381	0.07512	0.07642	0.07772	0.07902	0.08032	0.00026	0.000022
52	0.07350	0.07479	0.07609	0.07739	0.07868	0.07998	0.00026	0.000023
54	0.07318	0.07447	0.07576	0.07706	0.07835	0.07964	0.00026	0.000023
56	0.07287	0.07415	0.07544	0.07673	0.07801	0.07930	0.00026	0.000024
58	0.07255	0.07383	0.07512	0.07640	0.07768	0.07896	0.00026	0.000025
60	0.07224	0.07352	0.07479	0.07607	0.07734	0.07862	0.00026	0.000026
62	0.07193	0.07320	0.07447	0.07574	0.07701	0.07828	0.00026	0.000027
64	0.07161	0.07288	0.07414	0.07541	0.07668	0.07794	0.00026	0.000028
66	0.07130	0.07256	0.07382	0.07508	0.07634	0.07760	0.00026	0.000029
68	0.07098	0.07224	0.07350	0.07475	0.07601	0.07727	0.00026	0.000030
70	0.07067	0.07192	0.07317	0.07442	0.07568	0.07693	0.00026	0.000031
72	0.07035	0.07160	0.07285	0.07410	0.07534	0.07659	0.00025	0.000032
74	0.07004	0.07128	0.07252	0.07377	0.07501	0.07625	0.00025	0.000033
76	0.06972	0.07096	0.07220	0.07343	0.07467	0.07591	0.00025	0.000034
78	0.06940	0.07064	0.07187	0.07310	0.07434	0.07557	0.00025	0.000036
80	0.06909	0.07032	0.07155	0.07277	0.07400	0.07523	0.00025	0.000037
82	0.06877	0.07000	0.07122	0.07244	0.07366	0.07489	0.00024	0.000039
84	0.06845	0.06967	0.07089	0.07211	0.07333	0.07454	0.00024	0.000040
86	0.06812	0.06934	0.07056	0.07177	0.07299	0.07420	0.00024	0.000042
88	0.06780	0.06901	0.07022	0.07143	0.07264	0.07385	0.00024	0.000043
90	0.06748	0.06868	0.06989	0.07109	0.07230	0.07351	0.00024	0.000045
92	0.06715	0.06835	0.06955	0.07075	0.07195	0.07316	0.00024	0.000047
94	0.06682	0.06801	0.06921	0.07041	0.07161	0.07280	0.00024	0.000049
96	0.06648	0.06768	0.06887	0.07006	0.07126	0.07245	0.00024	0.000051
98	0.06615	0.06734	0.06853	0.06972	0.07091	0.07209	0.00024	0.000053
100	0.06581	0.06700	0.06818	0.06937	0.07055	0.07174	0.00024	0.000055

[a]Approximate average decrease in weight per 0.1 F rise in dry-bulb temperature equals 0.000017 lb per cubic foot.

32 F; and the relative proportions of the three phases depend upon the location of the state point within the wedge. Below the wedge, the mixture consists of saturated vapor and saturated solid.

The curved lines in the single vapor-phase region to the right of the saturation curve are lines of constant per cent saturation. Lines of constant dew-point are, of course, horizontal straight lines of constant humidity ratio. At point B, for example, the dry-bulb temperature is 60 F, the thermodynamic wet-bulb is 50 F, the dew-point is 40.8 F, the degree of saturation is 48.6 per cent, the humidity ratio is 0.00536 lb per pound dry air, and the enthalpy is $14.85 + (1000 \times 0.00536) = 20.21$ Btu per pound dry air.

With the aid of the Mollier diagram, it is easy to throw the definition of thermodynamic wet-bulb, Equation 20, into a more familiar form. Consider the three points 1, 2, 3, Fig. 2. Point 3 is located with respect to points 1 and 2 so that $W_3 = W_1$ and $t_3 = t_2$. Points 1 and 2, being on

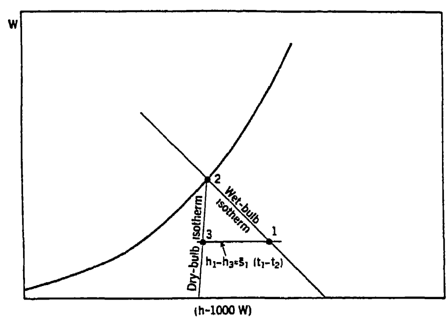

FIG. 2. DIAGRAM ILLUSTRATING THERMODYNAMIC WET-BULB TEMPERATURE

a line of constant thermodynamic wet-bulb, satisfy Equation 20; thus,

$$h_1 - h_3 + (W_2 - W_1)\, h'_{w,2} = h_2 - h_3$$

where h_3 has been subtracted from both sides. Under Dalton's Law, $h_2 - h_3 = (W_2 - W_1)\, h_{w,2}$; moreover, $h_1 - h_3$ may be replaced by $\bar{s}_1 (t_1 - t_2)$ where $\bar{s}_1$ is often referred to as *mean humid heat* and may be calculated with good approximation from

$$\bar{s}_1 = 0.240 + 0.444\, W_1 \tag{21}$$

Finally, introducing latent heat of vaporization at the wet-bulb temperature, namely, $(h_{fg})_2 = (h_w - h'_w)_2$, Equation 20 becomes, after omitting the subscript 2,

$$\frac{t_1 - t'}{W_s - W_1} = \frac{h_{fg}}{\bar{s}_1} \tag{22}$$

it being understood that W_s is the saturation humidity ratio and h_{fg}, the latent heat, at the wet-bulb temperature t'. Equation 22 was derived by Carrier [1].

Example 12. Work Example 10 using Equation 22.

Solution. A trial-by-error method is involved. Taking 48 F as a trial value of t', $(80 - 48) \div (0.007072 - 0) = 4520$; but $1066.7 \div 0.240 = 4440$. The trial value must, therefore, be revised upward, the final solution being 48.26 F as in Example 10.

TYPICAL AIR CONDITIONING PROCESSES

Illustrative Examples. The use of Table 6 and the Mollier diagram in analyzing typical air conditioning processes is best explained by the use of illustrative examples. In each of these examples, the observed pressure is assumed to be standard atmospheric pressure (29.921 in. Hg).

Example 13. Heating. Air at 20 F and 80 per cent saturation is to be heated to 120 F. Analyze the process as illustrated in Fig. 3.

Solution. The initial humidity ratio is 0.80 × 0.002144 = 0.001715 lb per pound dry air (table). This same value is read directly on the chart. The initial enthalpy is

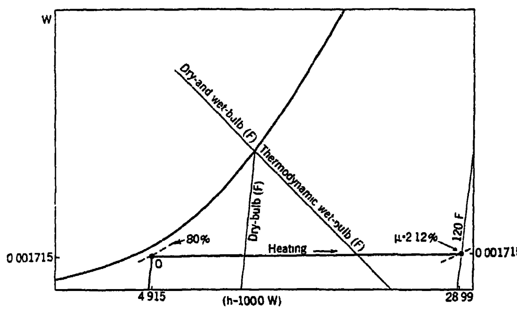

FIG. 3. DIAGRAM ILLUSTRATING EXAMPLE 13

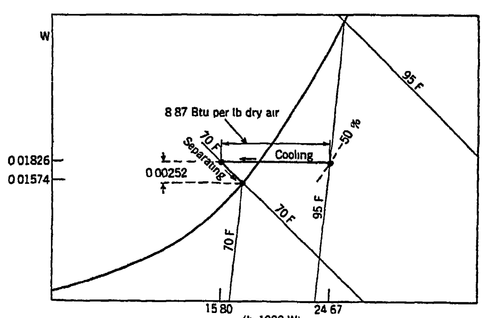

FIG. 4. DIAGRAM ILLUSTRATING EXAMPLE 14

4.798 + (0.80 × 2.290) = 6.630 Btu per pound dry air (table) or 4.915 + (1000 × 0.001715) = 6.630 (chart).

The final degree of saturation is 0.001715 ÷ 0.08093 = 0.0212 (table); hence the final enthalpy is 28.80 + (0.0212 × 90.09) = 30.71 Btu per pound dry air (table) or 28.99 + (1000 × 0.001715) = 30.71 (chart).

The increase in enthalpy is the quantity of heat to be supplied, namely, 30.71 − 6.63 = 24.08 Btu per pound dry air (table). Since humidity ratio W and therefore $1000W$ is constant, this is also simply the horizontal distance between the representative points on the chart; thus, the heat to be supplied is also 28.99 − 4.915 = 24.08 Btu per pound dry air (chart).

The final volume is 14.60 + (0.0212 × 1.90) = 14.64 cu ft per pound (table); or direct from the volume chart. Therefore, if 20,000 cfm of heated air is to be supplied, the quantity of heat required is (20,000 ÷ 14.64) × 24.08 = 32,900 Btu per minute.

Example 14. Cooling and Separating. Air at 95 F and 50 per cent saturation is to be cooled to 70 F and the liquid separated out. Analyze the process as shown in Fig. 4.

Solution. The initial humidity ratio is 0.50 × 0.03652 = 0.01826 (table). The initial enthalpy is 22.80 + (0.50 × 40.25) = 42.93 Btu per pound dry air (table) or 24.67 + (1000 × 0.01826) = 42.93 (chart).

The final state is in the two-phase region and consists of 0.01574 lb water per pound dry air in the vapor phase, and 0.00252 lb water per pound dry air in the liquid phase. The final enthalpy is therefor 33.96 + (0.00252 × 38.0) = 34.06 Btu per pound dry air (table) or 15.80 + (1000 × 0.01826) = 34.06 (chart).

The decrease of enthalpy is the refrigeration to be supplied and is 42.93 − 34.06 = 8.87 Btu per pound dry air (table). Since the weight of water per pound of dry air is constant, this is also the horizontal distance between the representative points on the chart, namely, 24.67 − 15.80 = 8.87 Btu per pound dry air (chart).

The initial volume is 13.97 + (0.50 × 0.82) = 14.38 cu ft per pound (table); or direct from the volume chart. Therefore, if 20,000 cfm of initial air is to be processed, the refrigeration required is (20,000 × 8 87) ÷ (14.38 × 200) = 61.7 tons. The weight of water to be removed is (20,000 × 0.00252) ÷ 14.38 = 3.51 lb per minute.

Example 15. Adiabatic Saturation with Recirculated Spray Water. Air at 75 F and 60 per cent saturation is saturated adiabatically with spray water which is recirculated.

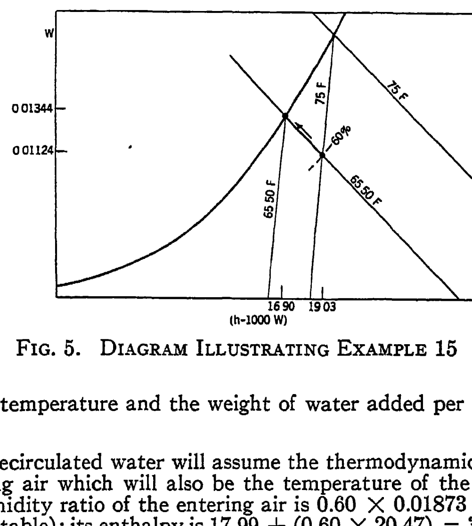

FIG. 5. DIAGRAM ILLUSTRATING EXAMPLE 15

Find the resulting temperature and the weight of water added per pound of dry air as outlined in Fig. 5.

Solution. The recirculated water will assume the thermodynamic wet-bulb temperature of the entering air which will also be the temperature of the resulting saturated mixture. The humidity ratio of the entering air is 0.60 × 0.01873 = 0.01124 lb water per pound dry air (table); its enthalpy is 17.99 + (0.60 × 20.47) = 30.27 Btu per pound dry air (table) or 19.03 + (1000 × 0.01124) = 30.27 Btu per pound dry air (chart). To determine the resulting temperature, the following equation must be solved.

$$30.27 + (W_s - 0.01124)\, h_w' = h_s$$

A trial value is 65 F corresponding to h_s = 30.27. The final value is 65.50 F corresponding to h_s = 30.27 + (0.01320 − 0.01124) × 33.05 = 30.34 Btu per pound dry air. The weight of water to be added is 0.01344 − 0.01124 = 0.00200 lb per pound dry air.

The volume of the entering air is 13.47 + (0.60 × 0.40) = 13.71 cu ft per pound. If 20,000 cfm of entering air is to be saturated, the weight of water to be added per minute is (20,000 × 0.00200) ÷ 13.71 = 2.92 lb per minute.

Adiabatic Mixing of Two Air Streams

A typical process requiring special discussion is the adiabatic mixing of two air streams. Let stream 1 contain M_1 pounds of dry air per minute and let its enthalpy be h_1 and its humidity ratio W_1. Using subscripts 2

and 3 in a similar manner to designate stream 2 and the resulting mixture respectively, write:

$$M_1 + M_2 = M_3 \quad \text{(weight balance for the dry air)}$$
$$M_1W_1 + M_2W_2 = M_3W_3 \quad \text{(weight balance for the water)}$$
$$M_1h_1 + M_2h_2 = M_3h_3 \quad \text{(energy balance, no heat absorbed)}$$

Eliminating M_3,

$$\frac{W_2 - W_3}{W_3 - W_1} = \frac{h_2 - h_3}{h_3 - h_1} = \frac{M_1}{M_2} \tag{23}$$

according to which: *on the Mollier Chart the representative point of the resulting mixture lies on the straight line connecting the representative points of the two streams being mixed, and divides the line into two segments which are in the same ratio as the weights of dry air in the two streams.* It must not be forgotten that this analysis assumes *adiabatic* mixing.

Example 16. Outside air at 0 F and 80 per cent saturation is to be mixed adiabatically with recirculated air at 70 F and 20 per cent saturation in the ratio, one pound of dry

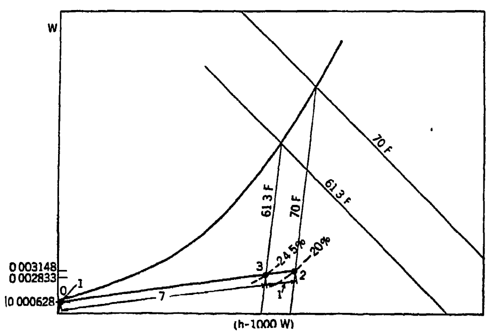

Fig. 6. Diagram Illustrating Example 16

air in the former to seven in the latter. Find the temperature and degree of saturation of the resulting mixture as shown in Fig. 6.

Solution. The humidity ratio and enthalpy of the resulting mixture satisfy

$$\frac{0.003148 - W_3}{W_3 - 0.000628} = \frac{20.23 - h_3}{h_3 - 0.666} = \frac{1}{7}$$

whence,

$W_3 = 0.002833$ lb water per pound dry air.

$h_3 = 17.78$ Btu per pound dry air.

The corresponding temperature and degree of saturation are 61.3 F and 24.5 per cent as is easily verified by use of Table 6. The numerical solution is somewhat tedious, but the graphical solution is easy.

Adiabatic Mixing with Injected Water

Another typical process is that of injecting water (solid, liquid or vapor) into an air stream to mix adiabatically with it. Let the subscripts 1 and 2 refer to the initial and final conditions, respectively; then write

$$1 + 0 = 1 \quad \text{(weight balance for the dry air)}$$
$$W_1 + (W_2 - W_1) = W_2 \quad \text{(weight balance for the water)}$$
$$h_1 + (W_2 - W_1)\,h_W = h_2 \quad \text{(energy balance, no heat absorbed)}$$

The first two are identities and are incorporated in the third. This may be rewritten as follows:

$$\frac{h_2 - h_1}{W_2 - W_1} = h_W \tag{24}$$

and shows that *the process is represented by a straight line on the Mollier diagram, the slope of the line being determined by the specific enthalpy of the injected water.* It must not be forgotten that the analysis assumes adiabatic mixing. *Energy convected with a fluid is not heat.*

Example 17. It is desired to increase the humidity ratio of air at 70 F without changing its temperature. Under what conditions may water be injected in order to accomplish the desired result.

Solution. Under Dalton's Law a line of constant (dry-bulb) temperature is straight on the Mollier diagram and its slope is determined by the specific enthalpy of water *vapor* at the given temperature. At 70 F, h_W = 1092.3 Btu per pound; hence injection of steam having this specific enthalpy will cause the representative point to move in a direction parallel to the 70 F isotherm. Saturated steam at 70 F may not be used because its pressure is only 0.7392 in. Hg and it cannot therefore be injected into air at atmos-

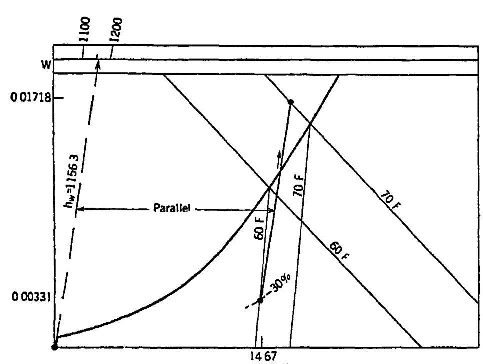

Fig. 7. Diagram Illustrating Example 18

pheric pressure. Saturated steam at 667.4 F, 2488 lb per square inch has the right specific enthalpy and can be throttled into a room at 70 F without altering the room temperature.

Border Scale

On the Mollier diagram is placed a border scale to facilitate the graphical solution of problems in which given quantities of energy and water are added (or withdrawn) simultaneously as in the case of adiabatic mixing with injected water. All marks in the upper half of this scale point to the lower left corner of the chart and each shows the *direction* that the representative point will move due to adiabatic mixing with injected water having the indicated specific enthalpy. All marks in the lower half of the scale point to the lower right corner of the chart.

Example 18. If dry saturated steam at 20 lb per square inch absolute is injected into air initially at 60 F and 30 per cent saturation to raise the temperature to 70 F, what is the final degree of saturation and how much water is added per pound dry air? (See Fig. 7.)

Solution. The initial humidity ratio is 0.30 × 0.01103 = 0.00331 lb water per pound dry air (or direct from chart). The initial enthalpy is 14.39 + (0.30 × 11.98) = 17.98 Btu per pound dry air (table) or 14.67 + (1000 × 0.00331) = 17.98 (chart). A pre-

liminary calculation shows that the final mixture contains liquid. The final weight of water per pound of dry air is determined from

$$\frac{33.96 + (W - 0.01574) \times 38.0 - 17.98}{W - 0.00331} = 1156.3$$

where the specific enthalpy of the injected water is 1156.3 Btu per pound. The answer is

$$W = 0.01718 \text{ lb water per pound dry air.}$$

Therefore, the weight of water added is $0.01718 - 0.00331 = 0.01387$ lb per pound dry air as shown in Fig. 7.

Cooling Load

In the calculation of the cooling load for an air conditioned space, the problem usually reduces to determining the quantity of inside air that must be withdrawn and the condition to which it must be brought by cooling, separating and possibly reheating so that return of the conditioned air will have the net effect of removing given amounts of energy and water from the air conditioned space.

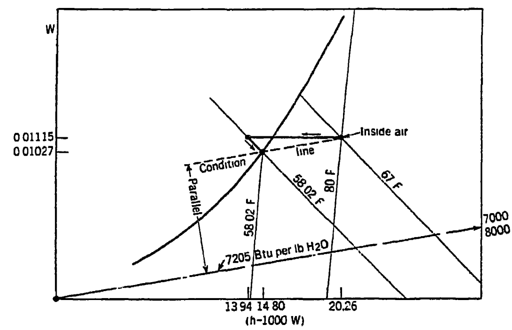

FIG. 8. DIAGRAM ILLUSTRATING EXAMPLE 19

Let m denote the weight of dry air withdrawn per hour. With it will be withdrawn *energy* of amount mh_1 Btu per hour and *water* of amount mW_1 pounds per hour, where h_1 and W_1 denote enthalpy and humidity ratio, respectively, of inside air. The weight of dry air returned per hour will be the same as that withdrawn but with it must be returned a smaller amount of *energy*, mh Btu per hour, and a smaller quantity of *water*, mW pounds per hour, where h and W denote enthalpy and humidity ratio of conditioned air.

With this understanding, the requirements of the cooling load problem are,

$$mh = mh_1 - \Delta Q$$
$$mW = mW - \Delta W$$

where ΔQ and ΔW are the *given* amounts of energy and water, respectively, to be removed. Eliminating m from these two equations,

$$\frac{h - h_1}{W - W_1} = \frac{\Delta Q}{\Delta W}$$

which says that all possible states for the conditioned air lie on a straight line, on the Mollier Chart, which passes through the state point of the in-

side air with a slope determined by the ratio of the quantity of energy to be removed to the quantity of water to be removed. This straight line is called the *condition line* for the given problem. The border scale facilitates the graphical solution of this problem.

In practice the point at which the condition line crosses the saturation curve may dictate an excessive number of air changes for the particular space to be conditioned. If so it might be necessary to cool to a lower temperature; but if the requirements of the problem are to be exactly met both as regards the removal of water and the removal of energy, the mixture returned to the conditioned space must contain a certain amount of liquid. In other words, its state point must lie on the condition line; otherwise excessive dehumidification will result.

Example 19. In order to maintain a condition of 80 F dry-bulb, 67 F wet-bulb in a certain store, it is found necessary to remove 115,060 Btu of energy per hour and 15.97 lb of water per hour. Analyze the problem illustrated in Fig. 8.

Solution. The state point of the inside air is easily located on the Mollier Chart. Through it draw a line having the slope 115,060 ÷ 15.97 = 7205 Btu per pound *water* as determined from the border scale. This line crosses the saturation curve at 58.02 F. Hence a possible conditioning process is to cool some of the inside air to 58.02 F, separate the liquid thus formed, and return the resulting saturated mixture to the store.

The thermodynamic properties entering the calculation are:

	Inside Air	*After Cooling*	*After Separating*
t	80.0 F	58.02 F	58.02 F
W	0.01115	0.01115	0.01027
h	31.41	25.09	25.07

The weight of dry air to be withdrawn is 115,060 ÷ (31.41 − 25.07) = 18,130 lb per hour.

Adiabatic Saturation

Any case of adiabatic mixing in which the resulting mixture is saturated may properly be called *adiabatic saturation.* For example, if enough water at 352 F be sprayed into dry air at 80 F to produce a saturated mixture, the resulting enthalpy will be $h_s = 19.19 + (W_s - 0)\ 324$; and since h_s and W_s are functions of the same temperature, this temperature is determined by the equation to be 53.0 F. Thus, adiabatic saturation of dry air at 80 F by injecting liquid water at 352 F results in a temperature of 53.0 F when saturation is reached.

But in practice, much more is usually read into the term adiabatic saturation, it being generally understood that saturation is to be produced by injecting liquid water at such a temperature as will *coincide* with that at which the saturation curve is reached. *With this understanding it may be said that thermodynamic wet-bulb temperature is the result of adiabatic saturation.* Thus, if liquid water at 48.26 F instead of 352 F be injected into dry air at 80 F a saturated mixture at 48.26 F instead of 53.0 F will be produced. Therefore, 48.26 F is the thermodynamic wet-bulb temperature of dry air at 80 F.

It is possible to produce adiabatic saturation, interpreting the term literally, by mixing two air streams neither of which is itself saturated. In order for this to be possible, the straight line connecting the representative points on the Mollier diagram must cut the saturation curve twice.

STEADY FLOW ENERGY EQUATION

It was previously stated that, in steady flow, the energy convected by the fluid at any section is the sum of (*a*) kinetic energy due to velocity; (*b*) gravitational energy due to elevation; (*c*) enthalpy due to the condition of pressure, temperature and composition of the fluid. A more detailed discussion of item (*a*) is in order.

Kinetic Energy

There are reasons to believe that the so-called *velocity pressure* h_V read by a Pitot tube is simply the kinetic energy per unit volume of the fluid immediately upstream from the tube, as application of Bernoulli's Equation suggests. Thus (see Equation 3, Chapter 34).

$$V = 1097.3 \sqrt{\frac{h_V}{d}} \tag{25}$$

where

V = velocity, feet per minute.
h_V = velocity pressure, inches of water at 60 F.
d = density of fluid, pounds per cubic foot.

In the case of flow through a duct, the velocity pressure is found to vary considerably over the section and a traverse has to be made. The cross-sectional area of the duct is divided into a number of equal concentric areas, and measuring stations are located at centroidal points in each area along two perpendicular diameters. Usually the ultimate object is to determine an *average* velocity $\overline{V}$ from which the weight of fluid crossing the section per unit time can be obtained on multiplying by the cross-sectional area of the duct and by the density of the fluid. This is obtained by simply averaging the square roots of all measured velocity pressures as follows:

$$\overline{V} = \frac{1097.3}{\sqrt{d}} \left(h_V^{1/2} \right)_{av} \tag{26}$$

where

$\overline{V}$ = average velocity, feet per minute.
$(h_V^{1/2})_{av}$ = arithmetic average of the square roots of all measured velocity pressures, inches of water at 60 F.

But the item of present importance is the *average kinetic energy* convected with each pound of fluid. Consistently with the previous discussion, this can be shown to be

$$\overline{KE} = 0.006678\, v \frac{\left(h_V^{3/2} \right)_{av}}{\left(h_V^{1/2} \right)_{av}} \tag{27}$$

where

$\overline{KE}$ = average kinetic energy, Btu per pound.
v = specific volume, cubic feet per pound.
$(h_V^{3/2})_{av}$ = arithmetic average of the 3/2-powers of all measured velocity pressures, inches of water at 60 F.

If the velocity pressure were uniform over the section, Equations 26 and 27 could be combined to give

$$\overline{KE} = \left(\frac{\overline{V}}{13{,}430}\right)^2 \tag{28}$$

But, it is interesting to note that if the velocity varies *parabolically* from zero at the walls to maximum at the center as it does in the case of purely viscous flow in a circular duct, *then the average kinetic energy is twice that given by* Equation 28.

Example 20. If 2000 cfm of air flows through an 8 in. diameter circular duct, find the average kinetic energy per pound of air.

Solution. The cross-sectional area of the duct is 0.349 sq ft; hence the average flow velocity is 5730 fpm. If the velocity were uniform over the section, the average kinetic energy would be $(5730 \div 13{,}430)^2 = 0.182$ Btu per pound. But it is more likely that the actual distribution of velocity would approximate that characteristic of viscous flow; hence the average kinetic energy would be more nearly $2 \times 0.182 = 0.364$ Btu per pound.

Gravitational Energy

The potential energy due to elevation Z (feet) above any convenient datum is simply $Z \div 778.3$ Btu per pound of fluid. In the case of moist air,

$$\overline{PE} = \frac{\overline{Z}\,(1 + W)}{778.3} \tag{29}$$

where

$\overline{PE}$ = average potential energy, Btu per pound dry air.
$\overline{Z}$ = average elevation, feet.
W = humidity ratio, pound water per pound dry air.

Enthalpy

No further discussion of enthalpy is required. It may be well to emphasize, however, that enthalpies have been figured on the basis of one pound of dry air.

Heat and Shaft Work

Between any two sections 1 and 2 in an apparatus through which steady flow occurs, there may be heat *absorbed* from outside, ${}_1q_2$, Btu per pound of dry air, and shaft work *removed* to outside, ${}_1l_2$, Btu per pound of dry air. If heat is actually rejected to outside, ${}_1q_2$ is intrinsically negative; and if shaft work is actually put in from outside ${}_1l_2$, is intrinsically negative.

Steady-flow Energy Equation

A complete energy accounting takes the form of Equation 30 which is usually referred to as the steady-flow energy equation.

$${}_1q_2 = (h_2 + \overline{KE}_2 + \overline{PE}_2) - (h_1 + \overline{KE}_1 + \overline{PE}_1) + {}_1l_2 \tag{30}$$

where

${}_1q_2$ = heat *added* from outside *between* sections 1 and 2, Btu per pound dry air.
h_2 = enthalpy of the mixture *at* section 2, Btu per pound dry air.
$\overline{KE}_2$ = average kinetic energy *at* section 2, Btu per pound dry air.
$\overline{PE}_2$ = average potential energy *at* section 2, Btu per pound dry air.

h_1 = enthalpy *at* section 1, Btu per pound dry air.
$\overline{KE}_1$ = average kinetic energy *at* section 1, Btu per pound dry air.
$\overline{PE}_1$ = average potential energy *at* section 1, Btu per pound dry air.
${}_1l_2$ = shaft work withdrawn *between* sections 1 and 2, Btu per pound dry air.

In Equation 30 all quantities are *per pound of dry air.* If Equation 27 is used in computing average kinetic energy, the result will be in Btu per pound of dry air if v is taken as volume per pound of dry air. If Equation 28 is used, multiplication by $(1 + W)$ as in Equation 29 is required though this is a refinement seldom justified.

Properties of saturated steam are given in Table 8 and for additional definitions refer to Chapter 46.

REFERENCES

[1] Rational Psychrometric Formulae, by W. H. Carrier (*A.S.M.E. Transactions*, Vol. 33, 1911, p. 1005).

[2] Heat and Thermodynamics, by J. K. Roberts (Chapter IV), (Blackie and Son, London, 1928).

[3] Das i, x-Diagramm für Dampfluftgemische, by R. Mollier (ZVDI, 1929).

[4] The Theory of the Psychrometer, by J. H. Arnold (*Physics*, Vol. 4, 1933).

[5] Thermodynamic Properties of Moist Air, by John A. Goff (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 199).

[6] Thermodynamic Properties of Steam, by J. H. Keenan and F. G. Keyes (John Wiley and Sons, Inc., New York, 1936).

[7] Notes on Thermodynamics, by John A. Goff (John S. Swift Company, St. Louis, 1939).

[8] Thermal Properties of Saturated Water and Steam, by N. S. Osborne, H. F. Stimson and D. C. Ginnings (*National Bureau of Standards*, Research Paper No. RP 1229, 1939).

[9] Psychrometric Chart, Its Application and Theory, by William Goodman (*Heating, Piping and Air Conditioning*, June, 1939 to August, 1940).

[10] The Interaction Constant for Moist Air, by John A. Goff and A. C. Bates (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, July 1941, p. 442).

Chapter 2

PHYSIOLOGICAL PRINCIPLES

Chemical Vitiation of Air, Physical Impurities in Air, Thermal Exchanges Between the Body and Its Environment, Adaptation to Hot Conditions, Adaptation to Cold Conditions, Relation of Air Conditioning Needs to Metabolism, Acclimatization, Effective Temperature Index, Physiological Objectives of Heating and Ventilation, Relation of Air and Wall Temperatures, Influence of Humidity, Influence of Air Movement, The Four Vital Factors

VENTILATION is defined in part as *the process of supplying or removing air by natural or mechanical means to or from any space.* (See Chapter 47). The word in itself implies quantity but not necessarily quality. From the standpoint of comfort and health, however, the problem is now considered to be one of securing air of the proper quality rather than of supplying only a given quantity.

The term *air conditioning* in its broadest sense implies control of any or all of the physical or chemical qualities of the air. When applied to comfort air conditioning, however, the A.S.H.V.E. Code of Minimum Requirements of Comfort Air Conditioning defines it "as the process by which simultaneously the temperature, moisture content, movement and quality of the air in enclosed spaces intended for human occupancy may be maintained within required limits. If an installation cannot perform all of these functions, it shall be designated by a name that describes only the function or functions performed."

CHEMICAL VITIATION OF AIR

Under the artificial conditions of indoor life, the air undergoes certain physical and chemical changes which are brought about by the occupants themselves. The oxygen content is somewhat reduced, and the carbon dioxide slightly increased by the respiratory processes. Organic matter, which is usually perceived as odors, comes from the nose, mouth, skin and clothing. The temperature of the air is increased by the metabolic processes, and the humidity raised by the moisture emitted from the skin and lungs.

Contrary to old theories, the usual changes in oxygen and carbon dioxide are of no physiological concern because they are too small to produce appreciable effects even under the worst conditions of normal human occupancy. Only in such unusually air-tight enclosures as submarines need the increase in carbon dioxide and the reduction in oxygen be considered. The amount of carbon dioxide in air is often used as an index of odors of human origin, but the information it affords rarely

justifies the labor involved in making the observation[1,2]. Little is known of the identity and physiological effects of the organic matter given off in the process of respiration. The former belief that the discomfort experienced in confined spaces was due to some toxic volatile matter in the expired air is now limited, in the light of numerous researches, to the much less dogmatic view that the presence of such a substance has not been demonstrated. The only certain fact is that expired air may be odorous and offensive, and it is capable of producing loss of appetite and a disinclination for physical activity. These reasons, whether esthetic or physiological, call for the introduction of a certain minimum amount of clean outdoor air to dilute odors from any source, including cooking and other processes to a concentration which is not objectionable.

In certain industrial processes toxic fumes and gases may be produced, whose removal by local exhaust ventilation is essential for the protection of human health. In the ordinary occupied spaces harmful chemical impurities may be contributed from certain types of cooking and heating appliances including carbon monoxide from imperfect combustion which may be a serious hazard to life and health.

The control of offensive or hazardous concentrations of chemical vitiation in the air is frequently brought about by the ventilating engineer through dilution. This method has found application when the source of contamination is the human occupant and not something of a particularly hazardous character.

In the case of vitiation by a few hazardous gases such as carbon monoxide from heating and cooking and certain industrial processes, no satisfactory chemical treatment for the elimination of the impurity has been found. The only really satisfactory solution is elimination at the source; or if this is impossible reduction to a safe concentration by dilution. In the case of contamination by other forms of material, including volatile vapors and gases, chemical treatment for the removal or chemical reduction of the impurities have been made available through air cleaning methods, which will be discussed in Chapter 29 on Air Cleaning Devices. The A.S.H.V.E. Research Technical Advisory Committee on Air Pollution and Air Purification has outlined[3] means for the reduction of atmospheric impurities.

When the only source of contamination is the human occupant, the minimum quantity of outdoor air needed, appears to be that necessary to remove objectionable body odors, or tobacco smoke. The concentration of body odor in a room, in turn, depends upon a number of factors, including the dietary and hygienic habits of the occupants (frequently reflecting their socio-economic status), the outdoor air supply, air space allowed per person, odor adsorbing capacity of air conditioning processes, and temperature and relative humidity. Perception of odor, like the perception of most of our other senses, is proportional to the logarithmic function of the intensity of the stimulus; or in the case of odors, the

[1]A.S.H.V.E. RESEARCH REPORT NO. 959—Indices of Air Changes and Air Distribution, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 261).

[2]A.S.H.V.E. RESEARCH REPORT NO. 1031—Ventilation Requirements, by C. P. Yaglou, E. C. Riley and D. J. Coggins (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p 133).

[3]Report of the A.S.H.V.E. Technical Advisory Committee on Air Pollution and Air Purification (Programs of the Research Technical Advisory Committee, June, 1941).

perception has been found to vary as the logarithmic function of the odor intensity, or inversely with the logarithmic function of the available air determined by the outdoor air supply and the air space per person.

The relation between air supply and occupancy has been reported by the *Harvard School of Public Health*[4] and the A.S.H.V.E. Research Laboratory[5]. The findings from the Harvard study are given in Table 1. Outdoor air requirements for removal of objectionable tobacco smoke odors are not accurately known but sources of information available and practice indicates the need of 15 cfm per person or more.

The total quantity of outside air to be circulated through an enclosure is governed by both chemical and physical considerations. The physical requirements for controlling temperature, air distribution and air velocity usually predominate. Other factors which must be taken into consideration include the type and usage of the building, locality, climate, height of rooms, floor area, window area, extent of occupancy, and the operation of the system distributing the air supply. Frequently, some of these factors, particularly the need for air movement and good distribution, may be satisfied by recirculation of inside air rather than by outside air.

It will be noted that, with adequate air space, the rate of air change indicated in Table 1 is from 10 to 30 cfm per person. In rooms occupied by only a few persons such a rate of air change will be automatically attained in cold weather by normal leakage around doors and windows while it can easily be secured in warm weather by the opening of windows. With a space allotment of 400 cu ft per person, only 1½ air changes per hour are necessary to provide an air change of 10 cfm per person. This space allotment is essential for other reasons.

Therefore, in the ordinary dwelling with adequate cubic space allotment, no special provision for controlling chemical purity of the air is necessary (aside from removal of fumes from heating appliances). For such conditions, the control of air temperature is the major factor to be considered.

In more crowded rooms (large offices, large workrooms, auditoriums), the whole picture changes. Cubic space per person is less and the size of the room makes it impossible to admit untempered outside air without drafts. Here, mechanical ventilation is essential, but as will be noted in a later paragraph, it is even more essential for thermal than for chemical reasons. It is control of the thermal properties of the air in order to effect the removal of the heat produced by human bodies, rather than dilution of chemical poisons, which must govern practice.

The Code of Minimum Requirements for Comfort Air Conditioning[6] prescribes definite minimum requirements which should be familiar to the designing engineer. It should be emphasized, however, that the provisions of the code aim to provide *minimum*, rather than *adequate*, requirements.

Notwithstanding the rapid advance in the field of air conditioning

[4]Loc. Cit. Note 2.

[5]Loc. Cit. Note 1.

[6]Code of Minimum Requirements for Comfort Air Conditioning (A.S.H.V.E. TRANSACTIONS Vol. 44. 1938, p. 27). Reprints of this code are available at $.10 a copy.

during the past few years, there still remain those who believe in a superior, stimulating quality of outdoor air (particularly country, mountain and seashore air) under ideal weather conditions, as compared with properly conditioned air. While this point of view is usually held by persons not intimately acquainted with the complicated factors involved they nevertheless carry some weight. It is apparent, however, to anyone acquainted with the factors involved and the conditions effecting comfort that in modern air conditioning, like in most other branches of engineering, modern science makes it possible to control the phenomena of nature for

Table 1. Minimum Outdoor Air Requirements to Remove Objectionable Body Odors

(Provisional values subject to revision upon completion of work)

Type of Occupants	Air Space per Person Cu Ft	Outdoor Air Supply cfm per Person
Heating season with or without recirculation. Air not conditioned.		
Sedentary adults of average socio-economic status	100	25
Sedentary adults of average socio-economic status	200	16
Sedentary adults of average socio-economic status	300	12
Sedentary adults of average socio-economic status	500	7
Laborers	200	23
Grade school children of average class	100	29
Grade school children of average class	200	21
Grade school children of average class	300	17
Grade school children of average class	500	11
Grade school children of poor class	200	38
Grade school children of better class	200	18
Grade school children of best class	100	22
Heating season. Air humidified by means of centrifugal humidifier. Water atomization rate 8 to 10 gph. Total air circulation 30 cfm per person.		
Sedentary Adults	200	12
Summer season. Air cooled and dehumidified by means of a spray dehumidifier. Spray water changed daily. Total air circulation 30 cfm per person.		
Sedentary Adults	200	<4

the service and comfort of man beyond any possibilities found in nature itself. When the requirements for optimum comfort as determined by the atmospheric environment are known (and our comprehensive studies to date lead us to believe that they are known at least to a high degree), the air conditioning engineer can supply these requirements indoors to the same perfection as may accidentally be found at times outdoors and keep them under control. The freedom of movement, action and thought, together with the variability of stimulae experienced by persons under ideal conditions in the country, mountains or seashore, and the psychological effect of these *wide open spaces* undoubtedly have some stimulating

effect, which when compared with the monotony of confinement indoors even in the most favorable atmospheric environment accounts for the contrast. Ultra-violet light and ionization have been suggested but the evidence so far is inconclusive or negative[7].

Ozone has been used with success for the destruction of micro-organisms (molds) in meat packing establishments and the like; and where considerable amounts of organic effluvia are present it may be useful as a deodorant. For ordinary ventilation practice, however, neither of these purposes can be usefully attained, since the concentration of ozone necessary for effectiveness would be likely to transcend the limit of comfort in ordinary occupied rooms. While ozone has been used in the treatment of certain diseases, there is no evidence that it has a tendency to increase comfort or to benefit health under conditions of normal human occupancy. The allowable concentrations in the breathing zone are very small, between 0.01 to 0.05 ppm parts of air. These are much too small to influence bacteria. Higher concentrations are associated with a pungent unpleasant odor and considerable discomfort to the occupants. One part per million causes respiratory discomfort, headaches, depression, a lowering of the metabolic rate and may even lead to coma[8].

PHYSICAL IMPURITIES IN AIR

Dust particles of various types, when present in considerable concentrations, produce an irritant effect upon the mucous membranes of nose and throat and may be associated with high prevalence of acute respiratory diseases such as bronchitis and pneumonia. Dust which contains free silica has special harmful effects, causing a primary disease of the lungs (silicosis) and predisposing the victim in a high degree to tuberculosis. These, however, are special problems of industrial hygiene which will not be discussed in detail in this chapter.

A certain part of the dissemination of disease in confined spaces is caused by the emission of pathogenic organisms from infected persons. Droplets sprayed into the air in talking, coughing, sneezing, etc., do not all fall immediately to the ground within a few feet from the source, as was formerly believed. The large droplets do, of course, but minute droplets less than 0.1 mm in diameter evaporate to dryness before they fall the height of a man. Nuclear residues from such sources, which may contain infective organisms drift long distances with the air currents and the virus may remain alive long enough to be transmitted to other persons in the same room or building. Droplet nuclei have been recovered from cultures of resistant micro-organisms a week after innoculation into a tight chamber of 3000 cu ft capacity, although the majority of disease

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[7]A.S.H.V.E. RESEARCH REPORT No. 921—Changes in Ionic Content in Occupied Rooms, Ventilated by Natural and Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 191). A.S.H.V.E. RESEARCH REPORT No. 965—Physiologic Changes During Exposure to Ionized Air, by C. P. Yaglou, A. D. Brandt and L. C. Benjamin (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 357). A.S.H.V.E. RESEARCH REPORT No. 985—Diurnal and Seasonal Variations in the Small Ion Content of Outdoor and Indoor Air, by C. P. Yaglou and L. C. Benjamin (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 271). The Nature of Ions in Air and Their Possible Physiological Effects, by L. B. Loeb (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 101). The Influence of Ionized Air upon Normal Subjects, by L. P. Herrington (*Journal Clinical Investigation*, 14, January, 1935). The Effect of High Concentrations of Light Negative Atmospheric Ions on the Growth and Activity of the Albino Rat, by L. P. Herrington and Karl L. Smith (*Journal Ind. Hygiene*, 17, November, 1935). Subjective Reactions of Human Beings to Certain Outdoor Atmospheric Conditions, by C.-E. A. Winslow and L. P. Herrington (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 119).

[8]*The British Medical Journal*, Editorial, June 25, 1932, p. 1182. See also Loc. Cit. Note 4.

germs died out within a few hours[9]. Practical epidemiological evidence indicates that the danger of such atmospheric transmission is slight with the bacterial diseases but may be appreciable with the diseases caused by the much smaller viruses. Avoidance of overcrowding is a major factor in avoiding such dangers. The microbic concentration in the atmosphere may be reduced by air change, but since the rate of contamination may be great at local points over short periods of time the hazardous concentration may not be eliminated quickly enough and may even be spread over larger areas by local drafts. The possibility of sterilizing the air supply at the source, or destroying the micro-organisms at their point of admission to the air by ultra-violet light is being studied and offers considerable promise[10].

While in some instances it may be possible to reduce the physical impurities of the air by dilution from a non-contaminated source, such non-contaminated sources are rarely available. Frequently the outside air contains a higher concentration of physical impurities than that within an enclosure. Therefore, it is usually desirable to reduce the concentration of physical impurities by air cleaning methods, as discussed in Chapter 29.

THERMAL INTERCHANGES BETWEEN THE BODY AND ITS ENVIRONMENT

The importance of the thermal factors arises from the profound influence which they exert upon body temperature, comfort and health. Body temperature depends upon the balance between heat production and heat loss. The heat resulting from the combustion of food within the body (metabolism) maintains the body temperature well above that of the surrounding air. At the same time, heat is constantly lost from the body by radiation, convection and evaporation. Since, under ordinary conditions, the body temperature is maintained at its normal level of about 98.6 F, the heat production must be balanced by the heat loss.

In conditioning air for comfort and health it is necessary to know the rate of sensible and latent heat liberation from the human body, which in conjunction with other heat loads (see Chapters 4, 6 and 7) determine the capacity required for proper conditioning. The data in common use are those of the A.S.H.V.E. Research Laboratory[11].

The fundamental thermodynamic processes concerned in heat interchanges between the body and its environment may be described by the equation:

$$M = \pm S + E \pm R \pm C \tag{1}$$

[9]Air-Borne Infection and Sanitary Air Control, by W. F. Wells (*Journal Industrial Hygiene*, November, 1935).

[10]Sanitary Ventilation in Wards, by W. F. Wells (*Heating and Ventilating*, April, 1939, p. 26). Measurement of Sanitary Ventilation, by W. F. Wells (*American Journal of Public Health*, Vol. 28, 1938, p. 343).

[11]A.S.H.V.E. Research Report No. 830—Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (A.S.H.V.E. Transactions, Vol. 35, 1929, p. 245). Thermal Exchanges Between the Human Body and Its Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (*American Journal of Physiology*, Vol. 88, 1929, p. 386). A.S.H.V.E. Research Report No. 908—Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (A.S.H.V.E. Transactions, Vol. 37, 1931, p. 541). Thermal Exchanges Between the Bodies of Men Working and the Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (*American Journal of Hygiene*, Vol. XIII, 1931, No. 2, p. 415). A.S.H.V.E. Research Report No. 1106—Air Conditioning in Industry, by W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten and M. B. Ferderber (A.S.H.V.E. Transactions, Vol. 45, 1939, p. 59).

where

M = rate of metabolism.
S = rate of storage.
E = rate of evaporative heat loss.
R = rate of radiative heat loss or gain.
C = rate of convective heat loss or gain.

Factor M, the rate of metabolism, is always positive. The storage, S, may be either positive or negative, depending upon whether heat is being stored or given off, accompanied by a rise or fall in body temperature. Under ordinary circumstances (when the dew-point of the air is below the body surface temperature) the evaporation loss, E, is always positive; that is, heat from metabolism supplies this loss. R and C are positive when the surface temperature of the body is above that of the walls and air, and negative when it is cooler.

The human body possesses remarkable powers of adaptation to a narrow range of atmospheric conditions around an ideal optimum where storage is zero, and metabolism and skin and tissue temperature are at optimum values. As skin temperature and body-tissue temperature rise or fall above or below an optimum, complex adaptive mechanisms come into play, chiefly associated with redistribution of blood supply between the skin and deeper tissues (in a cold environment) and with sweat secretion (in a hot environment). Under cold conditions, the need for more heat and shivering or other muscular movements increase metabolism, which is, again, a reaction favorable to temperature regulation; but under very hot conditions metabolism also rises, and this reaction is obviously harmful and indicates a balance of purely chemical over phsysiological control[12] resulting from increased chemical reactions with rise in temperature. In other words, it represents a breakdown or failure of the entire regulative processes. These reactions are governed by nervous or chemical stimuli from both skin and internal tissues. Nerves from the skin, for example, carry the sense impressions to the brain and the response comes back over another set of nerves, the motor nerves, to the musculature and to all the active tissues in the body, including the endocrine glands. In this way, a two-sided mechanism controls the body temperature by (1) regulation of internal heat production (chemical regulation), and (2) regulation of heat loss by means of automatic variation in the rate of cutaneous circulation and the operation of the sweat glands (physical regulation). The reactions involved in cold and in hot environments are on the whole radically different in nature. The mechanisms of adjustment involved are extremely complex and while they are receiving considerable study a complete understanding of their operation is still lacking.

In a certain middle range, normal and easy physiological regulation occurs by slight changes in the distribution of blood carrying heat between the skin and the inner organs, resulting in slight changes in the body surface temperature, and hence, in the rate of heat dissipation to the atmosphere. This easy balance gives a sensation of comfort. Above this

[12]Loc. Cit. Note 11.

range, the blood capillaries near the surface become dilated, allowing more blood and heat to flow into the skin, and thus increase its temperature and consequently its heat loss. If this method of cooling is not in itself sufficient, the stimulus is extended to the sweat glands which allow water to pass through the surface of the skin. This method of cooling is the most effective of all, as long as the vapor pressure and dew-point temperature of the air are sufficiently low to allow for evaporation. In high humidities, where the difference between the dew-point temperature of the air and body temperature is not sufficient to allow rapid evaporation, increase in heat loss may be had by increasing air movement. The body, under hot conditions, is in the *zone of evaporative regulation*, and for moderately extreme conditions perfect balance between heat production and heat loss may be attained, although at the cost of considerable discomfort.

In a cold environment, where environmental conditions are such as to remove heat too rapidly, the organism adapts in some degree by constricting the blood vessels leading to the surface, thereby reducing the blood flow and heat available for dissipation to the environmental surroundings. This adaptation is, however, partial and incomplete, and in an environment too cold for the clothing worn the temperature of the body tissues may fall, with accompanying discomfort and ultimate danger of serious chill. The process may go on for hours. The individual may move about and increase metabolism through muscular activity and thus balance the excessive heat demand of the environment, or he may reduce the loss by greater insulation of his body in the form of clothing.

Some of these phenomena which are important are shown graphically in Fig. 1. The dotted curves, from a study at the *John B. Pierce Laboratory of Hygiene*[13], are for subjects lightly clothed in a semi-reclining position and give the relation between the dry-bulb temperature of the environment (with about 45 per cent relative humidity) and the metabolic rate, the rate of heat dissipation by radiation and convection combined, and the latent heat loss due to evaporation of, perspiration and moisture from the respiratory tract. The smooth line curves, from the work of the A.S.H.V.E. Research Laboratory[14], give the same relationships for healthy, male subjects (18 to 24 years of age), seated at rest and normally clothed for winter-heated and air conditioned occupancy. The data for the semi-reclining subject also include the rate of heat storage (either positive or negative) due to a rise or fall in body temperature. For the normally clothed subjects a curve gives the total heat loss (that is, the sum of the radiation, convection and evaporative losses). Here, storage is given by the difference between the metabolism and total heat loss.

The small difference between the metabolic or heat production rates for the two types of subjects may be accounted for by the difference in activity. Heat exchange between the body and the environment by radiation and convection is greater for the lightly clothed subject, both for cool conditions where there is considerable heat loss, and for very warm conditions where there is a sensible transfer from the atmosphere to the body. The two curves for evaporative loss serve to show how physiological control

[13]A.S.H.V.E. Research Report No. 1107—Recent Advances in Physiological Knowledge and Their Bearing on Ventilation Practice, by C.-E. A. Winslow, T. Bedford, E. F. DuBois, R. W. Keeton, A. Missenard, R. R. Sayers and C. Tasker. (A.S.H.V.E. Transactions, Vol. 45, 1939, p. 111).

[14]Loc. Cit. Note 11.

uses evaporation of perspiration to maintain equilibrium, particularly at high temperatures. Below about 75 F for the normally clothed subject, and below about 85 F for the lightly clothed subject, evaporation loss is minimal and probably due to uncontrolled evaporation from the relatively dry skin and from the respiratory tract. Above these temperatures control is had by availability of perspiration for evaporation. The difference in the curves above 75 F is probably largely determined by the difference in clothing and activity. Above temperatures from 95 to 100 F (probably that of the average outside surface of the clothed body) radiation and convection combined changes from positive to negative, and slightly above this temperature even the greatly increased latent heat loss ceases to suffice to take care of the rate of heat production and the negative

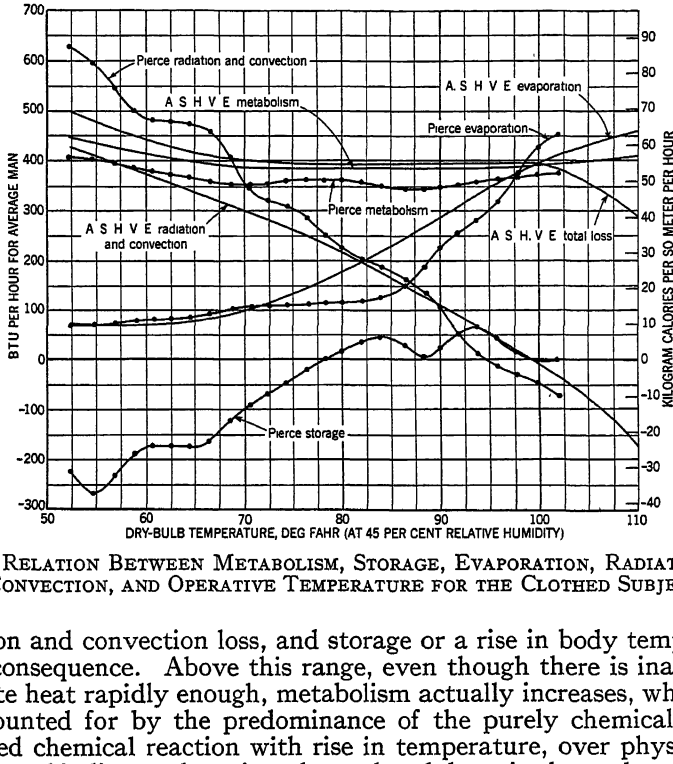

FIG. 1. RELATION BETWEEN METABOLISM, STORAGE, EVAPORATION, RADIATION PLUS CONVECTION, AND OPERATIVE TEMPERATURE FOR THE CLOTHED SUBJECT

radiation and convection loss, and storage or a rise in body temperature is the consequence. Above this range, even though there is inability to dissipate heat rapidly enough, metabolism actually increases, which may be accounted for by the predominance of the purely chemical laws of increased chemical reaction with rise in temperature, over physiological control, and indicates the point where a breakdown in thermal equilibrium begins. For higher temperatures life can only survive to the point where these accelerated processes will result in a rise in body temperature to the limiting level of from 106 to 108 F.

Air movement is an important factor in increasing heat loss by either convection or evaporation. The result is accomplished through removal of hot humid air from near the body surface and replacing it with cooler and relatively drier air. This is an important factor in maintaining thermal equilibrium either for persons at rest or at work in hot, humid conditions. For conditions in the comfort zone and below, excessive

velocities (particularly localized drafts) should be avoided since differential cooling of one area of the body may produce surprisingly unpleasant reactions in quite different parts of the body. In a recent experiment[15] it was shown that the application of an ice pack to an area of 60 sq cm on the back of the neck for 15 min caused a drop of 17 F in the skin temperature of the fingers and that this low temperature of the fingers persisted for one hour after the ice pack was removed.

ADAPTATION TO HOT CONDITIONS

It will be observed from Fig. 1 that a zone ranging from about 70 to 80 F, at 45 per cent relative humidity (or from 66 to 74 deg ET) the body is adequately able to maintain equilibrium through control of radiation and convection losses combined, and evaporative loss. This corresponds to the zone over which the largest percentage of persons find optimum comfort. For higher temperatures, up to an upper limit in the neighborhood of 100 F, at 45 per cent relative humidity (or approximately 87 deg ET) control is had through availability of perspiration on the body surface for evaporation. While a fair degree of temperature equilibrium is maintained over this range it is nevertheless had with considerable discomfort.

TABLE 2. PHYSIOLOGICAL RESPONSES TO HEAT OF MEN AT REST AND AT WORK[a]

Effective Temp	Actual Cheek Temp (Deg Fahr)	Men at Rest			Men at Work 90,000 ft-lb of Work per Hour			
		Rise in Rectal Temp (Deg Fahr per Hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Weight by Perspiration (Lb per Hr)	Total Work Accomplished (Ft-Lb)	Rise in Body Temp (Deg Fahr per Hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Wt by Perspiration (Lb per Hr)
60	....	....	....	....	225,000	0.0	6	0.5
70	....	0.0	0	0.2	225,000	0.1	7	0.6
80	96.1	0.0	0	0.3	209,000	0.3	11	0.8
85	96.6	0.1	1	0.4	190,000	0.6	17	1.1
90	97.0	0.3	4	0.5	153,000	1.2	31	1.5
95	97.6	0.9	15	0.9	102,000	2.3	61	2.0
100	99.6	2.2	40	1.7	67,000	4.0	103[b]	2.7
105	104.7	4.0	83	2.7	49,000	6.0[b]	158[b]	3.5[b]
110	....	5.9[b]	137[b].	4.0[b]	37,000	8.5[b]	237[b]	4.4[b]

[a]Data by A.S.H.V.E. Research Laboratory.
[b]Computed value from exposures lasting less than one hour.

Studies at the *John B. Pierce Laboratory of Hygiene*[16] have indicated the relation between discomfort and the degree of wetting of the body surface by perspiration for lightly clothed subjects in a semi-reclining position, and the investigators there have designated this as the *zone of evaporative regulation.* Work of the A.S.H.V.E. Research Laboratory[17]

[15]The Relative Influence of Radiation and Convection Upon the Temperature Regulation of the Clothed Body, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (*American Journal of Physiology*, Vol. 124, October, 1938, p. 51).

[16]Relations Between Atmospheric Conditions, Physiological Reactions, and Sensations of Pleasantness, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (*American Journal of Hygiene*, Vol. 26, July, 1937, p. 102). The Reactions of the Clothed Human Body to Variations in Atmospheric Humidity, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (*American Journal of Physiology*, Vol. 124, December, 1938, p. 692).

[17]Loc. Cit. Note 11.

over the past two decades has made available data on the relation between sensible perspiration and the atmospheric environment for normal persons at rest and at work.

For colder conditions below 70 F, with 45 per cent relative humidity (or about 66 deg ET) thermal equilibrium is maintained, first, by the amount of clothing worn; second, and to a smaller extent, by limiting availability of heat at the surface by decreasing peripheral blood circulation, which results in a drop in skin temperature; and third, by an increase in the metabolic rate. Here again, while thermal equilibrium is fairly well maintained, any drop in the skin temperature is accompanied by a certain degree of discomfort.

Studies at the A.S.H.V.E. Research Laboratory[18] and elsewhere[19] during the past two decades have made available a mass of information dealing with the physiological effects of hot atmospheres on workers and means to alleviate the distress and hazards associated therewith. This interest has been termed *air conditioning in industry*, or *the effects of hot atmospheres in industrial hygiene*, and is a growing factor in air conditioning applications. Table 2 gives some of the physiological responses of men at rest and at work, to hot environments. Recent physiological studies[20] indicate that frequent and continued exposure of workers to hot environments results in not only violent but subtle physiological derangement, affecting the leucocyte count of the blood and other factors dealing with man's mechanism of defense against infection.

Another of the deleterious effects of high temperatures is that the blood is diverted from the internal organs to the surface capillaries, in order to serve in the process of cooling. This affects the stomach, heart, lungs and other vital organs, and it is suggested that the feeling of lassitude and discomfort experienced is due in part to the anaemic condition of the brain. The stomach loses some of its power to act upon the food, owing to a diminished secretion of gastric juice, and there is a corresponding loss in the antiseptic and antifermentive action which favors the growth of bacteria in the intestinal tract[21]. These are considered to be the potent

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[18]A.S.H.V.E. RESEARCH REPORT No. 654—Some Physiological Reactions to High Temperatures and Humidities, by W. J. McConnell and F. C. Houghten (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 129). A.S.H.V.E. RESEARCH REPORT No. 672—Further Study of Physiological Reactions, by W. J. McConnell, F. C. Houghten and F. M. Phillips (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 353). A.S.H.V.E. RESEARCH REPORT No. 690—Air Motion, High Temperatures and Various Humidities—Reactions on Human Beings, by W. J. McConnell, F. C. Houghten and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 167). A.S.H V.E. RESEARCH REPORT No. 718—Work Tests Conducted in Atmospheres of High Temperatures and Various Humidities in Still and Moving Air, by W. J. McConnell and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p. 101). A.S.H.V.E. RESEARCH REPORT No. 719—Basal Metabolism Before and After Exposure to High Temperatures and Various Humidities, by W. J. McConnell, C. P. Yaglou and W. B. Fulton (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p. 123). A.S.H.V.E. RESEARCH REPORT No. 908—Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 541). A.S.H.V E. RESEARCH REPORT No. 1106—Air Conditioning in Industry—Physiological Reactions of Individual Workers to High Effective Temperatures, by W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten and M. B. Ferderber (A.S H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 59). A.S.H.V.E. RESEARCH REPORT No. 1153—Seasonal Variation in Reactions to Hot Atmospheres, by F. C. Houghten, A. A. Rosenberg and M. B. Ferderber (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940). Physiologic Effects of Hot Atmospheres, by F. C. Houghten, M. B. Ferderber and A. A. Rosenberg (*Industrial Medicine*, January, 1940, p. 7).

[19]A.S.H.V.E. RESEARCH REPORT No. 1151—The Peripheral Type of Circulatory Failure in Experimental Heat Exhaustion, by R. W. Keeton, F. K. Hick, Nathaniel Glickman and M. M. Montgomery (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940).

[20]A.S.H.V.E. RESEARCH REPORT No. 1153—Seasonal Variation in Reactions to Hot Atmospheres, by F. C. Houghten, A. A. Rosenberg and M. B. Ferderber (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940).

[21]Influence of Effective Temperature upon Bactericidal Action of Gastro-Intestinal Tract, by Arnold and Brody (*Proceedings Society Exp. Biol. Med.*, Vol. 24, 1927, p. 832).

factors in the increased susceptibility to gastro-intestinal disorders in hot summer weather.

In warm atmospheres, particularly during physical work, a considerable amount of chloride is lost from the system through sweating. The loss of this substance may lead to attacks of cramps, unless the salts are replaced in the drinking water. In order to relieve both cramps and fatigue, it

Table 3. Relation Between Metabolic Rate and Activity[a]

Activity	Hourly Metabolic Rate for Avg Person or Total Heat Dissipated, Btu per hour	Hourly Sensible Heat Dissipated, at 79 F, Btu per Hour	Hourly Latent Heat Dissipated, at 79 F, Btu per Hour	Moisture Dissipated per Hour	
				Grains	Pounds
Basal	291	145	145	978	0.140
Seated at Rest	384	225	159	1072	0.153
Reading Aloud (Seated)	420	225	195	1315	0.188
Standing at Rest	431	225	206	1389	0.198
Hand Sewing (Seated)	441	225	216	1457	0.208
Knitting 23 stitches per minute on Sweater	462	225	237	1598	0.228
Dressing and Undressing	468	225	243	1639	0.234
Tailor	482	225	257	1733	0.248
Singing	486	225	261	1760	0.251
Office Worker Moderately Active	490	225	265	1787	0.255
Light Work Standing	549	225	324	2185	0.312
Typewriting Rapidly	558	225	333	2246	0.321
Ironing with 5 lb iron	570	225	345	2326	0.332
Dishwashing—Plates, Bowls, Cups and Saucers	600	225	375	2529	0.361
Clerk Moderately Active Standing at Counter	600	225	375	2529	0.361
Book Binder	626	225	401	2704	0.386
Shoemaker	661	225	436	2940	0.420
Sweeping Bare Floor 38 Strokes per Minute	672	229	443	2987	0.427
Pool Player	680	230	450	3055	0.434
Walking 2 mph, Light Dancing	761	250	511	3446	0.492
Light Metal Worker (at Bench)	862	277	585	3945	0.564
Painter of Furniture (at Bench)	876	280	596	4019	0.574
Carpenter	954	307	647	4363	0.623
Restaurant Serving	1000	325	675	4552	0.650
Pulling Weight	1041	335	708	4774	0.682
Walking 3 mph	1050	339	711	4795	0.685
Walking 4 mph, Active Dancing, Roller Skating	1390	452	938	6325	0.904
Walking Down Stairs	1444	467	977	6588	0.941
Stone Mason	1490	490	1000	6744	0.963
Bowling	1500	490	1010	6811	0.973
Man Sawing Wood	1800	590	1210	8160	1.166
Swimming	1986	......	......	......	......
Running 5.3 mph	2268	......	......	......	......
Walking 5 mph	2330	......	......	......	......
Walking Very Fast 5.3 mph	2580	......	......	......	......
Walking Up Stairs	4365	......	......	......	......
Maximum Exertion Different People	3000-4800	......	......	......	......

[a]These metabolic rates were compiled by the A.S.H.V.E. Research Laboratory from actual tests, from other authoritative sources, and from estimates based upon various considerations. Division of the total heat dissipation into latent and sensible rates is based on actual test data and on various considerations for metabolic rates up to 1250 Btu per hour, and extrapolated for higher rates. Values for total heat dissipation for a person at rest apply for a dry-bulb temperature range from approximately 60 to 90 F; for other than rest conditions the values apply for a similar but lower temperature range. Below these temperature ranges metabolic rates and total rates of heat dissipation increase, while above these ranges metabolic rates increase slightly and total heat dissipation rates decrease rapidly. Division of total dissipation rates into sensible and latent heat holds only for a dry-bulb temperature of 79 F. For lower temperatures, sensible heat dissipation increases and latent heat decreases, while for higher temperatures the reverse is true.

is recommended that 6 g of sodium chloride and 4 g of potassium chloride be added to a gallon of water[22].

The deleterious physiologic effects of high temperatures exert a powerful influence upon physical activity, accidents, sickness and mortality. Both laboratory and field data show that physical work in warm atmospheres is a great effort, and that production falls progressively as the temperature rises. The incidence of industrial accidents reaches a minimum at about 68 F, increasing above and below that temperature. Sickness and mortality rates increase progressively as the temperature rises.

The need of air conditioning for workers in hot industries is growing rapidly and this should become an important field for the air conditioning engineer. The hot conditions may be remedied by any of the recognized comfort cooling applications. The choice of the type of system and cycle to be used in a given instance must be determined by the air conditioning engineer after a study of surrounding conditions.

Recently it has been shown that in some hot industries where a small number of workers are engaged in spaces of large volumetric capacity the worker himself, rather than the entire environment, can be cooled by either placing him in a small cooled and ventilated booth, by blowing cooled air over him, or by circulating cooled air through a loose-fitting suit[23].

RELATION OF AIR CONDITIONING NEEDS TO METABOLISM

The major objective of heating and ventilation is to balance heat losses from the human body. The basic factor is metabolism. The desirable environment, from the standpoint of heat loss, depends directly on the heat produced in the body and this heat may be over ten times as great when a man is exercising violently as when he is reclining and at rest. Therefore, there is no absolute optimum of air temperature or other environmental conditions, which will meet all cases. With moderate, (45 per cent) relative humidity and minimum air movement, an air temperature of 80 F has been found ideal for the lightly clothed subject at rest in a semi-reclining position, while normally clothed, healthy persons have been found comfortable at 72 and 77 F with 45 per cent relative humidity (or 67 and 71 deg ET, respectively) for winter and summer conditions. In factories where light work is performed in summer time, the ideal has been found to be about 76 F. For children (who have a high metabolism) at school, in winter clothing, 70 F has been considered correct; while in a gymnasium, 55 F has been recommended.

The wide variations in metabolic activities with which the engineer must be prepared to cope and the influence of such variations in metabolism on the heat load contributed by the human body to the environment are given in Table 3 and in Figs. 2, 3 and 4. It should be noted that metabolism and heat dissipation values are proportional to the body surface areas of the persons considered, and that the data referred to are

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[22]Some Effects of High Air Temperatures Upon the Miner, by K. N. Moss (*Transactions Institute of Mining Engineers*, Vol. 66, 1924, p. 284).

[23]A.S.H.V.E. RESEARCH REPORT NO. 1189—Local Cooling of Workers in Hot Industry, by F. C. Houghten, M. B. Ferderber and Carl Gutberlet (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, July, 1941, p. 462).

only for persons having an average surface area of 19.5 sq ft or that of the average adult American male, 5 ft 8 in. in height and weighing 150 lb, and will therefore not apply to many audiences made up largely of women or younger persons. Fig. 5, taken from the work of Du Bois[24] gives the relation of body surface area to height and weight, which may serve to correct the data for other audiences than adult men. The curves in the figures are based on certain averages of test results with different humidities, and are sufficiently accurate for most practical applications. Where

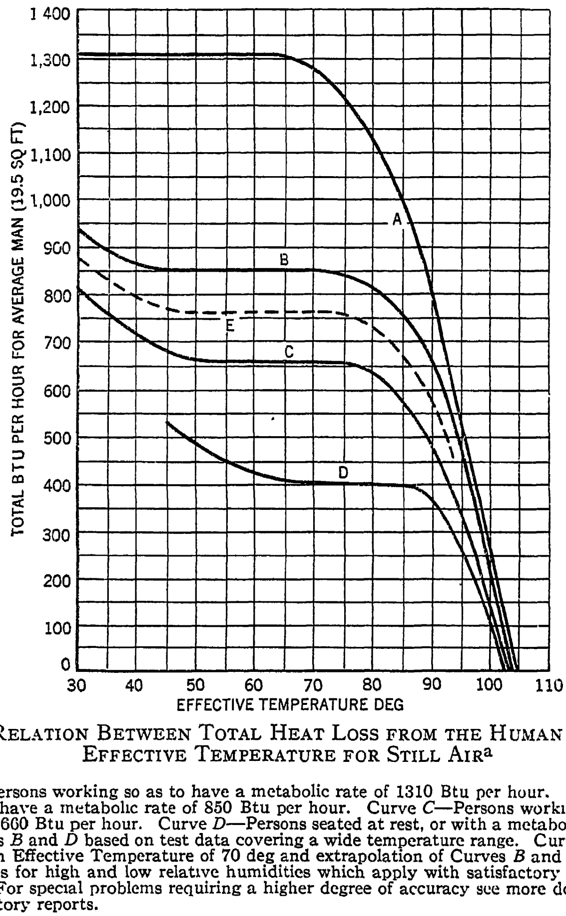

FIG. 2. RELATION BETWEEN TOTAL HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE FOR STILL AIR[a]

[a]Curve *A*—Persons working so as to have a metabolic rate of 1310 Btu per hour. Curve *B*—Persons working so as to have a metabolic rate of 850 Btu per hour. Curve *C*—Persons working so as to have a metabolic rate of 660 Btu per hour. Curve *D*—Persons seated at rest, or with a metabolic rate of 400 Btu per hour. Curves *B* and *D* based on test data covering a wide temperature range. Curves *A* and *C* based on test data at an Effective Temperature of 70 deg and extrapolation of Curves *B* and *D*. All curves are averages of values for high and low relative humidities which apply with satisfactory accuracy for most considerations. For special problems requiring a higher degree of accuracy see more detailed A.S.H.V.E. Research Laboratory reports.

greater precision in the applications of the results is required, or for extreme variations in temperature and humidity, the reports[25] covering the A.S.H.V.E. Laboratory work may be consulted.

The curves in Figs. 2, 3 and 4, and proper interpolation between these curves make it possible to apply the data to persons engaged in any type

[24]DuBois, D. and E. F. (*Archives of Internal Medicine*, 1916, Vol. 18, p. 865).
[25]Loc. Cit. Note 11.

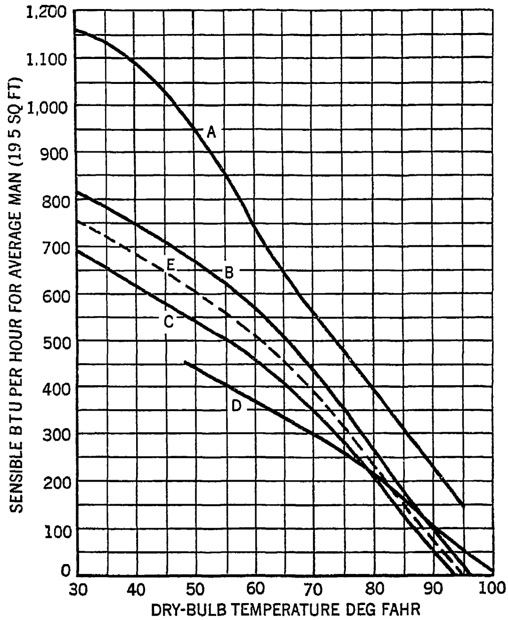

FIG. 3. RELATION BETWEEN SENSIBLE HEAT LOSS FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE FOR STILL AIR[a]

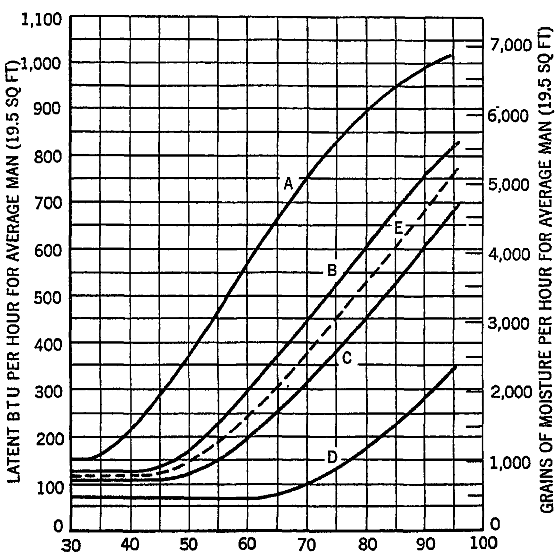

FIG. 4. LATENT HEAT AND MOISTURE LOSS FROM THE HUMAN BODY BY EVAPORATION, IN RELATION TO DRY-BULB TEMPERATURE FOR STILL AIR CONDITIONS[a]

[a]Loc. Cit. See footnote a, Fig. 2.

of work or physical activity, providing the resulting metabolic rate is known. As an example, if it is found that a certain type of work results in a metabolic rate of approximately 760 Btu per hour for an average person working in an atmosphere of 70 ET, then his total rate of heat dissipation to atmospheres of various temperature will be approximately as given by the broken-line curve in Fig. 2. The broken line curves in Figs. 3 and 4 give the rate of sensible and latent heat dissipation of the person for different dry-bulb temperatures.

ACCLIMATIZATION

Acclimatization and the factor of psychology are two important influences in air conditioning which cannot be ignored. The first is man's ability to adapt himself to changes in air conditions; the second is an intangible matter of habit and suggestion.

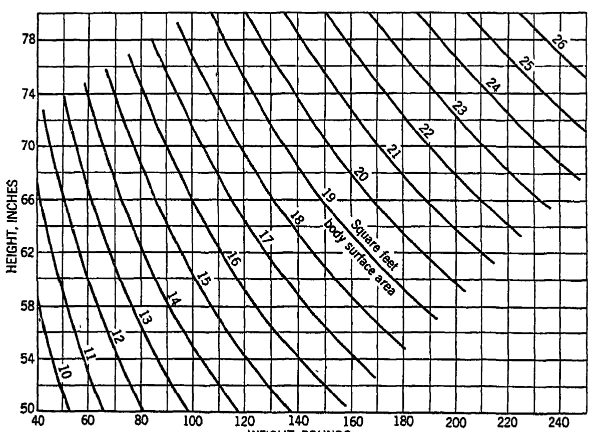

FIG. 5. CHART FOR DETERMINING SURFACE AREA OF INDIVIDUALS FOR HEIGHT AND WEIGHT GIVEN

Some persons regard the unnecessary endurance of cold as a virtue. They believe that the human organism can adapt itself to a wide range of air conditions with no apparent discomfort or injury to health. In the light of present knowledge of air conditioning these views are not justified. Acclimatization to extreme conditions involves a strain upon the heat regulating system and interferes with the normal physiologic functions of the human body. Thousands of years in the heat of Africa do not seem to have acclimatized the Negro to a temperature exceeding 80 F. The same holds true of northern races with respect to cold, although the effects are mitigated by artificial control. An environment averaging 64 F for the 24-hour period has been indicated as associated with minimal mortality[26].

Within limits, however, there does occur a definite adaptation to ex-

[26]Civilization and Climate, by Ellsworth Huntington, Yale University Press, 1928.

ternal temperature level. People and animals raised under conditions of tropical moist heat stand chilling poorly as they are unable quickly to increase internal combustion to keep up the body temperature. For this reason they have trouble standing the cold, stormy weather of the tem-

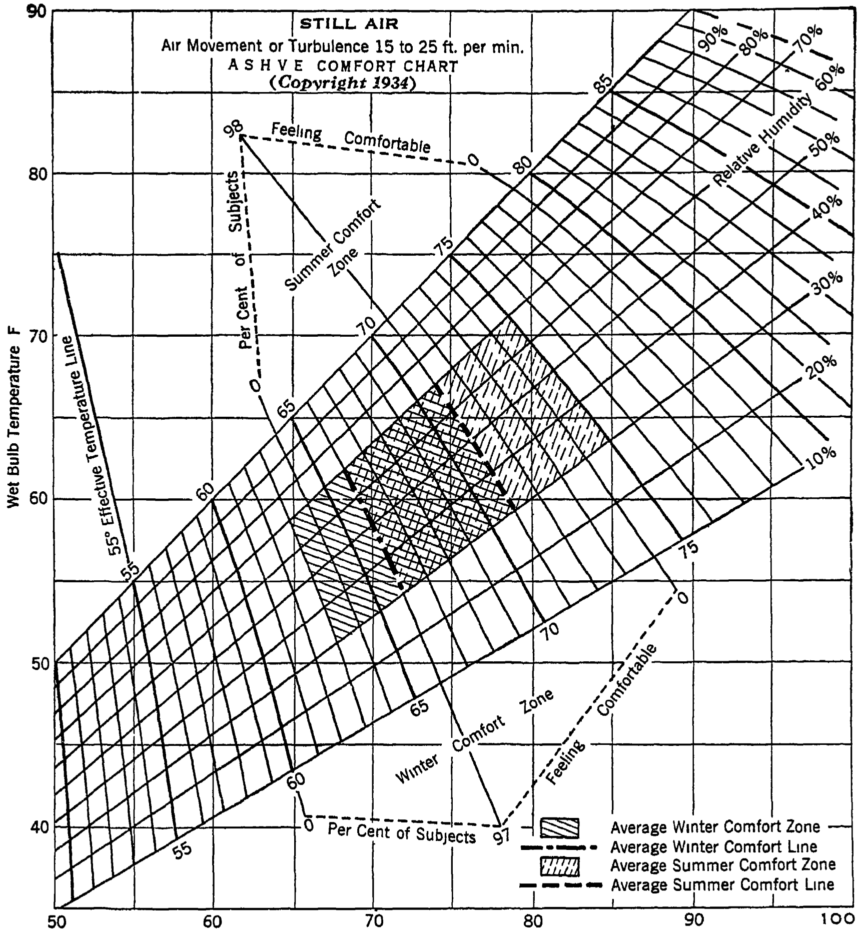

FIG. 6. A.S.H.V.E. COMFORT CHART FOR AIR VELOCITIES OF 15 TO 25 FPM (STILL AIR)

Note.—Both summer and winter comfort zones apply to inhabitants of the United States only. Application of winter comfort line is further limited to rooms heated by central station systems of the convection type. The line does not apply to rooms heated by radiant methods. Application of summer comfort line is limited to homes, offices and the like, where the occupants become fully adapted to the artificial air conditions. The line does not apply to theaters, department stores, and the like where the exposure is less than 3 hours.

perate zones, and when exposed to it are very susceptible to respiratory infections. Likewise, people living in cool climates suffer greatly in the moist heat of the tropics until their adaptive mechanism has been adjusted. Within a few years, however, they find themselves reacting as natives to the new environment.

The adaptive level changes somewhat with the season[27]. There are also marked differences between the sexes. In the cold zone the thickness of the thermal insulating tissues of women is almost double that of men, although the sensory responses to cold are similar. In the hot zone, the threshold of sweating and skin temperature levels are both higher for women.

Finally, the thickness and insulating value of the clothing worn is an important factor in the determination of the comfort level.

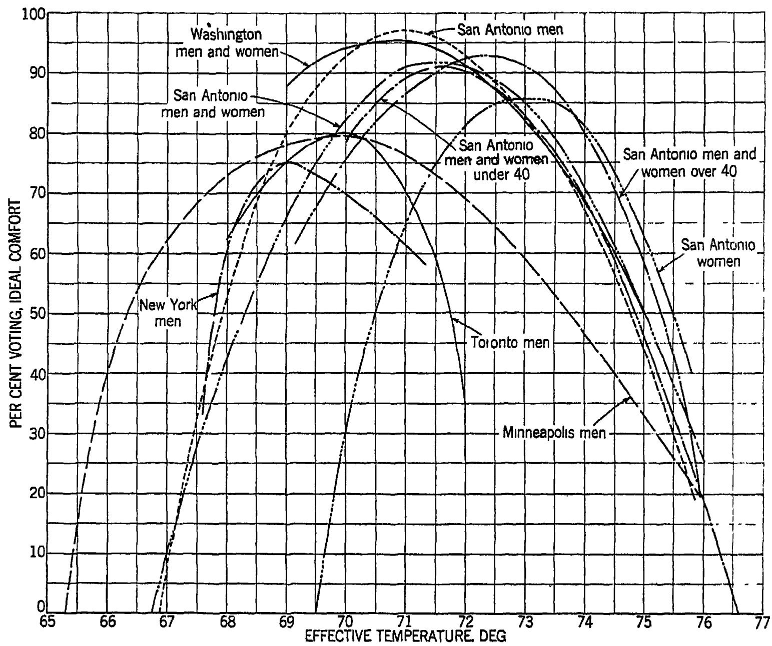

FIG. 7. RELATION BETWEEN EFFECTIVE TEMPERATURE AND PERCENTAGE OBSERVATIONS INDICATING COMFORT

EFFECTIVE TEMPERATURE INDEX

Sensations of warmth or cold depend, not only on the temperature of the surrounding air as registered by a dry-bulb thermometer, but also upon the temperature indicated by a wet-bulb thermometer, upon air movement and upon radiation effects. Dry air at a relatively high temperature may feel cooler than air of considerably lower temperature with a high moisture content. Air motion makes any moderate condition feel cooler. Radiation to cold or from warm surfaces is another important factor under certain conditions.

Combinations of temperature, humidity, and air movement which induce the same feeling of warmth are called thermo-equivalent condi-

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[27]The Reactions of the Clothed Human Body to Variations in Atmospheric Humidity, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (*American Journal of Physiology*, Vol. 124, December, 1938, p. 692).

tions. A series of studies[28] at the A.S.H.V.E. Research Laboratory, Pittsburgh, established the equivalent conditions met with in general air conditioning work. This scale of thermo-equivalent conditions not only indicates the sensation of warmth, but also to a considerable degree determines the physiological *effects* on the body induced by heat or cold. For this reason, it is called the *effective temperature* scale or index.

Effective temperature is an empirically determined index of the degree of warmth perceived on exposure to different combinations of temperature, humidity, and air movement. It was determined by trained subjects who compared the relative warmth of various air conditions in two adjoining conditioned rooms by passing back and forth from one room to the other.

The numerical value of the index for any given air conditions is fixed by the temperature of calm (15 to 25 fpm air movement) saturated air which induces a like sensation of warmth or cold. Thus, any air condition has an effective temperature of 60 deg, when it induces a sensation of warmth like that experienced in calm air at 60 deg saturated with moisture. The *effective temperature index* cannot be measured directly but is determined by the dry- and wet-bulb temperature observations and by reference to the Comfort Chart (see Figs. 6, 7 and 8) or tables. The relation of winter and summer sensations of comfort to wet- and dry-bulb temperature at low air movement is shown in Fig. 6. This chart, published by an A.S.H.V.E. Technical Advisory Committee[29], is based on research prior to 1932. Later studies by the A.S.H.V.E. Research Laboratory indicates somewhat higher temperatures for winter comfort, while Fig. 7 shows considerable variation in the requirements for comfort in summer cooled and air conditioned space. Relations between moisture content and various dry-bulb temperatures to wet-bulb readings and effective temperatures are depicted in Fig. 8. Effective temperatures for various combinations of wet- and dry-bulb temperatures and air movement are given in Fig. 9.

A long series of studies have been made to determine the optimum effective temperature for comfort of normal persons in both winter and summer air conditioned space, in different geographical regions and for different age groups of men and women. A group of these studies[30] was

[28]A.S.H.V.E. RESEARCH REPORT No. 673—Determination of the Comfort Zone, by F. C. Houghten and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p 361). A.S.H.V.E. RESEARCH REPORT No. 691—Cooling Effect on Human Beings by Various Air Velocities, by F. C. Houghten and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 193). A.S.H.V.E. RESEARCH REPORT No. 717—Effective Temperature with Clothing, by C. P. Yaglou and W. E. Miller (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p. 89). A.S.H.V E. RESEARCH REPORT No. 755—Effective Temperature for Persons Lightly Clothed and Working in Still Air, by F. C. Houghten, W. W. Teague and W. E. Miller (A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926, p. 315).

[29]How to Use the Effective Temperature Index and Comfort Charts, by C. P. Yaglou, W. H. Carrier, Dr. E. V. Hill, F. C. Houghten and J. H. Walker (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 410).

[30]A.S.H.V.E. RESEARCH REPORT No. 1035—Comfort Standards for Summer Air Conditioning, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 215). A.S.H.V.E. RESEARCH REPORT No. 1055—Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 145). A.S.H.V.E. RESEARCH REPORT No. 1088—Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley (A.S H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 337). Cooling Requirements for Summer Comfort Air Conditioning in Toronto, by C. Tasker (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 549). A.S.H.V.E. RESEARCH REPORT No. 1127—Reactions of Office Workers to Air Conditioning in South Texas, by A. J. Rummel, F. E. Giesecke, W. H. Badgett and A. T. Moses (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 459). A.S.H.V.E. RESEARCH REPORT No. 1136—Summer Cooling Requirements in Washington, D. C., and Other Metropolitan Districts, by F. C. Houghten, Carl Gutberlet and Albert A. Rosenberg (A S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 577). A.S.H.V.E. RESEARCH REPORT No. 1161—Reactions of 745 Clerks to Summer Air Conditioning, by W. J. McConnell and M. Spiegelman (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940,)

made between 1935 and 1940 by the A.S.H.V.E. Laboratory in Pittsburgh, and in several metropolitan districts of the United States in cooperation with the managements of offices employing large numbers of workers. Some of the results are shown in Fig. 7. Taking all of these studies together, women of all age groups studied indicate an average effective temperature for comfort 1.1 deg higher than for men. All men and

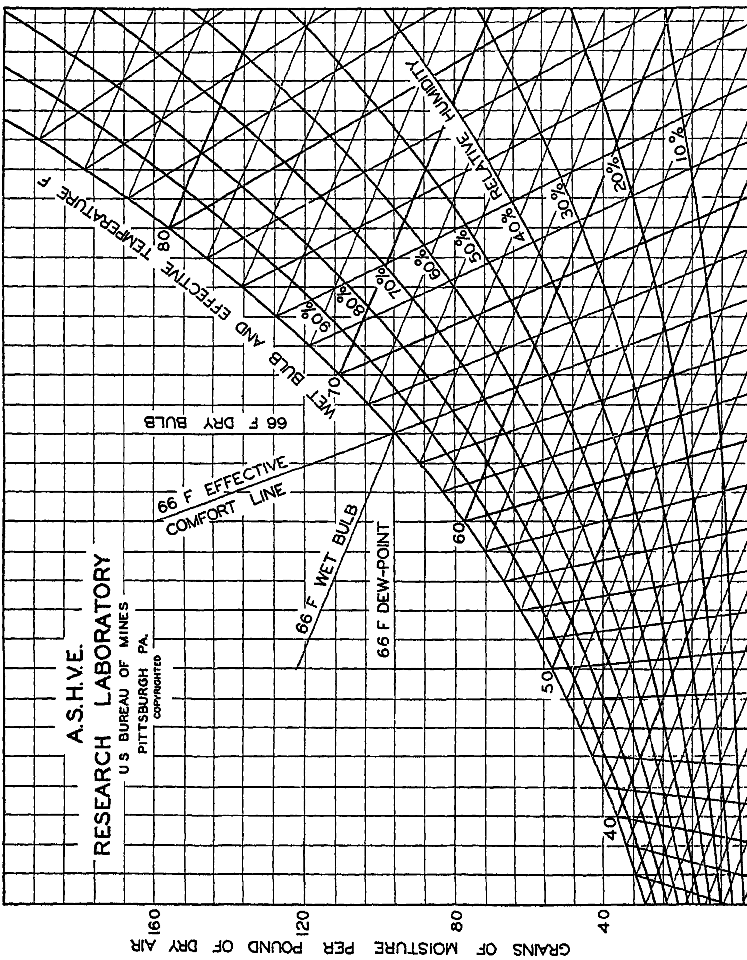

Fig. 8. Psychrometric Chart, Persons at Rest, Normally Clothed, in Still Air

women, beyond the age of 40 years, show an average desire for 0.9 deg ET higher than those below this age; while the men and women, respectively, beyond 40 desired effective temperatures of 0.8 and 1.2 deg higher than those below 40. The persons serving in all of these studies were representative of office workers clothed for air conditioned space in the summer season and engaged in the customary sedentary activity of office workers.

The 66 deg ET indicated as giving optimum comfort for winter con-

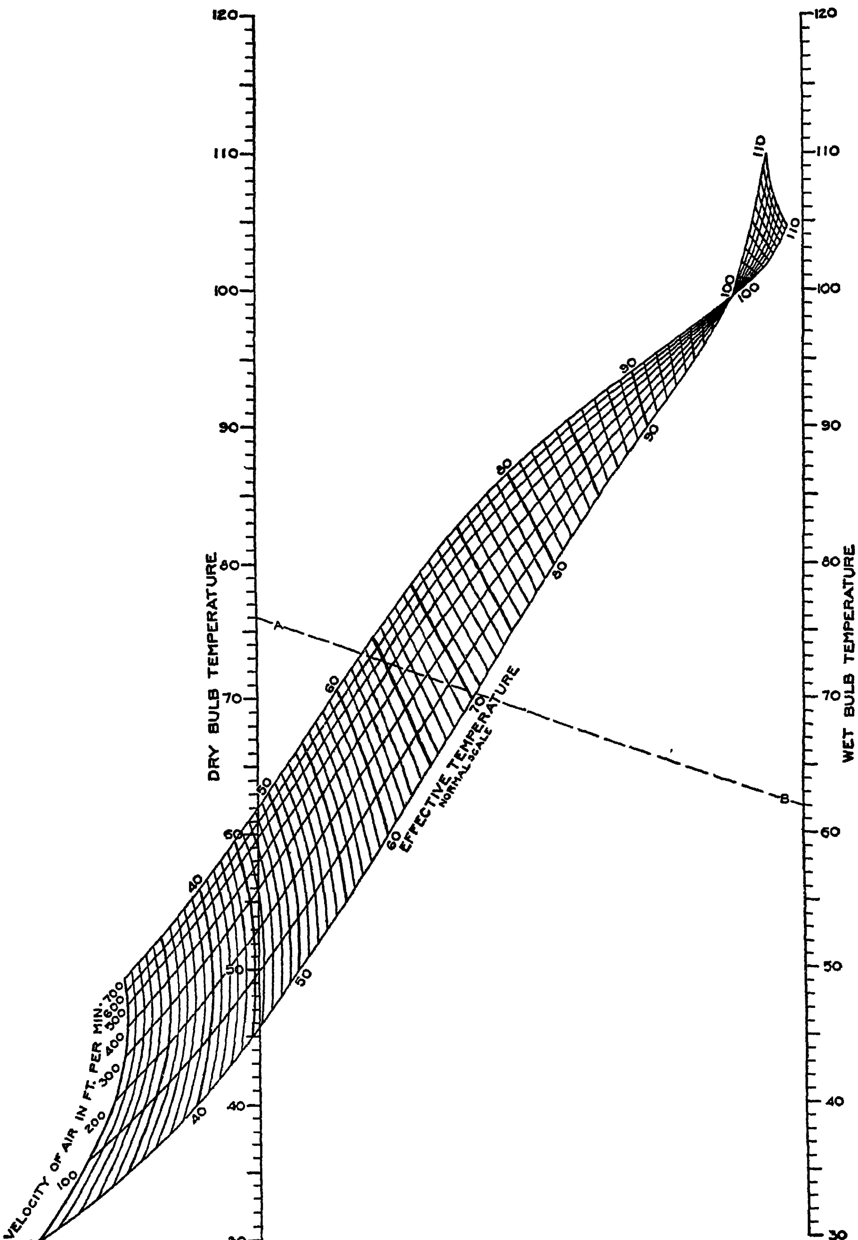

FIG. 9. EFFECTIVE TEMPERATURE CHART SHOWING NORMAL SCALE OF EFFECTIVE TEMPERATURE. APPLICABLE TO INHABITANTS OF THE UNITED STATES UNDER FOLLOWING CONDITIONS:

A. Clothing: Customary indoor clothing. *B. Activity*: Sedentary or light muscular work. *C. Heating Methods*: Convection type, *i.e.*, warm air, direct steam or hot water radiators, plenum systems.

ditions in Fig. 6 and determined prior to 1932, has more recently been checked; first, by occasional observations, and second, by consistent laboratory study[31], indicating that a higher optimum of 67 deg ET, or possibly as high as 68 deg ET, is now desired by the majority of occupants.

All of these studies aimed to determine the optimum effective temperature for persons during both winter and summer, and for different geographical regions and age groups indicate a total spread in optimum conditions ranging from a low of 67 deg ET for winter heating and air conditioning, to a high of 73 deg ET for summer cooling and air conditioning.

The spread for summer cooling and air conditioning is confined entirely to an effective temperature range of from 69 to 73 deg, and it may be presumed that for winter conditioning a like spread would be had; while for inter-seasonal conditions there will be a fluctuation between these two ranges.

Recent studies[32] indicate that for the average individual a temperature change of about 3 deg ET is required to change a person's sensation from *ideally comfortable* to *cool* or *warm*. From this it may be observed that necessary variations in the effective temperature of air conditioned space for optimum comfort for most persons, regardless of age or geographical location, need little differentiation for either the winter season or the summer season, and not more than about 4 deg on the average between seasons.

Acclimatization and habits of clothing and diet account for these variations. As a result of a recent analysis[33] of all of the evidence available by the A.S.H.V.E. Technical Advisory Committee on Sensations of Comfort, a variation of 3 deg spread in optimum effective temperature for summer cooling and air conditioning with geographical location has been proposed. The available information indicates rather clearly that changes in weather conditions over a period of a few days do not acclimate people to a desire for different indoor conditions, but in general, people experiencing low temperatures over an extended period of time become acclimated to desiring lower indoor temperatures, while those experiencing higher temperatures become acclimated to a desire for higher indoor temperatures. It is obvious that a person spending a considerable portion of his time in space conditioned to his comfort will become acclimated to his indoor environment. While few people enjoy air conditioning for more than a small percentage of the total time, there is some evidence that persons experiencing comfort air conditioning a large part of the time tend to become acclimated to about 70 or 71 deg ET.

The entering shock to occupants of summer cooled and air conditioned space may at times be important, and is due to the rapid evaporation of perspiration accumulated during the occupant's previous stay in the hot outside. While recent studies[34] show that for healthy individuals this

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[31]A S H.V.E RESEARCH REPORT No. 1173—Radiation as a Factor in the Sensation of Warmth, by F. C. Houghten, S. B. Gunst and J. Suciu, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941,)

[32]Loc. Cit. Note 31.

[33]Comfort with Summer Air Conditioning, by Thomas Chester, N. D. Adams, C. R. Bellamy, G. D. Fife, E. P. Heckel, Dr. W. J. McConnell, F. C. McIntosh, A. B. Newton, B. F. Raber and C. Tasker. (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1941).

[34]A.S.H V.E. RESEARCH REPORT No. 1102—Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices, by A. B. Newton, F. C. Houghten, Carl Gutberlet, R. W. Qualley and M. C. W. Tomlinson (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 571).

shock is usually a pleasant experience, for others it may result in unpleasant or even harmful chills. This fact should be taken into consideration in applying summer cooling and air conditioning, particularly to spaces where a large number of the occupants may enter for only a short time, 15 min or less. Such occupants will be satisfied with less cooling. For long occupancy very little deviation from the optimum effective temperature is practical.

Radiation between the occupant of an enclosure and the surfaces of the room itself and objects within the room, including windows, heating and cooling equipment, and other occupants, has an important bearing on the feeling of warmth and may alter to some measurable degree the optimum

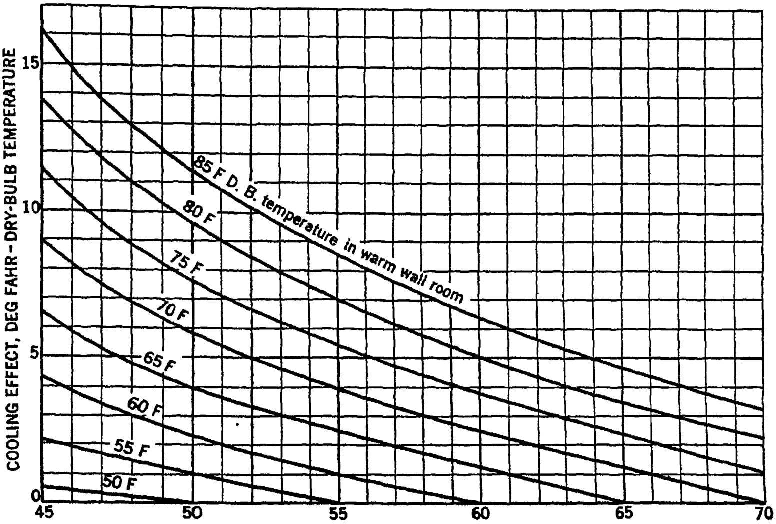

FIG. 10. COOLING EFFECT OF THREE COLD WALLS IN A SMALL EXPERIMENTAL ROOM, AS DETERMINED BY COMPARISON WITH SENSATIONS IN A ROOM OF UNIFORM WALL AND AIR TEMPERATURE

conditions for comfort indicated previously. Fig. 10[35] shows the necessary elevation in the dry-bulb temperature of the air to compensate for the lower temperature of three of four side-wall surfaces, which indicates that for this condition each degree reduction in the average of the three wall surface temperatures there must be an elevation of 0.3 deg in the dry-bulb temperature of the air to compensate. Recent studies by the A.S.H.V.E. Research Laboratory[36] on the effect of radiation within an enclosure, including the effect of panel heating, indicate that for each degree elevation or depression of the mean radiant temperature above or below the air temperature requires about 0.5 deg counterchange in effective temperature of the air. Since the mean radiant temperature of the surroundings

[35]A.S.H.V.E. RESEARCH REPORT No. 946—Cold Walls and Their Relation to Feeling of Warmth, by F. C. Houghten and Paul McDermott. (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 83).

[36]Loc. Cit. Note 31.

is affected by cold, uninsulated walls and windows, particularly single glazed windows, as well as by heating units placed within the room, including panel heaters, these factors must be compensated. Likewise, in densely occupied spaces, such as classrooms, theaters and auditoriums, somewhat lower temperatures may be necessary than those indicated by the comfort line on account of counter-radiation between the bodies of occupants in close proximity to each other, which also will elevate the mean radiant temperature of the room.

The sensation of comfort, insofar as the physical environment is concerned, is not absolute but varies considerably among certain individuals. Therefore, in applying the air conditions indicated, it should not be expected that all the occupants of a room will feel perfectly comfortable. The curves in Fig. 7 indicate that some persons require temperatures as much as 4 and 6 deg lower and higher than the optimum for the average. In this connection it is of interest to note that from the characteristic shape of the curves that in general people will object more quickly to a few degrees drop in temperature from the average optimum than will be the case for the same number of degrees overheating. However, when optimum comfort temperatures are applied in accordance with foregoing recommendations, the majority of the occupants should be comfortable, and it should be expected that there will be a few too warm and a few too cold. These individual differences among the minority should be counteracted by suitable clothing.

Satisfactory comfort conditions for persons at work[37] are found to vary depending upon the rate of work and the amount of clothing worn. In general, the greater the degree of activity, the lower the effective temperature necessary for optimum comfort. However, recent work by the A.S.H.V.E. Research Laboratory[38] indicates that under certain conditions moderate activity on the part of a person standing up and moving about may result in a slightly higher optimum effective temperature than for a person seated at rest, because of the larger body surface area exposed to heat elimination and the increase in effective air movement over his body. Where few workers occupy a large space in hot industries, recent work by the A.S.H.V.E. Research Laboratory[39] shows that they may be made reasonably comfortable by blowing relatively small volumes of slightly cooled air over them or through their clothing.

For prematurely born infants, the optimum temperature varies from 100 to 75 F, depending upon the stage of development. The optimum relative humidity for these infants is placed at 65 per cent[40]. No data are yet available on the optimum air conditions for full term infants and young children up to school age. Satisfactory air conditions for these age groups are assumed to vary from 75 to 68 F with natural indoor humidities. For school children, the studies of the New York State Commission on

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[37]A.S.H.V.E. Research Report No. 755—Effective Temperature for Persons Lightly Clothed and Working in Still Air, by F. C. Houghten, W. W. Teague and W. E. Miller (A.S.H.V.E. Transactions, Vol. 32, 1926, p. 315).

[38]A.S.H V.E. Research Report No. 1106—Air Conditioning in Industry, by W L. Fleisher, A. E. Stacey, Jr., F. C. Houghten and M. B. Ferderber (A.S.H.V.E. Transactions, Vol. 45, 1939, p. 59).

[39]Loc. Cit. Note 23.

[40]Application of Air Conditioning to Premature Nurseries in Hospitals, by C. P. Yaglou, Philip Drinker and K. D. Blackfan (A.S H.V.E. Transactions, Vol. 36, 1930, p 383).

Ventilation place the optimum air conditions at 66 to 68 F temperature with a moderate humidity and a moderate but not excessive amount of air movement[41]. A great number of persons seem to be fairly content with a higher plane of indoor temperature, particularly when the matter of first cost and operating cost of a cooling plant is given due consideration. Recent studies by the University of Illinois[42] in cooperation with the A.S.H.V.E. Committee on Research indicate that effective temperatures as high as 74.5 deg are acceptable in the living quarters of a residence, and while this condition is not representative of optimum comfort it provides sufficient relief in hot weather to be acceptable to the majority of users. It should be emphasized, however, that these high temperatures are borderline cases that may be acceptable largely in the interest of economy. Comprehensive studies by the A.S.H.V.E. Research Laboratory[43] in cooperation with office staffs in widely distributed regions, including San Antonio, Texas, Minneapolis, Washington, D. C., and the Metropolitan Life Building, New York City (see Fig. 7), show conclusively that lower effective temperatures are required for optimum comfort, while in a few instances this optimum has been found as high as 73 deg and as low as 69 deg, or an average of about 71 deg.

PHYSIOLOGICAL OBJECTIVES OF HEATING AND VENTILATION

Aside from the removal of toxic fumes and dusts from heating appliances and industrial processes, the chief task of the heating and ventilating engineer is to keep his clients warm in winter and cool in summer.

For the normally vigorous person, normally clothed, and at rest, an air temperature of 65 F should be provided at knee-height, 18 in. in order to prevent chilling of the legs and feet. With some heating systems, this will correspond to 70 F at a 5 ft height. Air temperature may be increased or decreased in order to compensate for deviations of *mean radiant temperature* above or below air temperature.

In rooms occupied by persons of sub-normal vitality, knee-height temperatures must be higher than 65 F. Since dwellings are designed for occupancy by old people and children, the heating system should be able to provide a temperature of 70 F at knee-height under ordinary winter conditions.

The maintenance of such conditions as these in winter depends on three major factors, the heat produced in the occupied space, the heat absorbed from the sun and the heat loss through the walls, floor and ceiling of the structure to cold air and earth. Taking these up in the order in which they occur, in planning a new structure it is essential to remember the important effect of orientation and fenestration of the building with respect to the absorption of radiant heat from the sun. It has recently been shown that, in the vicinity of New York, effective sun-heat on a wall facing south is almost five times as great in winter as

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[41]Ventilation Report of the New York State Commission on Ventilation (E. P. Dutton Co., N. Y., 1923).

[42]A.S.H.V.E. RESEARCH REPORT No. 1012—Study of Summer Cooling in the Research Residence for the Summer of 1934, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 207).

[43]Loc. Cit. Note 30.

in summer, but on a wall facing west-north-west it is six times as great in summer as in winter[44]. The orientation of the same one-story house (in a laboratory model) was changed from a position in which its principal rooms faced northwest to a position in which these rooms (with re-arranged and slightly increased fenestration) faced west of south. This change decreased average summer sun-heat to one-ninth and increased average winter sun-heat to fourfold of its value with the original orientation.

The choice between the various methods of heating depends, of course, on many engineering and other factors. From the standpoint of human health and comfort, however, it is important to minimize floor-ceiling differentials as far as possible to avoid hot heads and cold feet. Furthermore, when the problem is a heating one, low air movement is desirable, since air temperature must be raised to balance the cooling effect of air motion.

Where occupants are closely aggregated, a new problem comes in, the removal of the excess heat produced by the human body itself. If the temperature of such a space be correctly adjusted when the occupants enter, it will steadily rise during the period of occupancy as a result of the heat produced by the occupants in the process of metabolism. Of the 400 Btu given off in metabolism 100 would perhaps be lost in evaporation, leaving 300 Btu per person per hour to warm the air. In a room containing many persons, the effects of this body heat can be neutralized by outside air without producing unpleasant and dangerous drafts on those near the windows or other inlets. The supply of air before it reaches the occupant should be so tempered as to avoid drafts but in an amount and at a temperature which will remove the sensible heat produced by metabolism. With no heat loss through walls (as in an interior auditorium) this will require 28 cfm of air per person when admitted at 60 F, and an average temperature of 70 F for air leaving the room. Under practical conditions, with one or more cold walls, and a room containing a moderate number of occupants and ample cubic space, window ventilation with deflectors and a gravity exhaust duct may suffice. With crowded rooms, and with any rooms containing 50 or more occupants, forced ventilation will be essential.

SUMMER COMFORT

The problem of keeping cool in summer is physiologically as important as keeping warm in winter. In summer the relative humidity of the atmosphere is of great importance, along with air temperature, air movement, and wall temperature. There is no very practical method of cooling walls, but summer comfort can be promoted by modifying either one of the other three factors involved.

Increase of comfort by air movement may be had by the promotion of natural circulation by cross or through ventilation; and here the architect is responsible for providing fenestration which will make such natural

[44]Solar Radiation as Related to Winter Heating in Residences, by H. N. Wright (*Report of John B. Pierce Foundation*, January 20, 1936).

ventilation possible. In the lowest cost housing this should be considered as essential.

The direct control of air temperature and humidity is, of course, the ideal solution where the cost of a complete air conditioning equipment can be met. Where this objective is attained, there are two schools of thought concerning the relation between temperature and humidity to be maintained. For a given effective temperature some engineers favor comparatively low temperature with a high humidity as this results in a reduction of refrigeration requirements. Preliminary experiments at the A.S.H.V.E. Laboratory[45] would seem to indicate no appreciable impairment of comfort with relative humidities as high as 80 per cent, provided the effective temperature is between 70 and 72 or 73 deg.

The second school favors a higher dry-bulb temperature, according to the prevailing outdoor dry-bulb, with a comparatively low humidity (well below 50 per cent), the main purpose being an assumed reduction in temperature contrasts upon entering and leaving the cooled space and to keep the clothing and skin dry. This second scheme requires more refrigeration with the present conventional type of apparatus.

INFLUENCE OF HUMIDITY

The limitation of the comfort zones in Fig. 6 with respect to humidity are not final but must be adhered to closely. Relative humidities below 30 per cent may prove satisfactory from the standpoint of comfort. In mild weather comparatively high relative humidities seem to be entirely feasible, but in cold weather they are objectionable on account of condensation and frosting on the windows. Information on this subject is given in Chapter 4.

As to the effects of dryness of the air, *per se*, and irrespective of thermal effects, there is a common belief that dry air in itself exerts a harmful effect upon the skin and mucous membranes; but there is no convincing evidence that the increase of atmospheric moisture which can practically be introduced by humidification into the air of cool occupied rooms has any effect upon health and comfort. All controlled experiments on this point have yielded negative results; and the respiratory membranes of industrial workers exposed to hot moist air are distinctly abnormal compared with those of workers exposed to hot dry air[46].

For the premature infant, a high relative humidity of about 65 per cent is demonstrably beneficial to health and growth[47] until the infants reach a weight of about 5 lb. No such clear-cut evidence exists in the case of adults. In the comfort zone experiments of the A.S.H.V.E. Research Laboratory, the relative humidity was varied between the limits of 30 and 70 per cent approximately, but the most comfortable range has not been determined. In similar experiments at the *Harvard School of Public Health*, the majority of the subjects were unable to detect sensa-

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[45]A.S.H V.E. RESEARCH REPORT NO. 1035—Comfort Standards for Summer Air Conditioning, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 215). A.S.H.V.E. RESEARCH REPORT NO. 1055—Cooling Requirements for Summer Air Conditioning, by F. C. Houghten, F. E. Giesecke C. Tasker and Carl Gutberlet (A.S.H V.E. TRANSACTIONS, Vol. 43, 1937, p. 145).

[46]Loc. Cit. Note 37.

[47]Loc. Cit. Note 40.

tions of humidity (*i.e.*, too high, too low, or medium) when the relative humidity was between 30 per cent and 60 per cent with ordinary room temperatures which is in accord with other studies[48,49].

INFLUENCE OF AIR MOVEMENT

Air movement has a powerful influence on the factors involved in thermal equilibrium of the body. An understanding of the phenomena involved is best had through a consideration of the purely physical factors involved in the effect of air movement on heat dissipation from inanimate surfaces by radiation, convection and evaporation. Thermal equilibrium of the human body is more complex because of the physiological control exercised in permitting the body surface temperature to drop when factors influencing heat loss are unavoidably increased without additional clothing and by the making available of perspiration for evaporation.

Air movement does not affect radiation loss, provided there is no change in the skin temperature. However, if there is excessive cooling and lowering of the skin temperature due to increased convection loss, then radiation loss (which varies as the differences of the fourth power of the absolute temperatures of the radiator and receiver) decreases. It has been shown by the work at the *John B. Pierce Laboratory of Hygiene*[50] and by the A.S.H.V.E. Research Laboratory[51] that radiation may thus actually descrease due to air movement in relatively cool atmospheres.

Convection loss from any surface, including that of the clothed body, is greatly increased by air movement, provided the surface temperature remains the same. In cool atmospheres, unless increased clothing is worn, heat loss due to air movement may be accompanied by a drop in body surface temperature.

Heat loss by evaporation is greatly increased by air movement, provided surface temperature and moisture available for evaporation (or the wetness of the surface) are constant. However, since in the human body perspiration is only made available when there is need for increased evaporative heat loss due to reduction in convection loss increased air movement is accompanied by decreased perspiration and evaporative cooling in moderately cool atmospheres. In very hot atmospheres, particularly with low vapor pressure, evaporative cooling may be increased by air movement so as to increase the maximum temperature level at which thermal equilibrium may be maintained. Results of studies at the A.S.H.V.E. Research Laboratory[52] and at the *John B. Pierce Labora-*

[48]Humidity and Comfort, by W. H. Howell (*The Science Press*, April, 1931).

[49]Effect of Variation in Relative Humidity upon Skin Temperature and Sense of Comfort, by U. Miura (*American Journal of Hygiene*, Vol. 13, 1931, p. 432).

[50]Loc. Cit. Note 13.

[51]Loc. Cit. Note 11.

[52]A.S.H.V.E. RESEARCH REPORT No. 691—Cooling Effects on Human Beings Produced by Various Air Velocities, by F. C. Houghten and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 193). A.S.H.V.E. RESEARCH REPORT No. 717—Effective Temperature with Clothing, by C. P. Yaglou and W. E. Miller (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p. 89). A.S.H.V.E. RESEARCH REPORT No. 690—Air Motion, High Temperatures and Various Humidities-Reactions on Human Beings, by W. J. McConnell, F. C. Houghten and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 167). A.S.H.V.E. RESEARCH REPORT No. 718—Work Tests Conducted in Atmospheres of High Temperatures and Various Humidities in Still and Moving Air, by W. J. McConnell and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p. 101).

tory of Hygiene[53] give data on the effect of air movement on heat dissipation for normally clothed, standing and seated subjects, and for lightly clothed and semi-reclining subjects, respectively. Fig. 9 resulting from A.S.H.V.E. research, shows the increase in dry-bulb and wet-bulb temperatures for the same effective temperature with air velocities ranging from 20 to 700 fpm.

Air velocities may be used for effective cooling; however, great care must be exercised to avoid drafts due to uneven cooling of the body surface. During the heating season air velocities in excess of 25 to 30 fpm usually give undersirable effects. With summer cooling and air conditioning higher velocities up to 40 or 50 fpm, if properly controlled, seem to give satisfactory conditions free from sensation of draft, while with higher ambient temperatures even higher air velocities may be used. In this connection it may be emphasized that drafts are interpreted[54] as local sensations of excessive coolness, and that even while very high air movement in relatively warm air increases the rate of heat loss from local parts of the body, it may improve the comfort of the occupant, so long as that part of his body surface is not excessively cooled.

THE FOUR VITAL FACTORS

From the preceding discussion it is clear that thermal environment cannot properly be adjusted to the requirements of human health and comfort without control of all the four basic factors:

1. Air temperature (free from radiation effects).
2. Air movement.
3. Humidity.
4. Mean radiant temperature of surrounding surfaces.

According to the recommendations of the Sub-Committee on the Hygiene of Environmental Conditions in the Dwelling[55], it is of great importance in all research studies to make an accurate record of each of the four independent factors governing bodily heat exchanges, temperature, movement and humidity of the air, and mean radiant temperature of the surrounding surfaces. For this purpose the committee suggested in the interest of comparability the use of the following four types of instruments or others yielding similar data:

1. Silvered dry-bulb thermometers or hair-pin thermometers (Bargeboer).
2. Silvered dry Kata-thermometers or the hot wire anemometer.
3. Psychrometer, wet- and dry-bulb, whirling or ventilated.
4. Globe thermometer (Vernon) or the dry resultant thermometer (Missenard).

In this country, the shielded thermometer, or a very fine wire thermocouple have been found more convenient for determining the true dry-bulb temperature. Fine, hot-wire anemometers are rapidly replacing the use of the Kata thermometer for measuring low air velocities, while some

[53]The Influence of Air Movement on Heat Losses for the Clothed Human Body, by C.-E. A. Winslow, A. P. Gagge and L. P. Herrington (*American Journal of Physiology*, October, 1939, Vol. 127, p. 505).

[54]A.S.H.V.E. Research Report No. 1086—Draft Temperatures and Velocities in Relation to Skin Temperature and Feeling of Warmth, by F. C. Houghten, Carl Gutberlet and Edward Witkowski (A.S.H.V.E. Transactions, Vol. 44, 1938, p. 289).

[55]Housing Commission of the League of Nations, adopted at Geneva, June 25, 1937.

adaptations of the Vernon Globe thermometer, incorporating thermocouples rather than mercury thermometers, have been found more satisfactory and convenient.

Such instruments as these, when properly calibrated and their readings are compared, can be used for determining the four basic physical factors concerned separately or in certain combinations. The results of the four physical measurements thus determined can generally be translated into the terms of any special instrument combining two or more of them.

The work of the A.S.H.V.E. Research Laboratory has made available psychrometric charts with effective temperature scale superimposed thereon, including Figs. 8 and 9, and others[56], while recent studies[57] have indicated the degree to which mean radiant temperature of the surroundings modify the effective temperature index.

In some instances it may be important to record not only the movement and temperature of the air at various levels, but also the temperature of each wall and window, of the flooring, and of the ceiling, and to measure the total effective radiation of the surroundings in 6 directions; in order to trace the exact causes of defects in the building which have an unfavorable influence on the heat exchanges of its inhabitants. Facts of this type are of great practical importance.

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[56]A.S.H.V.E. RESEARCH REPORT NO. 691—Cooling Effects on Human Beings Produced by Various Air Velocities, by F. C. Houghten and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 193).

[57]Loc. Cit. Note 31

Chapter 3

FUNDAMENTALS OF HEAT TRANSFER

Thermal Conduction, Thermal Convection, Thermal Radiation, Solutions for Steady-State Conduction Problems, Unit Conductances for Convection Flow Systems, Angle and Emissivity Factors for Radiant Heat Transfer Systems

HEAT is that form of energy which is transferred from place to place by virtue of an existing temperature difference. The temperature difference is the potential which causes the transfer, the latter in turn being resisted by the thermal properties of the material combined in a simple term and known as the resistance. Energy exchange associated with mass transfer from place to place (evaporation, condensation, etc.) due to concentration differences will be treated elsewhere such as the section on *cooling tower design* in Chapter 27. The objectives of this chapter are to:

1. Describe the mechanisms and present the rate equations for the different modes of heat transfer.
2. Illustrate the application of the basic concepts to steady-state problems (temperature independent of time or a cyclic variable thereof) by means of several typical solutions of heat transfer systems.

Further applications to specific systems will be found throughout the Guide.

CONDUCTION, CONVECTION AND RADIATION

Thermal conduction is the term applied to the mechanism of heat transfer whereby in fluids the molecules of higher random kinetic energies transmit by direct molecular collision part of their energy to adjacent molecules of lower random kinetic energy. Since the temperature is proportional to the random kinetic energy of the molecules, thermal transfer will occur in the direction of decreasing temperature. The molecules oscillate about a mean position at fairly high velocities and frequencies, but there is no net material flow associated with the conduction mechanism.

In solids the significant mechanism of heat transport is thermal conduction and is ascribed to a transfer mechanism associated with the free electrons[1]. Even in the case of fluids, thermal conduction is significant in the region very close to a solid boundary, for in this region the fluid

[1]The Metallic State, by H. Hume-Rothery (Oxford Press, 1931).

flows laminarly with practically no cross flow (*i.e.*, in the direction of heat transfer). Finally, in fluid systems in which the fluid is moved only by gravity (free convection) and the hot fluid exists at the top, transfer of thermal energy will occur only by conduction (an example is the room with the hot air at the ceiling in the absence of forced circulation).

Contrasted to the thermal conduction mechanism, *thermal convection* involves energy transfer by eddy mixing and diffusion[2] in addition to conduction. This condition is pictured schematically in Fig. 1 which exhibits transfer from a pipe wall at surface temperature t_s to a colder fluid at a bulk temperature t_f. In the *laminar* sublayer, immediately adjacent to the wall, the heat transfer is by thermal conduction, in the *transition* region, which is called the buffer layer, eddy mixing as well as conduction effects are significant, while in the *eddy* or *turbulent* region the major fraction of the transfer is by eddy mixing.

In the case of laminar (also called streamlined) flow there is no transition or eddy region. However, in most commercial equipment, except that involving low velocity flows of viscous liquids, the flow is turbulent.

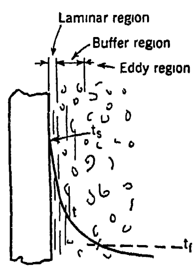


FIG. 1. THERMAL CONVECTION CONDITIONS

When the fluid currents are induced by sources external to the heat transfer region, as for example a pump, the described solid to fluid heat transfer is termed *forced convection*. In contrast, if the fluid currents are internally generated, as a result of non-homogeneous densities arising from the temperature variations, the heat transfer is termed *free convection*.

In the conduction and convection mechanisms heat is transferred as internal energy, *i.e.*, the random molecular kinetic energy associated with the material temperature. For radiant heat transfer, however, a change in energy form takes place from internal energy at the source to electromagnetic energy for transmission, then back to internal energy at the receiver. Since visible radiant energy exhibits characteristic wave lengths, the solution of thermal radiation problems is in many respects similar to the solution of problems in the field of illumination.

The rate of thermal current flow (*i.e.*, rate of heat transfer) corresponding to the transfer mechanisms previously described, may be expressed by the following rate equations. These are similar to Ohm's

[2]Absorption and Extraction, by T. K. Sherwood (McGraw-Hill Co., 1937).

Law for electrical flow, the current flow through a resistance being proportional to the potential difference.

Thermal Conduction Equation

$$\frac{dq}{dA} = -k\frac{dt}{dx} \tag{1}$$

This expression states symbolically that the thermal conduction current per unit transfer area normal to the flow, $(dq)/(dA)$, Btu per hour per square foot, is proportional to the temperature gradient $(dt)/(dx)$, degree Fahrenheit per foot. The proportionality factor is termed the *thermal conductivity*, k, Btu per hour per square foot per degree Fahrenheit per foot of thickness.

It should be emphasized that the thermal conductivity used should be expressed in consistent units; either using the inch or foot throughout.

Expressions of conductivity used in the heating field are usually inconsistent in this sense, in that it is customary to refer to the conductivity *per square foot* but for one inch of thickness. This custom has

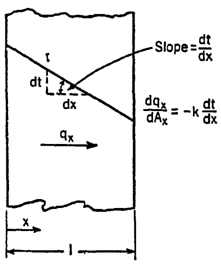


FIG. 2. THERMAL CONDUCTION IN A FLAT SLAB

been adopted for the reason that wall thicknesses are usually expressed in inches, whereas if expressed in feet, decimal or fractional thicknesses would result. When dealing with flat walls no complication is involved in using the inconsistent expression of conductivity. However, when curved or spherical walls are considered, considerable complication is involved. Therefore, in this discussion the consistent units of conductivity expressed in Btu per hour per square foot per degree Fahrenheit for one foot of thickness is used throughout. Conductivity values obtained from Chapter 4 or Table 1 in this chapter, which are expressed in inconsistent units, must therefore be converted for use in the calculations of this chapter, by dividing by 12. As an example, the conductivity of brick, expressed in inconsistent units as 5.0 in Table 2 of Chapter 4, becomes 0.42 when used in the calculations of this chapter. Also, it should be emphasized that in order to make the calculations and applications consistent in this chapter, all dimensions of thickness must be expressed in feet.

The minus sign on the right side of the equation is introduced to indicate positive current flow in the direction of decreasing temperature. The physical significance of indicated quantities are illustrated further by the schematic diagram Fig. 2.

TABLE 1. APPROXIMATE UNIT THERMAL CONDUCTIVITIES OF MISCELLANEOUS MATERIALS[a]

MATERIAL	CONDUCTIVITY, k BTU PER HOUR PER SQ FT PER DEG F FOR ONE INCH THICKNESS
Air	0.168
Aluminum	1416.0
Brass (70 − 30)	720.0
Cast Iron	336.0
Copper	2640.0
Glass	3.6—7.32
Lead	240.0
Nickel	330.0
Soil	2.4—12.0
Steel, mild	312.0
Water, liquid	4.08

[a]Thermal conductivities depend to some extent on temperature. The above magnitudes are approximate only. Refer to Heat Transmission, by W. H. McAdams (McGraw-Hill Co., 1937) for additional values.

Thermal Convection Equation

$$\frac{dq}{dA} = h_c\,(t_s - t_f) \tag{2}$$

This rate equation states that the thermal convection current per unit transfer area $(dq)/(dA)$, Btu per hour per square foot, is proportional to the temperature difference, $(t_s - t_f)$ which is the temperature of the surface less that of the fluid[3]. The proportionality factor is termed the *unit convection conductance* (sometimes called the film coefficient), h_c, Btu per hour per square foot per degree Fahrenheit. These convection conditions are illustrated in Fig. 1.

The heat transmission by free or natural convection can be conveniently expressed as in Equation 2a:

$$q_c = C\left(\frac{1}{D}\right)^{0.2}\left(\frac{1}{T\,\text{av.}}\right)^{0.181} dt^{1.266} \tag{2a}$$

where

q_c = heat transmission by convection, Btu per square foot per hour.
C = a constant depending upon the surface shape.
D = diameter of pipe or circular duct or height of vertical wall, inches. (Effect of diameter or height becomes constant at 24 in.)
T av. = average wall surface and surrounding air temperature, degrees Fahrenheit absolute.
dt = temperature excess between wall surface and surrounding air, degrees Fahrenheit.

For horizontal cylinders, the value of C = 1.016 has been well established by various investigations. For vertical plates, the value of C = 1.394 has been fairly well established. A value of C = 1.79 for horizontal plates warmer than the surrounding air facing upward and 0.89 for horizontal plates warmer than air facing downward is indicated by recent investigations[4].

[3]The particular fluid temperature to use for a given system will be noted under the discussion of that system.

[4]The Transmission of Heat by Radiation and Convection, by Griffith and Davis (*Special Report* No. 9, 1922, *Department of Scientific and Industrial Research*, His Majesty's Stationery Office, London, England).

The heat transmission by free convection from vertical walls 24 in. or more in height is given in Table 2 as calculated from Equation 2a for ambient air temperature of 80 F. The values in Table 2 will not be changed appreciably by a considerable change in air temperature for a given temperature excess. For instance, a change in air temperature from 80 to 40 F will increase the heat transmission given in Table 2 by only 1.3 per cent.

Table 2 can also be used for calculating the free convection rate of transmission for various commercial shapes such as pipes and ducts. These calculations are simplified by the use of the factors in Tables 3 and 4. Table 3 gives factors by which the values in Table 2 must be multiplied to obtain the free convective transfer from various shapes whose characteristic dimensions are 24 in. or over, and Table 4 gives the

TABLE 2. HEAT TRANSMISSION BY FREE CONVECTION FOR LARGE VERTICAL SURFACES

Expressed in Btu per square foot per hour

Temp. Deg F	Temperature Difference between Body and Surrounding Still Air at 80 F													
	0	10	20	30	40	50	60	70	80	90	100	110	120	130
0	0	4.4	10.4	17.4	25.0	33.2	41.8	50.6	59.9	69.4	79.4	89.2	99.4	109.8
1	0.3	4.9	11.1	18.1	25.8	34.1	42.6	51.5	60.8	70.3	80.4	90.2	100.4	110.9
2	0.6	5.5	11.8	18.9	26.7	34.9	43.5	52.4	61.8	71.3	81.4	91.2	101.5	112.0
3	1.0	6.0	12.5	19.7	27.5	35.7	44.3	53.4	62.7	72.3	82.4	92.2	102.6	113.0
4	1.4	6.6	13.2	20.5	28.3	36.6	45.2	54.3	63.7	73.3	83.3	93.3	103.6	114.1
5	1.8	7.3	13.9	21.2	29.2	37.4	46.1	55.2	64.6	74.3	84.2	94.3	104.7	115.2
6	2.3	7.9	14.6	22.0	30.0	38.3	47.0	56.1	65.6	75.3	85.2	95.3	105.7	116.3
7	2.8	8.5	15.3	22.7	30.8	39.1	47.8	57.1	66.5	76.3	86.2	96.3	106.7	117.3
8	3.3	9.1	16.0	23.5	31.6	40.0	48.7	58.0	67.5	77.4	87.2	97.4	107.8	118.4
9	3.8	9.7	16.7	24.3	32.4	40.9	49.7	59.0	68.4	78.4	88.2	98.4	108.8	119.5

TABLE 3. FREE CONVECTION FACTORS FOR VARIOUS SHAPES

Shapes	Factor
Horizontal cylinders 24 in. in diam. or over	0.73
Long vertical cylinders 24 in. in diam. or over	0.88
Vertical plates 24 in. in height or over	1.00
Horizontal plates warmer than air facing upward	1.28
Horizontal plates warmer than air facing downward	0.64
Horizontal plates cooler than air facing upward	0.64
Horizontal plates cooler than air facing downward	1.28

TABLE 4. FREE CONVECTION FACTORS FOR VARIOUS DIAMETER PIPES OR VARIOUS HEIGHT PLATES

Actual o. d., or height, in.	1	2	3	4	5	6	7	8
Factor	1.88	1.64	1.52	1.43	1.37	1.32	1.28	1.25
Actual o. d., or height, in.	9	10	12	14	16	18	20	22
Factor	1.22	1.19	1.15	1.11	1.09	1.06	1.04	1.02

factors to be used in conjunction with the factors in Table 3 for obtaining the free convection from Table 2 for pipes and ducts whose characteristic dimensions are less than 24 in.

For example, the free convection transfer from a 3 in. o.d. horizontal cylinder for a temperature difference for 40 F = 25.0 × 0.73 × 1.52 = 27.7 Btu per square foot per hour.

The increased rate of heat transfer due to forced convection can be calculated from Equation 2b:

$$q_{fc} = 1 + 0.225V \tag{2b}$$

where

q_{fc} = heat transfer by forced convection, Btu per square foot per hour per degree Fahrenheit temperature difference.

V = velocity of air, feet per second.

This equation is approximately correct for large surfaces exposed to air currents at temperatures of approximately 70 to 80 F.

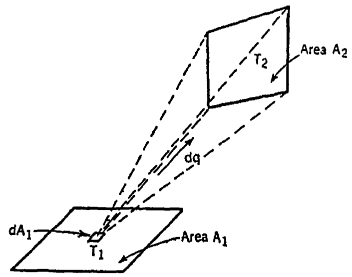


FIG. 3. RADIATION BETWEEN SURFACES

Thermal Radiation Equation

$$\frac{dq}{dA_1} = \sigma F_A F_E \left(T_1^4 - T_2^4\right) \tag{3}$$

This relation, which is sometimes applicable to systems in which radiant exchange takes between the surfaces of solids, states that the net radiation current per unit transfer area of surface 1, $(dq)/(dA_1)$, Btu per hour per square foot; which *sees* surface 2 through a non-absorbing medium, is proportional to the difference of the fourth powers of the absolute surface temperatures $(T_1^4 - T_2^4)$. The proportionality factor $(\sigma\ F_A F_E)$ may be conveniently separated into three parts:

σ = the Stefan-Boltzmann radiation constant.

= 1730×10^{-12} Btu per hour per square foot per degree Fahrenheit absolute temperature to the fourth power.

F_A = the angle factor is dimensionless and $\leqq 1$. This factor accounts for the relative geometry of the two surfaces, and is called the shape factor.

F_E = the emissivity factor is also dimensionless and $\leqq 1$. This factor accounts for the absorption and emission characteristics of the surfaces for the radiation which exists.

The radiation conditions described are schematically shown in Fig. 3. Gaseous and luminous radiation will not be discussed here.

TABLE 5. HEAT TRANSMISSION BY RADIATION FOR BLACK-BODY CONDITIONS[a]

Expressed in Btu per square foot per hour

TEMP. DEG F	0	−1	−2	−3	−4	−5	−6	−7	−8	−9
−30	59.3	58.7	58.2	57.7	57.2	56.7	56.2	55.7	55.2	54.7
−20	65.2	64.7	64.1	63.5	62.9	62.3	61.7	61.1	60.5	59.9
−10	71.4	70.8	70.1	69.5	68.9	68.3	67.7	67.1	66.4	65.8
0	78.0	77.4	76.7	76.0	75.4	74.7	74.0	73.4	72.7	72.1
	0	+1	+2	+3	+4	+5	+6	+7	+8	+9
0	78.0	78.7	79.4	80.1	80.8	81.5	82.2	82.9	83.6	84.3
10	85.0	85.7	86.5	97.2	88.0	88.7	89.4	90.2	90.9	91.7
20	92.4	93.3	94.0	94.8	95.6	96.4	97.2	98.0	98.8	99.6
30	100	101	102	103	104	105	105	106	107	108
40	109	110	111	112	112	113	114	115	116	117
50	118	119	120	121	122	123	123	124	125	126
60	127	128	129	130	131	132	133	134	135	136
70	137	138	139	140	142	143	144	145	146	147
80	148	149	150	151	152	153	154	155	156	157
90	159	160	161	162	163	164	166	167	168	169
100	170	171	173	174	175	176	178	179	180	182
110	183	184	185	187	188	189	191	192	193	195
120	196	197	199	200	201	203	204	206	207	209
130	211	212	214	215	217	218	220	221	222	224

[a]*Example:* Radiation from walls of room at 32 F to surface at − 25 F for effective emissivity of 0.95 = (102 − 62 3) 0.95 = 37.7 Btu per square foot per hour.

The radiation under black-body conditions, or for an emissivity of 1.0, is given in Table 5 for cold surfaces as low as − 39 F to warmer surfaces as high as 139 F. The emissivities of a number of surfaces ordinarily encountered in engineering practice are shown in Table 9[5].

In many problems involving heat transfer by the mechanisms of radiation and convection in parallel (simultaneous heat transfer by the two mechanisms to the same surface), it is algebraically convenient to employ a first power radiation rate equation of a form similar to Equation 2 for convection, namely:

$$\frac{dq}{dA_1} = h_r\,(t_1 - t_2) \tag{3a}$$

where h_r is the unit radiation conductance. Comparison of this equation with Equation 3 yields the relation:

$$h_r = \sigma\, F_A F_E \frac{(T_1^4 - T_2^4)}{(T_1 - T_2)} \approx \sigma\, F_A F_E\,(4\,T_{ave}^3) \tag{3b}$$

where

$$T_{ave} = \frac{(T_1 + T_2)}{2}$$

This unit conductance (h_r) is to be used with caution for its behavior is

[5]Heat Insulation in Air Conditioning, by R. H. Heilman (*Industrial and Engineering Chemistry*, Vol. 28, July 1936, p. 782).

quite different from (h_c). Its use is permissible if the temperature difference is small.

The steady-state heat transfer problems encountered in air conditioning applications are not normally of the type involving action of a single mode of heat transfer through a single homogeneous material. Usually more than one of the heat transfer mechanisms is effective, and the thermal current flows through several thermal resistances either in a series or in parallel. This type of problem may best be illustrated by a consideration of a particular system such as the heat transfer from the air to cold water inside an insulated pipe, Fig. 4. Since air is sensibly transparent to radiation, there will be some heat transfer by both radiation and convection to the outer insulation surface. The mechanisms act in parallel on the air side. The total current by radiation and convection then passes through the insulating layer and the pipe wall by thermal conduction, and thence by convection into main cold water streams. Radiation is not

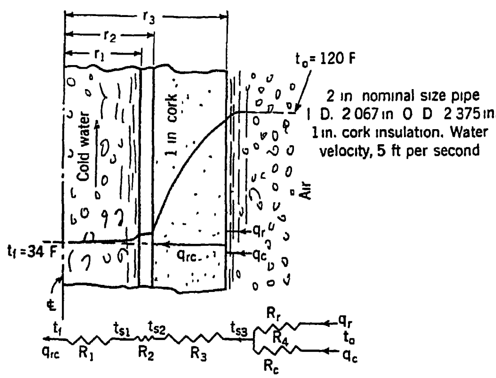


FIG. 4. HEAT TRANSFER CONDITIONS IN THE INSULATED COLD WATER LINE

significant on the water side as liquids are sensibly opaque to radiation, although water transmits energy in the visible region. The contact resistance between the insulation and the pipe wall is presumed to be equal to zero.

Referring to Fig. 4, the thermal current for a given length N of pipe, q_{rc} Btu per hour, may be thought of as flowing through the parallel resistances R_r and R_c, associated with the insulation surface radiation and convection transfer. Then the flow is through the resistance offered to thermal conduction by the insulation, R_3, through the pipe wall resistance, R_2, and into the water stream through the convection resistance, R_1. Note the analogy to the direct current electrical circuit problem. A temperature (potential) drop is required to overcome these resistances to the flow of thermal current. The total resistance to heat transfer, R_T, hour degrees Fahrenheit per Btu, is the summation of the individual resistances:

$$R_T = R_1 + R_2 + R_3 + R_4 \tag{4}$$

where the resultant parallel resistance R_4 is obtained from:

$$\frac{1}{R_4} = \frac{1}{R_r} + \frac{1}{R_c}$$

Provided the individual resistances may be evaluated, the total resistance can be obtained from the above relation. Then the heat transfer current for the length of pipe (N, ft) can be established by the relation:

$$q_{rc} \text{ (Btu per hour)} = \frac{t_o - t_f}{R_T} \tag{5}$$

For a unit length of the pipe the heat transfer rate is:

$$\frac{q_{rc}}{N} \text{ (Btu per hour foot)} = \frac{(t_o - t_f)}{R_T N} \tag{5a}$$

The temperature drop, Δt, through an individual resistance may then be calculated from the relation:

$$\Delta t = R\, q_{rc}$$

where R is the resistance in question.

The problem is now reduced to one of evaluating the individual resistances of the system. This entails suitable integration of the rate Equations 1, 2, 3, or 3a to produce expressions of the form:

$$q = \frac{\Delta t}{R} \tag{6}$$

where q is the heat transfer rate, and Δt is the potential drop or temperature difference through the resistance R. Table 6 lists such solutions for six different conduction systems. Table 2 in Chapter 4 and Table 1 of this chapter indicate the magnitudes of the thermal conductivities, k, to be employed in the expressions of Table 6.

The solution applicable to the problem depicted in Fig. 4, for the calculation of R_2 and R_3, is case 2 in Table 6. Thus for a *1 ft length* of 2 in. nominal size pipe (I. D. = 2.067 in., O. D. = 2.375 in.) insulated with 1 in. of cork:

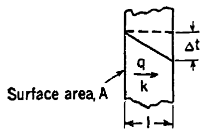
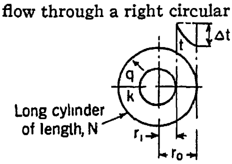
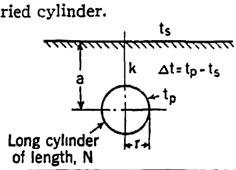
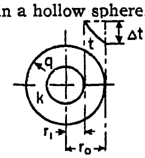
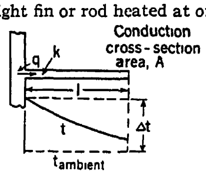
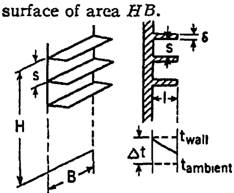
$$R_2 = \frac{\log_e \dfrac{1.188}{1.033}}{2\pi \times 26 \times 1} = 8.5 \times 10^{-4} \text{ hr degree Fahrenheit per Btu.}$$

$$R_3 = \frac{\log_e \dfrac{2.188}{1.188}}{2\pi \times 0.025 \times 1} = 3.9 \text{ hr degree Fahrenheit per Btu.}$$

The convection resistances to heat transfer from the pipe wall to the cold water, R_1, and from the air to the surface of the insulating material, R_c, are dependent on the flow conditions prevailing at these surfaces, and on the thermal properties of the fluids. The unit conductances for thermal convection, h, Btu per hour per square foot per degree Fahrenheit, have been determined by test for many flow systems. These data may be employed to predict the conductances for *similar flow systems*. Table 7 summarizes some empirical equations expressing such test results.

For the problem under consideration (Fig. 4) case 3 of Table 7 is applicable for the calculation of the cold water side convection resistance R_1. Corresponding to the water velocity of 5 ft per second, the mass velocity is:

TABLE 6. SOLUTIONS FOR SOME STEADY-STATE THERMAL CONDUCTION PROBLEMS[a][b]

No.	SYSTEM	Expressions for the resistance R entering into the equation: $q = \Delta t/R$ (Btu per hour)
1.	Flat wall or curved wall if curvature is small (wall thickness less than 0.1 of inside diameter). 	$R = \dfrac{l}{kA}$
2.	Radial flow through a right circular cylinder. 	$R = \dfrac{\log_e \dfrac{r_0}{r_1}}{2\pi k N}$ (See footnote c).
3.	The buried cylinder. 	$R \approx \dfrac{\log_e \dfrac{2a}{r}}{2\pi k N}$; $R = \dfrac{\cosh^{-1} \dfrac{a}{r}}{2\pi k N}$ for $\dfrac{a}{r} \geqq 3$ (See footnote c).
4.	Radial flow in a hollow sphere. 	$R = \dfrac{\dfrac{1}{r_1} - \dfrac{1}{r_0}}{4\pi k}$
5.	The straight fin or rod heated at one end. 	$R = \dfrac{m}{h_s p \tanh ml}$ (see footnotes d and e). For $ml > 2.3$, $\tanh m\,l \approx 1$ $m = \sqrt{h_s p/kA}$ A = conduction cross section area. p = perimeter of cross section A. h_s = unit conductance to the surroundings from the fin surface. k = thermal conductivity fin material. Δt = wall temperature—ambient temp.
6.	Finned surface of area HB. 	$R = \dfrac{(s + \delta)}{h_s \left(\dfrac{2}{m} \tanh m\,l + s\right) HB}$ $m = \sqrt{\dfrac{h_s p}{kA}} = \sqrt{\dfrac{2\,h_s}{k\delta}}$ Δt defined as in Case 5 above.

[a] The dimensions to be employed in these solutions are: length of dimension p, l, r = feet; units of k = Btu per hour per square foot per degree Fahrenheit for one foot thickness; units of h, Btu per hour per square foot per degree Fahrenheit; units of area, A = square feet.

[b] The thermal conductivity, k, in these solutions should be taken at the average material temperature (see Table 7).

[c] $\log_e x = 2.303 \log_{10} x$.

[d] This expression can also be employed as an approximation for tapered fins or of annular fins by employing average magnitudes of A and p.

[e] Tanh is the hyperbolic tangent.

G = 5 (ft per sec) × 62.4 (lbs per cu ft) × 3600 (sec per hr) = 11.2×10^5 lb per hour per square foot.

The hydraulic radius for the flow in the 2 in. line is, by definition:

$$r_H = \frac{\pi D^2}{\pi D 4} = \frac{D}{4} = \frac{2.067}{12 \times 4} = \frac{0.1725}{4} \text{ ft.}$$

The average water film temperature will be estimated as 36 F (mixed mean fluid temperature of 34 F). Then case 3, Table 7 yields:

$$h = 0.00486\ (1 + 0.36)\ \frac{(11.2 \times 10^5)^{0.8}}{(0.1725)^{0.2}} = 650 \text{ Btu per hour per square foot per degree Fahrenheit.}$$

The transfer area on which this conductance is based is the inside tube area. Associated with 1 ft length of pipe there are:

$$\pi \times \frac{2.067}{12} \times 1 = 0.542 \text{ sq ft.}$$

Thus the resistance for 1 ft of tube length is:

$$R_1 = \frac{1}{h\pi D \times 1} = \frac{1}{650 \times 0.542} = 2.8 \times 10^{-3} \text{ hr degree Fahrenheit per Btu.}$$

Case 9, Table 7 is applicable for calculating the free thermal convection resistance, R_c, existing between the surrounding air and the insulation. The air temperature is given as 120 F. As an approximation a 20 F temperature difference between the air and the pipe surface will be assumed. Then case 9 yields:

$$D = \frac{4.375}{12} = 0.364 \text{ ft.}$$

$$h = 0.23 \left(\frac{20}{0.364}\right)^{0.25} = 0.63 \text{ Btu per hour per square foot per degree Fahrenheit.} \quad (7)$$

This result may not be deemed conservative inasmuch as the expression is for *still* air. If, however, the air is not still, but flows at approximately 5 mph or 7 ft per second the mass velocity corresponds to:

$$G = 7 \times 0.07 \times 3600 = 1770 \text{ lb air per hour per square foot.}$$

A magnitude of k = 0.014 Btu per hour per square foot per degree Fahrenheit for one foot thickness applied to Case 4 yields:

$$h = 0.45 \left(\frac{0.014}{0.364}\right) + 0.178\ (1770)^{0.56} \left(\frac{0.014}{0.364}\right)^{0.44}$$

$$= 0.017 + 2.8 = 2.8 \text{ Btu per hour per square foot per degree Fahrenheit.}$$

This conductance is based on 1 sq ft of outside lagging area. Thus, since there are $\pi \times \frac{4.375}{12} = 1.14$ sq ft of outside lagging area associated with 1 ft length of pipe:

$$R_c = \frac{1}{2.8 \times 1.14} = 0.312 \text{ hr degree Fahrenheit per Btu.}$$

The radiation resistance, R_r, which acts in parallel with the convection resistance, R_c, for the transfer of heat to the surface of the insulation, may

TABLE 7. APPROXIMATE UNIT CONDUCTANCES FOR THERMAL CONVECTION FOR SEVERAL FLOW SYSTEMS[a]

Expressed in Convenient Empirical Form

CASE	SYSTEM	UNIT CONDUCTANCE EQUATION[b]
	FORCED CONVECTION	
1.	Longitudinal flow in cylinders, turbulent region. Fluid being heated[c].	$\frac{4hr_H}{k} = 0.0225 \left(\frac{4r_H G}{\mu}\right)^{0.8} \left(\frac{c_p \mu}{k}\right)^{0.4}$ For $\left(\frac{4r_H G}{\mu}\right) > 3000$
2.	For longitudinal air flow in cylinders case 1 reduces to[c].	$h = 0.0036\, G^{0.8}/(4r_H)^{0.2}$ For $\left(\frac{4r_H G}{\mu}\right) > 3000$
3.	For longitudinal water flow in cylinders case 1 reduces to[c].	$h = 0.00486\,(1 + 0.01t)\,\frac{G^{0.8}}{4r_H^{\,0.2}}$ For $\left(\frac{4r_H G}{\mu}\right) > 3000$
4.	Air flow normal to a single right circular cylinder.	$h = 0.45 \left(\frac{k}{D}\right) + 0.178\, G^{0.56} \left(\frac{k}{D}\right)^{0.44}$
5.	Air flow over staggered pipe banks.	$h = 0.061 \left(\frac{k}{D}\right)^{0.31} G^{0.69}$
6.	Air flow over single spheres.	$h = 0.040\, \frac{G^{0.52}}{D^{0.48}}$ $0 < t < 250$ F
7.	Air flow over plane surfaces.	$h = 1 + 0.22\, V_s$ For $V_s < 16$ ft per second or $h = 0.53\, V_s^{0.8}$ 16 ft per second $< V_s <$ 100 ft per second
8.	Air flow normal to finned cylinders.	$h = 6.2 \left(\frac{G}{3600}\right)^{0.8} \frac{s^{0.32}}{D^{0.53}}$ $0 < t < 250$ F
	FREE CONVECTION[d]	
9.	Single horizontal right circular cylinder in air.	$h = 0.23 \left(\frac{\Delta t}{D}\right)^{0.25}$
10.	Single vertical right circular cylinder in air.	$h = 0.22 \left(\frac{\Delta t}{D}\right)^{0.25}$
11.	Vertical surfaces in air.	$h = 0.3\, (\Delta t)^{0.25}$
12.	Top surface of horizontal plates to air.	$h = 0.4\, (\Delta t)^{0.25}$
13.	Bottom surface of horizontal plates to air.	$h = 0.2\, (\Delta t)^{0.25}$

[a]Heat Transmission, by W. H. McAdams.

[b]Fluid properties should be evaluated at the arithmetic mean fluid temperature, $t_f = (t_{surface} + t_{fluid})$ divided by 2.

[c]These expressions are applicable to longitudinal flow in other than right circular cylinders provided the hydraulic radius is employed as the conduit dimension parameter. For right circular cylinders $4r_H = D$.

[d]For low rates of heat transfer by free convection the exponent decreases towards zero, and for higher rates increases towards 0.33. The following equations employing an exponent equal to 0.25 are applicable in the intermediate range.

NOMENCLATURE AND DIMENSIONS FOR TABLE 7

c_p = fluid unit heat capacity at constant pressure, Btu per pound per degree Fahrenheit.

D = cylinder diameter, feet.

G = $V_s\gamma$ = fluid mass velocity, pounds per hour per square foot of flow cross section.

γ = density, pounds per cubic foot.

h = unit conductance for thermal convection, Btu per hour per square foot per degree Fahrenheit.

k = unit thermal conductivity of the fluid, Btu per hour per square foot per degree Fahrenheit for one foot thickness.

r_H = hydraulic radius of the flow cross section[c].

= flow cross section area per wetted perimeter, feet.

s = fin spacing, feet.

t = average fluid film temperature, degree Fahrenheit.

Δt = temperature difference surface to main fluid, degree Fahrenheit.

V_s = fluid velocity, foot per second.

μ = fluid viscosity, pounds per hour per foot.
viscosity in centipoises $\times$ 2.42 = viscosity in pounds per hour per foot.

be calculated by means of Equation 3b. For the purposes of this illustrative problem it will be assumed that the insulated pipe is exposed to (*sees*) surroundings, which exist at 120 F. For this condition case 2, Table 8 is applicable. Then the angle factor, F_A, is unity and for an estimated surface emissivity of 0.9 (see Table 9), $F_e = 0.9$. As a first approximation the insulation surface temperature will be estimated as 20 F lower than the surroundings at 120 F. Thus t_{ave} = 110 F and T_{ave} = (460 + 110) degree absolute temperature. (Note that even if the assumed temperature difference is in error by 10 F this fact will only affect T_{ave} by 2 per cent. Then by Equation 3b:

$h_r = 1730 \times 10^{-12} \times 1 \times 0.9 \times 4\,(460 + 110)^3 = 1.15$ Btu per hour per square foot per degree Fahrenheit.

The outside surface area of the insulation associated with 1 ft of pipe length was previously calculated as 1.14 sq ft. Thus:

$$R_r = \frac{1}{1.15 \times 1.14} = 0.76 \text{ hr degree Fahrenheit per Btu.}$$

TABLE 8. ANGLE FACTORS, F_A, AND EMISSIVITY FACTORS, F_E, FOR TWO RADIANT HEAT TRANSFER SYSTEMS[a]

CASE	SYSTEM	F_A	F_E
1.	Large parallel planes	1	$\frac{1}{e_1} + \frac{1}{e_2} - 1$
2.	Completely enclosed body small compared to the enclosing body. Enclosed body of area A_1.	1	e_1

[a]Solutions for other systems are available in Heat Transmission, by W. H. McAdams. e denotes the surface emissivity, see Table 9.

TABLE 9. SOME LOW TEMPERATURE RADIATION EMISSIVITIES FOR COMMERCIAL SURFACES[a]

Surface	Emissivity e
Metal Surfaces	
Aluminum:	
Clean plate	0.04–0.06
Strongly oxidized	0.11–0.19
Brass:	
Polished	0.096
Dull finish	0.22
Strongly oxidized	0.6
Copper:	
Polished	0.023
Clean	0.07
Strongly oxidized	0.8
Cast Iron, rough, strongly oxidized	0.95
Sheet Iron	0.87
Steel Plate, rough	0.94—0.97
Wrought Iron, dull oxidized	0.94
Zinc, Galvanized Sheet Iron:	
Fairly bright	0.228
Gray, oxidized	0.276
Painted Surfaces	
Aluminum paint	0.3–0.7
Other pigment paints	0.8–0.95
Miscellaneous	
Asbestos board	0.96
Asbestos paper	0.93—0.945
Carbon, candle soot	0.952
Glass	0.80
Glazed brick	0.75
Lamp black and soot	0.90—0.97
Linoleum	0.85
Marble, light gray polished	0.931
Paper	0.90—0.95
Plaster	0.95
Porcelain, glazed	0.924
Roofing paper	0.91
Rough brick	0.80—0.93
Rough concrete	0.95
Rough mortar	0.91
Smooth concrete	0.90
Stone	0.87
Water	0.95
Wood unpolished	0.89

[a]These emissivities should not be employed for solar radiation calculations. For this type of calculation reference should be made to Heat Transmission, by W. H. McAdams and The Calculation of Heat Transmission, by Margaret Fishenden and Owen A. Saunders.

The resultant resistance of R_c and R_r acting in parallel (see Fig. 4) can now be evaluated as:

$$\frac{1}{R_4} = \frac{1}{R_c} + \frac{1}{R_r} = \frac{1}{0.312} + \frac{1}{0.76} = 4.51 \text{ Btu per hour per degree Fahrenheit.}$$

$$R_4 = 0.222 \text{ hr degree Fahrenheit per Btu.}$$

The individual resistances for a 1 ft length of pipe applying to the illustrative problem depicted in Fig. 4 have now been calculated and are summarized as follows:

R_1 convection from the pipe wall to the cold water $= 2.8 \times 10^{-3}$ hr degree Fahrenheit per Btu.

R_2 conduction through the pipe wall $= 8.5 \times 10^{-4}$ hr degree Fahrenheit per Btu.

R_3 conduction through the cork insulation $= 3.9$ hr degree Fahrenheit per Btu.

R_4 parallel convection and radiation from the surroundings $= 0.22$ hr per degree Fahrenheit per Btu.

Then

R_T the overall resistance surroundings to cold water $= R_1 + R_2 + R_3 + R_4 = 4.1$ hr degree Fahrenheit per Btu.

Note that the controlling resistances are R_3 and R_4. That is, the neglect of R_1 and R_2 would not significantly influence the total resistance, R_T.

On the basis of this resistance calculation the heat transfer from the surroundings to the cold water may be evaluated as:

$$\frac{q_{rc}}{N} = \frac{\Delta t}{R_T} = \frac{120 - 34}{4.1} = 21 \text{ Btu per hour per foot.}$$

or about 0.175 tons of refrigeration per 100 ft of pipe.

Since the calculation is based on a 1 ft pipe length:

$$q_{rc} = 21 \text{ Btu per hour.}$$

The temperature drops through the various resistances are now readily evaluated by Equation 6 as:

Δt air to insulation surface $= R_4\, q_{rc} = 0.22 \times 21 = 4.6$ F.

Δt through the insulation $= R_3\, q_{rc} = 3.9 \times 21 = 82$ F.

Δt through the pipe wall $= R_2\, q_{rc} = 8.5 \times 10^{-4} \times 21 = 0.02$ F.

Δt pipe wall to cold water $= R_1\, q_{rc} = 2.8 \times 10^{-3} \times 21 = 0.06$ F.

The solution was obtained on the assumption that the air temperature and the outside temperature differed by 20 F. In order to obtain a slightly better estimate of the rate of heat transfer the numerical solution should be repeated using the temperatures calculated from the previous listed temperature differences.

The foregoing problem serves to illustrate a general method of solving steady-state heat transfer problems. There are many problems which cannot be approximated by steady-state solutions. For instance, the problem of pipe line insulation in transient service; the behavior of automatically controlled thermoflow circuits; or the periodic absorption of solar energy by roof and wall structures during the day and nocturnal radiation to the *cold* sky at night. The transient heat transfer problem differs from the steady-state in that energy storage rates need to be

considered. Thus thermal capacity in addition to resistance effects are significant. The vector sum of the thermal capacitance and resistance is the thermal impedance. It is not within the scope of this chapter to deal with many of these problems. These are, however, solutions available in graphical form for certain special cases. Also a general approximate method may be employed which is analogous to the treatment of capacity-resistance lumped parameter electrical circuits.

REFERENCES

Heat and Thermodynamics, by Mark W. Zemansky (McGraw-Hill Co., 1937, p. 48)

Heat Transmission, by W. H. McAdams (McGraw-Hill Co., 1937).

The Calculation of Heat Transmission, by Margaret Fishenden and Owen A. Saunders (His Majesty's Stationery Office, London).

Industrial Heat Transfer, by A. Schack, translated from the German by H. Goldschmidt and E. P. Partridge (John Wiley and Sons, Inc., 1933).

The Metallic State, by W. Hume-Rothery (Oxford Press, 1931).

Absorption and Extraction, by T. K. Sherwood (McGraw-Hill Co., 1937).

Chapter 4

HEAT TRANSMISSION COEFFICIENTS AND TABLES

Methods of Heat Transfer, Coefficients, Conductivity of Homogeneous Materials, Surface Conductance Coefficients, Air Space Conductance, Practical Coefficients, Table of Conductivities and Conductances, Tables of Over-all Coefficients of Heat Transfer for Typical Building Construction, Combined Coefficients of Transmission

IN order to maintain comfortable living temperatures within a building it is necessary to supply heat at the same rate that it is lost from the building. The loss of heat occurs in two ways, by direct transmission through the various parts of the structure and by air leakage or filtration between the inside and outside of the building. The purpose of this chapter is to show methods of calculation and to give practical transmission coefficients which may be applied to various structures to determine the heat loss by direct transmission. The amount lost by air filtration is determined by different methods, as outlined in Chapter 5, and must be added to that lost by direct transmission to obtain the total heating plant requirements.

METHODS OF HEAT TRANSFER

Heat transmission between the air on the two sides of a structure takes place by three methods, namely, radiation, convection and conduction. In a simple wall built up of two layers of homogeneous materials separated to give an air space between them, heat will be received from the high temperature surface by radiation, convection and conduction. It will then be conducted through the homogeneous interior section by conduction and carried across to the opposite surface of the air space by radiation, conduction and convection. From here it will be carried by conduction through to the outer surface and leave the outer surface by radiation, convection and conduction. The process of heat transfer through a built-up wall section is complicated in theory, but in practice it is simplified by dividing a wall into its component parts and considering the transmission through each part separately. Thus the average wall may be divided into external surfaces, homogeneous materials and interior air spaces. Practical heat transmission coefficients may be derived which will give the total heat transferred by radiation, conduction and convection through any of these component parts and if the selection and method of applying these individual coefficients is thoroughly understood it is usually a comparatively simple matter to calculate the over-all heat transmission coefficient for any combination of materials.

HEAT TRANSFER COEFFICIENTS

The symbols representing the various coefficients of heat transmission and their definitions are as follows:

U = thermal transmittance or over-all coefficient of heat transmission; the amount of heat expressed in Btu transmitted in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 F between the air on the inside and that on the outside of the wall, floor, roof or ceiling.

k = thermal conductivity; the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a homogeneous material 1 in. thick for a difference in temperature of 1 F between the two surfaces of the material. The conductivity of any material depends on the structure of the material and its density. Heavy or dense materials, the weight of which per cubic foot is high, usually transmit more heat than light or less dense materials, the weight of which per cubic foot is low.

C = thermal conductance; the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a non-homogeneous material for the thickness or type under consideration for a difference in temperature of 1 F between the two surfaces of the material. Conductance is usually used to designate the heat transmitted through such heterogeneous materials as plasterboard and hollow clay tile.

f = film or surface conductance; the amount of heat expressed in Btu transmitted by radiation, conduction and convection from a surface to the air surrounding it, or vice versa, in one hour per square foot of the surface for a difference in temperature of 1 F between the surface and the surrounding air. To differentiate between inside and outside wall (or floor, roof or ceiling) surfaces, f_i is used to designate the inside film or surface conductance and f_o the outside film or surface conductance.

a = thermal conductance of an air space; the amount of heat expressed in Btu transmitted by radiation, conduction and convection in one hour through an area of 1 sq ft of an air space for a temperature difference of 1 F. The conductance of an air space depends on the mean absolute temperature, the width, the position and the character of the materials enclosing it.

R = resistance or resistivity which is the reciprocal of transmission, conductance, or conductivity, *i.e.:*

$$\frac{1}{U} = \text{over-all or air-to-air resistance.}$$

$$\frac{1}{k} = \text{internal resistivity.}$$

$$\frac{1}{C} = \text{internal resistance.}$$

$$\frac{1}{f} = \text{film or surface resistance.}$$

$$\frac{1}{a} = \text{air space resistance.}$$

As an example in the application of these coefficients assume a wall with over-all coefficient U. Then,

$$H = A\,U\,(t - t_o) \qquad (1)$$

where

H = Btu per hour transmitted through the material of the wall, glass, roof or floor.

A = area in square feet of wall, glass, roof, floor, or material, taken from building plans or actually measured. (Use the net inside or heated surface dimensions in all cases.)

$t - t_o$ = temperature difference between inside and outside air, in which t must always be taken at the proper level. Note that t may not be the *breathing-line* temperature in all cases.

If the heat transfer between the air and the inside surface of the wall is being considered, then,

$$H = A f_i (t - t_1) \qquad (2)$$

where

f_i = inside surface conductance.
t and t_1 = the temperatures of the inside air and the inside surface of the wall respectively.

In practice it is usually the over-all heat transmission coefficient that is required. This may be determined by a test of the complete wall, or it may be obtained from the individual coefficients by calculation. The simplest method of combining the coefficients for the individual parts of the wall is to use the reciprocals of the coefficients and treat them as resistance units. The total over-all resistance of a wall is equal numerically to the sum of the resistances of the various parts, and the reciprocal of the over-all resistance is likewise the over-all heat transmission coefficient of the wall. For a wall built up of a single homogeneous material of conductivity k and x inches thick the over-all resistance,

$$\frac{1}{U} = \frac{1}{f_i} + \frac{x}{k} + \frac{1}{f_o} \qquad (3)$$

If the coefficients f_i, f_o and k, together with the thickness of the material x are known, the over-all coefficient U may be readily calculated as the reciprocal of the total heat resistance.

For a compound wall built up of three homogeneous materials having conductivities k_1, k_2 and k_3 and thicknesses x_1, x_2 and x_3 respectively, and laid together without air spaces, the total resistance,

$$\frac{1}{U} = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{1}{f_o} \qquad (4)$$

For a wall with air space construction consisting of two homogeneous materials of thicknesses x_1 and x_2, and conductivities k_1 and k_2, respectively, separated to form an air space of conductance a, the over-all resistance,

$$\frac{1}{U} = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{1}{a} + \frac{x_2}{k_2} + \frac{1}{f_o} \qquad (5)$$

Likewise any combination of homogeneous materials and air spaces can be put into the wall and the over-all resistance of the combination may be calculated by adding the resistances of the individual sections of the wall. In certain special forms of construction such as tile with irregular air spaces it is necessary to consider the conductance C of the unit as built instead of the unit conductivity k, and the resistance of the section is equal to $\frac{1}{C}$. The method of calculating the over-all heat transmission coefficient for a given wall is comparatively simple, but the selection of the proper coefficients is often complicated. In some cases the construction of the wall is such that the substituting of coefficients in the accepted formula will give erroneous results. This is the case with irregular cored

out air spaces in concrete and tile blocks, and walls in which there are parallel paths for heat flow through materials having different heat resistances. In such cases it is necessary to resort to test methods to check the calculations, and in practically all cases it has been necessary to determine fundamental coefficients by test methods.

Conductivity of Homogeneous Materials

The thermal conductivity of homogeneous materials is affected by several factors. Among these are the density of the material, the amount of moisture present, the mean temperature at which the coefficient is determined, and for fiberous materials the arrangement of fiber in the material. There are many fiberous materials used in building construction and considered as homogeneous for the purpose of calculation, whereas they are not really homogeneous but are merely considered so as a matter of convenience. In general, the thermal conductivity of a material increases directly with the density of the material, increases with the amount of moisture present, and increases with the mean temperature at which the coefficient is determined. The rate of increase for these various factors is not the same for all materials, and in assigning proper coefficients one should make certain that they apply for the conditions under which the material is to be used in a wall. Failure to do this may result in serious errors in the final coefficients.

Surface Conductance Coefficients

Heat is transmitted to or from the surface of a wall by a combination of radiation, convection and conduction. The coefficient will be affected by any factor which has an influence on any one of these three methods of transfer. The amount of heat by radiation is controlled by the character of the surface and the temperature difference between it and the surrounding objects. The amount of heat by conduction and convection is controlled largely by the roughness of the surface, by the air movement over the surface and by the temperature difference between the air and the surface. Because of these variables the surface coefficients may be subject to wide fluctuations for different materials and different conditions. The inside and outside coefficients f_i and f_o are in general affected to the same extent by these various factors and test coefficients determined for inside surfaces will apply equally well to outside surfaces under like conditions. Values for f_i in still and moving air at different mean temperatures have been determined for various building materials[1].

The relation obtained between surface conductances for different materials at mean temperatures of 20 F is shown in Fig. 1. These values were obtained with air flow parallel to the surface and from other tests in which the angle of incidence between the direction of air flow and the surface was varied from zero to 90 F it would appear that these values might be lowered approximately 15 per cent for average conditions. While for average building materials there is a difference due to mean temperature, the greatest variation in these coefficients is caused by the character of the surface and the wind velocity. If other surfaces, such as aluminum foil with low emissivity coefficients were substituted, a large

[1]A.S.H.V.E. RESEARCH REPORT No. 869—Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B Algren and J. L Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 429).

part of the radiant heat would be eliminated. This would reduce the total coefficient for all wind velocities by about 0.7 Btu and would make but very little difference for the higher wind velocities. In many cases in building construction the heat resistance of the internal parts of the wall is high as compared with the surface resistance and the surface factors become of small importance. In other cases such as single glass windows the surface resistances constitute practically the entire resistance of the structure, and therefore become important factors. Due to the wide variation in surface coefficients for different conditions their selection for a practical building becomes a matter of judgment. In calculating the

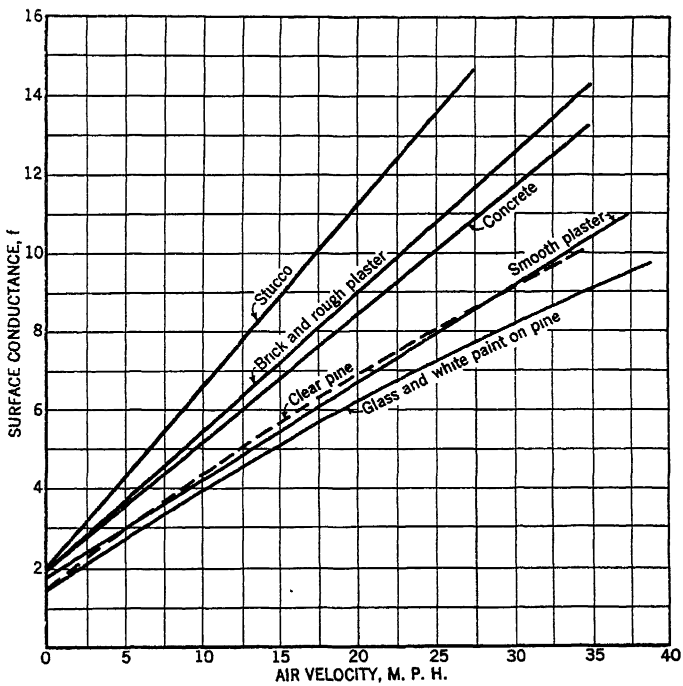


FIG. 1. CURVES SHOWING RELATION BETWEEN SURFACE CONDUCTANCES FOR DIFFERENT SURFACES AT 20 F MEAN TEMPERATURE

over-all coefficients for the walls of Tables 3 to 12, 1.65 has been selected as an average inside coefficient and 6.0 as an average outside coefficient for a 15-mile wind velocity. In special cases where surface coefficients become important factors in the over-all rate of heat transfer more selective coefficients may be required.

The surface conductance values given in Table 1, **Section A** are based on recent tests and are for still air conditions and emissivities of 0.83 and 0.05, respectively.

Air Space Conductance

Heat is conducted across an air space by a combination of radiation, conduction and convection. The amount of heat by radiation is governed

Table 1. Conductances (C) for Surfaces and Air Spaces

All conductance values expressed in Btu per hour per square foot per degree Fahrenheit temperature difference.

Section A. Surface Conductances for Still Air[a]

Position of Surface	Direction of Heat Flow	Surface Emissivity $e = 0.83$	Surface Emissivity $e = 0.05$
Horizontal	Upward	1.95	1.16
Horizontal	Downward	1.21	0.44
Vertical		1.52	0.74

Section B. Conductance of Vertical Spaces at Various Mean Temperatures[b]

Mean Temp Deg Fahr	Conductances of Air Spaces for Various Widths in Inches						
	0.128	0.250	0.364	0.493	0 713	1.00	1.500
20	2.300	1.370	1.180	1.100	1.040	1.030	1.022
30	2.385	1.425	1.234	1.148	1.080	1.070	1.065
40	2.470	1 480	1.288	1.193	1.125	1.112	1.105
50	2.560	1.535	1.340	1.242	1.168	1.152	1.149
60	2.650	1.590	1.390	1.295	1.210	1.195	1.188
70	2.730	1.648	1.440	1.340	1.250	1.240	1.228
80	2.819	1.702	1.492	1.390	1.295	1.280	1.270
90	2.908	1.757	1.547	1.433	1 340	1.320	1.310
100	2.990	1.813	1.600	1.486	1.380	1.362	1.350
110	3.078	1.870	1.650	1.534	1.425	1.402	1.392
120	3.167	1.928	1.700	1.580	1.467	1.445	1.435
130	3.250	1.980	1.750	1.630	1.510	1.485	1.475
140	3.340	2.035	1.800	1.680	1.550	1.530	1.519
150	3.425	2.090	1.852	1.728	1.592	1.569	1.559

Section C. Conductances and Resistances of Air Spaces Faced with Reflective Insulation[c]

Location and Position of Air Space	Direction of Heat Flow	Temp[d] Diff Deg Fahr		Conductance[e] (C) No. of Air Spaces			Resistance[e] $\left(\frac{1}{C}\right)$ No. of Air Spaces		
		Winter	Summer	1	2	3	1	2	3
Rafter Space (8 in.)									
Horizontal	Down	45			0.10	0.07		10.00	14.29
Horizontal	Up	45			0.27	0.17		3.70	5.88
Horizontal	Down		25		0.09	0.06		11.11	16.67
Horizontal	Up		25		0.24	0.16		4.17	6.25
30 deg slope	Down	45			0.15	0.10		6.67	10.00
30 deg slope	Up	45			0.25	0.17		4.00	5.88
30 deg slope	Down		25		0.13	0.09		7.69	11.11
30 deg slope	Up		25		0.23	0.14		4.35	7 14
Stud Space (3⅝ in.)									
Vertical[f]		30		0.34			2.94		
Vertical		40			0.23	0.13		4.35	7.69
Vertical[f]			15	0.32			3.13		
Vertical			20		0.18	0.11		5.56	9.09
Vertical[g]		30		0.46			2.17		

[a]Radiation and Convection from Surfaces in Various Positions, by G. B. Wilkes and C. M. F. Peterson (A.S.H.V.E. Transactions, Vol. 44, 1938, p. 513).

[b]A S.H.V.E. Research Report No. 825—Thermal Resistance of Air Spaces, by F. B. Rowley and A. B. Algren (A.S.H.V.E. Transactions, Vol. 35, 1929, p. 165).

[c]Thermal Test Coefficients of Aluminum Insulation for Buildings, by G. B. Wilkes, F. G. Hechler and E. R. Queer (A.S H.V.E. Transactions, Vol. 46, 1940).

[d]Temperature difference is based on total space between plaster base and sheathing, flooring or roofing.

[e]These air space conductance and resistance values are based on one reflective surface (aluminum) having an emissivity of 0 05 facing each space and are based on total space between plaster base and sheathing, flooring or roofing. The rafter and stud spaces are divided into equal spaces.

[f]Stud space is lined on plaster base side with loose paper with aluminum on surface facing air space. The resistance of the small air space between the plaster base and paper was 0.43.

[g]Radiation and Convection Across Air Spaces in Frame Construction, by G. B. Wilkes and C. M. F. Peterson (A S.H.V.E. Transactions, Vol. 43, 1937, p. 351).

largely by the nature of the surface and the temperature difference between the boundary surfaces of the air space. Conduction and convection are controlled largely by the width and shape of the air space and the roughness of the boundary surfaces.

The conductances of vertical air spaces bounded by such materials as paper, wood, plaster, etc., are given in Table 1, **Section B,** having emissivity coefficients of 0.8 or higher, and with extended parallel surfaces perpendicular to the direction of heat flow. A conductance of 1.10 Btu per hour per square foot per degree Fahrenheit temperature difference (resistance = 0.91) based on this table was used for calculating the overall coefficients given in Tables 3 to 12 inclusive for air spaces ¾ in. or more in width. Air space tests[2] reported by Wilkes and Peterson resulted in comparable values. For 3⅝ in. horizontal air spaces having an effective emissivity of 0.83, the conductance for heat flow upward was 1.32 and for heat flow downward, 0.94. The conductance for a similar vertical air space was 1.17, the resistances of course being the reciprocals of these values in each case.

A large part of the heat transferred across air spaces bounded by ordinary materials is by radiation. Therefore, if such air spaces are faced with metallic surfaces such as aluminum foil, coated sheet steel or other low-emissivity, infra-red reflective metal surfaces, the radiant heat transfer will be substantially reduced, thus causing the major portion of the remaining transmitted heat to be by convection. Table 1, **Section C,** gives conductances and resistances for air spaces bounded by one reflective surface having an emissivity of 0.05. It will be noted that the conductance values given in this table are a function of the temperature differences across the space rather than mean temperature, the larger the temperature difference, the larger the conductance. The radiant heat transfer is the same regardless of whether the low emissivity surface is on the high or low temperature surface of the space, and is independent of the width of the space. To minimize the convection transfer the vertical air space should be at least ¾ in. in width. A conductance of 0.46 was used for computing the overall coefficients in Tables 3 to 12 inclusive for air spaces bounded by aluminum foil applied to plasterboard.

When referring to reflective heat-insulating surfaces, the term *brightness* which deals with visible light has no specific meaning and should be avoided[3]. *Emissivity* and *reflectivity* definitely define the radiating and reflecting properties and values may be determined directly for long wavelength radiation corresponding to room temperature. As previously stated, the values in Table 1, **Section C,** are based on an emissivity of the reflective surface of 0.05. Obviously for higher emissivity values the conductances will increase accordingly. For example, non-metallic reflective materials are available having emissivity values approximately midway between those of metallic reflective insulations and ordinary building material surfaces. These materials will have a correspondingly higher radiant heat transfer and where such materials are under consideration, due allowance should be made for the higher emissivity value in arriving at the proper air space conductance.

[2]Radiation and Convection Across Air Spaces in Frame Construction, by G. B. Wilkes and C. M. F. Peterson (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 351).

[3]Some Reflection and Radiation Characteristics of Aluminum, by C. S. Taylor and J. D. Edwards (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 179).

Where reflective insulating materials are involved the possible increase in the emissivity coefficient due to surface coatings or chemical action[4] should be studied by the engineer in order to satisfy himself as to the permanence of the reflective surface for the conditions under which this material will be used. In making installations of this material the partitions between air spaces should be tight, particularly at the top and bottom so that air cannot circulate between adjacent spaces.

When reflective insulating materials are installed with multiple air spaces, the position (vertical, horizontal or inclined) of the material in the structure must be taken into consideration. For example, the resistance to heat flow upward is about one-third that of downward flow in a horizontal position in the same construction, as will be apparent from Table 1, Section C. However, the difference between upward heat flow through single horizontal or sloping air spaces and through single vertical air spaces is comparatively small for the same temperature difference. Consequently the same conductance value (0.46) was used for computing the coefficients in Tables 8 and 12, involving horizontal and sloping air spaces bounded on one side by aluminum foil applied to plasterboard, as for similar vertical air spaces in Tables 3, 4, 5 and 6.

As already stated, a conductance value of 1.10 was similarly used in all cases for calculating the coefficients of construction involving vertical, horizontal and sloping air spaces bounded on both sides by ordinary building materials.

PRACTICAL COEFFICIENTS

For practical purposes it is necessary to have average coefficients that may be applied to various materials and types of construction without the necessity of making tests on the individual material or combination of materials. In Table 2 coefficients are given for a group of materials which have been selected from various sources. Wherever possible the properties of material and conditions of tests are given. However, in selecting and applying these values to any construction a reasonable amount of caution is necessary; variations will be found in the coefficients for the same materials, which may be partly due to different test methods used but which are largely due to variations in materials. The recommended coefficients which have been used for the calculation of over-all coefficients as given in Tables 3 to 12 are marked by an asterisk.

It should be recognized in these tables of calculated coefficients that space limitations will not permit the inclusion of all the combinations of materials that are used in building construction and the varied applications of insulating materials to these constructions. Typical examples are given of combinations frequently used, but any special construction not given in Tables 3 to 12 can generally be computed by using the conductivity values given in Table 2 and the fundamental heat transfer formulae. For example, the tabulation of all of the values for multiple layers of insulating materials would present extensive and detailed problems of calculations for the varied application combinations, but the engineer having the fundamental conductivity values can quickly obtain the proper coefficients.

[4]Thermal Test Coefficients of Aluminum Insulation for Buildings, by G. B. Wilkes, F. G. Hechler and E. R. Queer (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940).

Attention is called to the fact that the conductivity values per inch of thickness do not afford a true basis for comparison between insulating materials as applied, although they are frequently used for that purpose. The value of an insulating material is measured in terms of the coefficient (U_1) of the insulated construction as compared to the coefficient (U) of the construction without insulation. Certain types of blanket insulations are designed to be installed between the studs of a frame building in such manner as to give two air spaces. In order to get the full value of such materials they should be so installed that each air space is approximately 1 in. or more in thickness and the air spaces should be sealed at the top and bottom to prevent the circulation of air from one space to the other. Another common error in installing such a material is to nail the blanket on the outside of the studs underneath the sheathing, in which case one air space is lost and also the thickness of the insulating material is materially reduced at the studs. There are certain other types of insulation which are very porous, allowing air circulation within the material if not properly installed. The architect or engineer must carefully evaluate the economic considerations involved in the selection of an insulating material as adapted to various building constructions. Lack of good judgment in the intelligent choice of an insulating material, or its improper installation, frequently represents the difference between good or unsatisfactory results.

Computed Transmission Coefficients

Computed heat transmission coefficients of many common types of building construction are given in Tables 3 to 13, inclusive, each construction being identified by a serial number. For example, the coefficient of transmission (U) of an 8-in. brick wall and ½ in. of plaster is 0.46, and the number assigned to a wall of this construction is *1-B*, Table 3.

Example 1. Calculate the coefficient of transmission (U) of an 8-in. brick wall with ½ in. of plaster applied directly to the interior surface, based on an outside wind exposure of 15 mph. It is assumed that the outside course is of hard (high density) brick having a conductivity of 9.20, and that the inside course is of common (low density) brick having a conductivity of 5.0, the thicknesses each being 4 in. The conductivity of the plaster is assumed to be 3.3, and the inside and outside surface coefficients are assumed to average 1.65 and 6.00, respectively for still air and a 15 mph wind velocity.

Solution. k (hard high density brick) = 9.20; x = 4.0 in.; k (common low density brick) = 5.0; x = 4.0 in.; k (plaster) = 3.3; x = ½ in.; f_i = 1.65; f_o = 6.0. Therefore,

$$U = \frac{1}{\frac{1}{6.0} + \frac{4.0}{9.20} + \frac{4.0}{5.0} + \frac{0.5}{3.3} + \frac{1}{1.65}}$$

$$= \frac{1}{0.167 + 0.435 + 0.80 + 0.152 + 0.606}$$

= 0.46 Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides.

The coefficients in the tables were determined by calculations similar to those shown in Example 1, using Fundamental Formulae 3, 4 and 5 and the values of k (or C), f_i, f_o and a indicated in Table 2 by asterisks. In computing heat transmission coefficients of floors laid directly on the ground (Table 10), only one surface coefficient (f_i) is used. For example,

the value of U for a 1-in. yellow pine floor (actual thickness, 25/32 in.) placed directly on 6-in. concrete on the ground, is determined as follows:

$$U = \frac{1}{\frac{1}{1.65} + \frac{0.781}{0.80} + \frac{6.0}{12.0}} = 0.48 \text{ Btu per hour per square foot per degree difference}$$

in temperature between the ground and the air immediately above the floor.

Rigid insulation refers to so-called insulating board which may be used structurally, such as for sheathing. Flexible insulation refers to the blankets, quilts or semi-rigid types of insulation.

Actual thicknesses of lumber are used in the computations rather than nominal thicknesses. The computations for wood shingle roofs applied over wood stripping are based on 1 by 4 in. wood strips, spaced 2 in. apart. Since no reliable figures are available concerning the conductivity of Spanish and French clay roofing tile, of which there are many varieties, the figures for such types of roofs were taken the same as for slate roofs, as it is probable that the values of U for these two types of roofs will compare favorably.

The thicknesses upon which the coefficients in Tables 3 to 13 inclusive, are based are as follows:

Brick veneer	4 in.
Plaster and metal lath	3/4 in.
Plaster (on wood lath, plasterboard, rigid insulation, board form, or corkboard)	1/2 in.
Slate (roofing)	1/2 in.
Stucco on wire mesh reinforcing	1 in.
Tar and gravel or slag-surfaced built-up roofing	3/8 in.
1-in. lumber (S-2-S)	25/32 in.
1 1/2-in. lumber (S-2-S)	1 5/16 in.
2-in. lumber (S-2-S)	1 5/8 in.
2 1/2-in. lumber (S-2-S)	2 1/8 in.
3-in. lumber (S-2-S)	2 5/8 in.
4-in. lumber (S-2-S)	3 5/8 in.
Finish flooring (maple or oak)	13/16 in.

Solid brick walls are based on 4 in. hard brick (high density) and the remainder common brick (low density). Stucco is assumed to be 1 in. thick on masonry walls. Where metal lath and plaster are specified, the metal lath is neglected.

The coefficients of transmission of the pitched roofs in Table 12 apply where the roof is over a heated attic or top floor so the heat passes directly through the roof structure including whatever finish is applied to the underside of the roof rafters.

It is the practice of many engineers in calculating heat losses to use a minimum coefficient of 0.10 to allow for possible defects in workmanship, poor construction and other factors which would increase the heat loss. The lower the theoretical wall or roof coefficient the greater will be the percentage of error due to construction defects or failure of the insulation to perform as rated.

Combined Coefficients of Transmission

If the attic is unheated, the roof structure and ceiling of the top floor must both be taken into consideration, and the combined coefficient of transmission determined. The formula for calculating the combined

Table 2. Conductivities (k) and Conductances (C) of Building Materials and Insulators[a]

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness unless otherwise indicated.

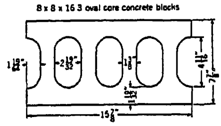
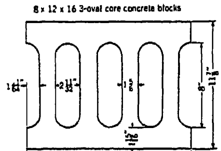
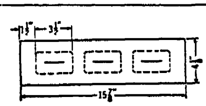
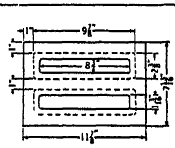
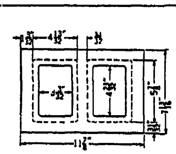
Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
MASONRY MATERIALS						
Brick	Low density			5.00*	0.20	—
	High density			9.20*	0.11	—
	Adobe			3.56*	0.28	—
Brickwork	Damp or wet			5.00[c]	0.20	(2)
Cement Mortar	Typical			12.00*	0.08	—
Concrete	Typical			12.00*	0.08	—
	Various ages and mixes[d]			11.35 to 16 36		(5)
	Concrete plank	76	75	2 5	4.0	(3)
	Cellular	40.0	75	1.06	0.94	(3)
	Cellular	50.0	75	1.44	0.69	(3)
	Cellular	60.0	75	1.80	0.56	(3)
	Cellular	70.0	75	2.18	0.46	(3)
	Typical fiber gypsum, 87.5% gypsum and 12.5% wood chips	51.2	74	1.66*	0.60	(4)
	Special concrete made with an aggregate of hardened clay—1-2-3 mix	101.0	70	3.98	0.25	(3)
	Sand and gravel	142 0	75	12.6	0.08	(4)
	Limestone	132 0	75	10.8	0.09	(4)
	Cinder	97 0	75	4.9	0.22	(4)
	Burned clay aggregate	75.0	75	4.0	0.25	(4)
	Blast furnace slag aggregate	76 0	70	1.6	0.63	(3)
	Expanded vermiculite aggregate	20	90	0 68	1 47	(3)
	Expanded vermiculite aggregate	26 7	90	0 76	1 32	(3)
	Expanded vermiculite aggregate	35	90	0 86	1.16	(3)
	Expanded vermiculite aggregate	50	90	1 10	0 91	(3)
Stone	Typical			12.50*	0.08	—
Stucco	Typical			12.00*	0.08	—
Tile	Typical hollow clay (4 in.)			1.00†*	1.00	—
	Typical hollow clay (6 in.)[e]			0.64†*	1.57	—
	Typical hollow clay (8 in.)[e]			0.60†*	1.67	—
	Typical hollow clay (10 in.)[e]			0.58†*	1.72	—
	Typical hollow clay (12 in.)[e]			0.40†*	2.50	—
	Typical hollow clay (16 in.)[e]			0.31†*	3.23	—
	Hollow clay (2 in.) ½-in. plaster both sides	120.0	110	1.00†	1.00	(2)
	Hollow clay (4 in.) ½-in. plaster both sides	127.0	100	0.60†	1.67	(2)
	Hollow clay (6 in.) ½-in. plaster both sides	124.3	105	0.47†	2.13	(2)
	Hollow gypsum (4 in.)			0.46†*	2.18	—
	Solid gypsum	51.8	70	1.66	0.60	(4)
	Solid gypsum	75.6	76	2.96	0.34	(4)
Tile or Terrazzo	Typical flooring			12.00*	0.08	—

Authorities:

[1]U. S. Bureau of Standards, tests based on samples submitted by manufacturers.
[2]A. C. Willard, L. C. Lichty, and L. A. Harding, tests conducted at the University of Illinois.
[3]J. C. Peebles, tests conducted at Armour Institute of Technology, based on samples submitted by manufacturers.
[4]F. B. Rowley, tests conducted at the University of Minnesota.
[5]A.S.H.V.E. Research Laboratory.
[6]E. A. Allcut, tests conducted at the University of Toronto.
[7]Lees and Charlton.
*Recommended conductivities and conductances for computing heat transmission coefficients.
†For thickness stated or used on construction, not per 1 in. thickness.
[a]For additional conductivity data see *A.S.R.E. Data Book.*
[b]If outside surface of block is painted with an impervious coat of paint, add 0.07 to resistance for sand and gravel blocks. Add 0.18 to resistance for cinder blocks Add 0.17 to resistance for burned clay aggregate blocks.
[c]Recommended value. See Heating, Ventilating and Air Conditioning, by Harding and Willard, revised edition, 1932.
[d]See A.S.H.V.E. Research Report No. 915—Conductivity of Concrete, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. Transactions, Vol. 38, 1932, p. 47).
[e]The 6-in., 8-in., and 10-in. hollow tile figures are based on two cells in the direction of heat flow. The 12-in. hollow tile is based on three cells in the direction of heat flow. The 16-in. hollow tile consists of one 10-in. and one 6-in. tile, each having two cells in the direction of heat flow.
[f]Not compressed.
[g]Roofing, 0.15-in. thick (1.34 lb per sq ft), covered with gravel (0.83 lb per sq ft), combined thickness assumed 0.25.

Table 2. Conductivities (k) and Conductances (C) of Building Materials and Insulators[a]—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness unless otherwise indicated.

Material	Description: Cement	Fine Aggregate 0-No. 4	Coarse Aggregate No. 4-1/2	Slump	Per Cent	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
MASONRY MATERIALS —Continued										
Steam Treated Limestone Slag	1	7.00			27.1	74.6	74.49	2.27	0.44	(4)
Pumice Mined in Calif.	1	8.00			26.5	65.0	74.68	2.42	0.41	(4)
By-Product of Manufacture of Phosphates	1	8.00	Fineness Modulus 3.75		25.5	86.6	74.62	3.19	0.31	(4)
	1	8.00			21.1	91.1	74.43	3.42	0.29	(4)
Expanded Burned Clay	1	8.00			18 4	57 9	75.57	2.28	0.44	(4)
Burned Clay Aggregate	1	8.50			21.8	67.1	75.89	2.89	0.35	(4)
Burned Clay Aggregate	1	8.50			21.8	67.1	74.60	2.82	0.35	(4)
8 x 8 x 16 3 oval core concrete blocks 	Sand and gravel aggregate					126.4	40	0.90†	1.11	(4)
	Sand and gravel aggregate used for calculations							1.00†*	1.00	(4)
	Cores filled with 5 14 lb density cork						40	0.56†*	1.79	(4)
	Crushed limestone aggregate					134.3	40	0.86†*	1.16	(4)
	Cinder aggregate					86.2	40	0.58†	1.73	(4)
	Cinder aggregate used for calculations							0.60†*	1.66	(4)
	Cores filled with 69.7 lb density cinders						40	0.39†	2.56	(4)
	Cores filled with 5 12 lb density cork						40	0.25†*	4.00	(4)
	Cores filled with 14.2 lb density rock wool						40	0.27†*	3.70	(4)
	Burned clay aggregate					67.7	40	0.50†*	2.00	(4)
	Cores filled with 5.06 lb density cork						40	0.21†	4.76	(4)
	Expanded blast furnace slag aggregate, 60% fine, 40% course						40	0.49†	2.04	(4)
8 x 12 x 16 3-oval core concrete blocks 	Sand and gravel aggregate					124.9	40	0.78†	1.28	(4)
	Sand and gravel aggregate used for calculations							0.80†*		(4)
	Cinder aggregate					86.2	40	0.53†*	1.88	(4)
	Cores filled with 5.24 lb density cork						40	0.24†*	4.16	(4)
	Burned clay aggregate					76.7	40	0.47†	2.13	(4)
	Cores filled with 5 6 lb density cork						40	0.17†*	5.88	(4)
	Cinder aggregate					100.0	40	1.00†	1.00	(4)
	Double wall with 1 in air space between					100.0	40	0.36†	2.78	(4)
	1 in. space filled with 9.97 lb density rock wool					100.0	40	0.20†	5.00	(4)
	5 x 8 x 12 block sand and gravel aggregate					133.7	40	0.38†	2.63	(4)
	5 x 8 x 12 block sand and gravel aggregate[b]					134.0	40	0.95†	1.15	(4)

For notes see page 93.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS[a]—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness unless otherwise indicated.

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
INSULATION—BLANKET OR FLEXIBLE TYPES						
Fiber	Typical	—	—	0.27*	3.70	—
	Chemically treated wood fibers held between layers of strong paper[f]	3.62	70	0.25	4.00	(3)
	Eel grass between strong paper[f]	4.60	90	0.26	3.85	(1)
	" " " " "	3.40	90	0.25	4.00	(1)
	Flax fibers between strong paper[f]	4.90	90	0.28	3.57	(1)
	Chemically treated hog hair between kraft paper[f]	5.76	71	0.26	3.85	(3)
	Chemically treated hog hair between kraft paper and asbestos paper[f]	7.70	71	0.28	3.57	(3)
	Hair felt between layers of paper[f]	11 00	75	0.25	4.00	(3)
	Kapok between burlap or paper[f]	1.00	90	0.24	4.17	(1)
	Jute fiber[f]	6.70	75	0.25	4.00	(3)
	Ground paper between two layers, each ⅜-in. thick made up of two layers of kraft paper (sample ¾-in. thick)	12.1	75	0.40†	2.50	(4)
	Stitched and creped expanding fibrous blanket	1.50	70	0.27	3.70	(3)
	Paper and asbestos fiber with emulsified asphalt binder	4 2	94	0.28	3.57	(1)
	Cotton insulating bat	0 875	72	0.24	4 17	(3)
INSULATION—SEMI-RIGID TYPE						
Fiber	Felted cattle hair[f]	13.00	90	0.26	3.84	(1)
	" " "	11.00	90	0 26	3.84	(1)
	Felted hair and asbestos[f]	7.80	90	0.28	3.57	(1)
	75% hair and 25% jute[f]	6.30	90	0.27	3.70	(1)
	50% hair and 50% jute[f]	6.10	90	0.26	3.85	(1)
	Jute[f]	6.70	75	0.25	4.00	(3)
	Felted jute and asbestos[f]	10.00	90	0.37	2.70	(1)
	Compressed peat moss	11.00	70	0.26	3.84	(3)
INSULATION—LOOSE FILL OR BAT TYPE						
Fiber	Made from ceiba fibers[f]	1.90	75	0.23	4.35	(3)
	" " " "	1.60	75	0.24	4.17	(3)
Fiber	Fibrous material made from dolomite and silica	1.50	75	0.27	3.70	(3)
	Fibrous material made from slag	9.40	103	0.27	3.70	(1)
	Redwood bark	3.00	90	0.31	3.22	(1)
	Redwood bark	5.00	75	0.26	3.84	(3)
Glass Wool	Glass fibers 0.0003 in. to 0.0006 in. in diameter	1.50	75	0.27	3.70	(3)
Granular	Made from combined silicate of lime and alumina	4.20	72	0.24	4.17	(3)
	Expanded vermiculite, particle size −10+20	6 32	86	0.29	3 45	(3)
	Expanded vermiculite, particle size −8+16	5.2	—	0 41	2 44	(1)
Gypsum	Flaked, dry and fluffy[f]	34.00	90	0.60	1.67	(1)
	" " " "	26.00	90	0.52	1.92	(1)
	" " " "	24.00	75	0.48*	2.08	(3)
	" " " "	19.80	90	0.35	2.86	(1)
	" " " "	18.00	75	0.34	2.94	(3)
Mineral Wool	All forms, typical	—	—	0.27*	3.70	.
Regranulated Cork	About 5/16-in. particles	8.10	90	0.31	3.22	(1)
Rock Wool	Fibrous material made from rock	21.00	90	0.30	3.33	(1)
	" " " " "	18.00	90	0.29	3.45	(1)
	" " " " "	14.00	90	0.28	3.57	(1)
	" " " " "	10.00	90	0.27*	3.70	(1)
	Rock wool with a binding agent	14.50	77	0.33	3.03	(1)
	Rock wool with flax, straw pulp, and binder	14.50	75	0.38	2.63	(3)
	Rock wool with vegetable fibers	11.50	72	0.31	3.22	(3)
Sawdust	Various	12.00	90	0.41	2.44	(1)
Shavings	Various from planer	8.80	90	0.41	2.44	(1)
	From maple, beech and birch (coarse)	13.20	90	0.36	2.78	(1)

For notes see Page 93.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS[a]—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in thickness unless otherwise indicated.

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
INSULATION—RIGID						
CORKBOARD	Typical	—	—	0.30*	3.33	—
	No added binder	14.00	90	0.34	2.94	(1)
	" " "	10.60	90	0.30	3.33	(1)
	" " "	7.00	90	0.27	3.70	(1)
	" " "	5.40	90	0.25	4.00	(1)
	Asphaltic binder	14.50	90	0.32	3.12	(1)
FIBER	Typical	—	—	0.33*	3.03	—
	Chemically treated hog hair covered with film of asphalt	10.00	75	0.28	3.57	(3)
	Made from corn stalks	15.00	71	0.33	3.03	(3)
	" " exploded wood fibers	17.90	78	0.32	3.12	(4)
	" " hard wood fibers	15.20	70	0.32	3.12	(3)
	Insulating plaster 9/10 in. thick applied to ⅜ in. plaster board base	54.00	75	1.07†	0.93	(3)
	Made from licorice roots	16.10	81	0.34	2.94	(3)
	" " 85% magnesia and 15% asbestos	19.30	86	0.51	1.96	(1)
	" " shredded wood and cement	24.20	72	0.46	2.17	(3)
	" " sugar cane fiber	13.50	70	0.33	3.03	(3)
	Sugar cane fiber insulation blocks encased in asphalt membrane	13.80	70	0.30	3.33	(3)
	Made from wheat straw	17.00	68	0.33	3.03	(3)
	" " wood fiber	15.90	72	0.33	3.03	(3)
	" " " "	15.00	70	0.33	3.03	(3)
	" " " "	—	52	0.33	3.03	(6)
	" " " "	8.50	72	0.29	3.45	(3)
	" " " "	15.20	—	0.33	3.03	(3)
	" " " "	16.90	90	0.34	2.94	(1)
INSULATION-REFLECTIVE	See Table 1, Section C					
BUILDING BOARDS						
ASBESTOS	Compressed cement and asbestos sheets	123.00	86	2.70	0.37	(1)
	Corrugated asbestos board	20.40	110	0.48	2.08	(2)
	Pressed asbestos mill board	60.50	86	0.84	1.19	(1)
GYPSUM	Gypsum between layers of heavy paper	62.80	70	1.41	0.71	(3)
	Rigid, gypsum between layers of heavy paper (½ in. thick)	53.50	90	2.60†	0.38	(1)
	Gypsum mixed with sawdust between layers of heavy paper (0.39 in. thick)	60.70	90	3.60†	0.28	(1)
PLASTERBOARD	(⅜ in.)	—	—	3.73†*	0.27	—
	(½ in.)	—	—	2.82†*	0.35	—
ROOFING CONSTRUCTION						
ROOFING	Asphalt, composition or prepared	70.00	75	6.50†*	0.15	(3)
	Built up—⅜ in. thick	—	—	3.53†*	0.28	—
	Built up, bitumen and felt, gravel or slag surfaced[e]	—	—	1.33	0.75	(2)
	Plasterboard, gypsum fiber concrete and 3-ply roof covering 2½ in. thick	52.40	76	0.58†	1.72	(4)
SHINGLES	Asbestos	65.00	75	6.00†*	0.17	(3)
	Asphalt	70.00	75	6.50†*	0.15	(3)
	Slate	201.00	—	10.37*	0.10	(7)
	Wood	—	—	1.28†*	0.78	—
PLASTERING MATERIALS						
PLASTER	Cement	—	—	8.00	0.13	(2)
	Gypsum, typical	—	—	3.30*	0.30	—
	Gyysum and expanded vermiculite mix 4 to 1	39.9	75	0.85	1.18	(3)
	Thickness ⅜ in	—	73	8.80†	0.11	(4)
METAL LATH AND PLASTER	Total thickness ¾ in	—	—	4.40†*	0.23	—
WOOD LATH AND PLASTER	⅜ in plaster, total thickness ¾ in	. ..	70	2 50†*	0.40	(4)
BUILDING CONSTRUCTIONS						
FRAME	1-in. fir sheathing and building paper	—	30	0.86†*	1.16	(4)
	1-in fir sheathing, building paper, and yellow pine lap siding	—	20	0.50†*	2.00	(4)
	1-in fir sheathing, building paper and stucco	—	20	0.82	1.22	(4)
	Pine lap siding and building paper—siding 4 in. wide	—	16	0.85†*	1.18	(4)
	Yellow pine lap siding	—	—	1.28†*	0.78	—

For notes see Page 93.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS[a]—Concluded

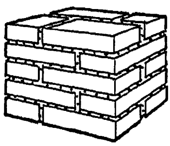
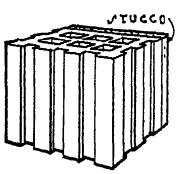
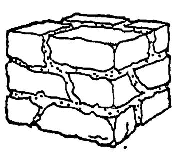
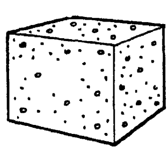
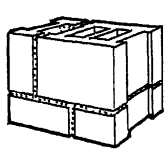
The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness unless otherwise indicated

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
BUILDING CONSTRUCTIONS—(Continued)						
FLOORING	Maple—across grain	40.00	75	1.20	0.83	(3)
	Battleship linoleum (¼ in.)	—	—	1.36†*	0.74	—
WOODS (Across Grain)						
BALSA		20.0	90	0.58	1.72	(1)
		8.8	90	0.38	2.63	(1)
		7.3	90	0.33	3.03	(1)
CALIFORNIA REDWOOD	0% moisture	22.0	75	0.66	1.53	(4)
	0% "	28.0	75	0.70	1.43	(4)
	16% "	22.0	75	0.74	1.35	(4)
	16% "	28.0	75	0.80	1.25	(4)
CYPRESS		28.7	86	0.67	1.49	(1)
DOUGLAS FIR	0% moisture	26.0	75	0.61	1.64	(4)
	0% "	34.0	75	0.67	1.49	(4)
	16% "	26.0	75	0.76	1.32	(4)
	16% "	34.0	75	0.82	1.22	(4)
EASTERN HEMLOCK	0% moisture	22.0	75	0.60	1.67	(4)
	0% "	30.0	75	0.76	1.32	(4)
	16% "	22.0	75	0.67	1.49	(4)
	16% "	30.0	75	0.85	1.18	(4)
HARD MAPLE	0% moisture	40.0	75	1.01	0.99	(4)
	0% "	46.0	75	1.05	0.95	(4)
	16% "	40.0	75	1.15	0.87	(4)
	16% "	46.0	75	1.21	0.83	(4)
LONGLEAF YELLOW PINE	0% moisture	30.0	75	0.76	1.32	(4)
	0% "	40.0	75	0.86	1.16	(4)
	16% "	30.0	75	0.89	1.12	(4)
	16% "	40.0	75	1.03	0.97	(4)
MAHOGANY		34.3	86	0.90	1.11	(1)
MAPLE		44.3	86	1.10	0.91	(1)
MAPLE OR OAK		—	—	1.15*	0.87	—
NORWAY PINE	0% moisture	22.0	75	0.62	1.61	(4)
	0% "	32.0	75	0.74	1.35	(4)
	16% "	22.0	75	0.74	1.35	(4)
	16% "	32.0	75	0.91	1.10	(4)
RED CYPRESS	0% moisture	22.0	75	0.67	1.49	(4)
	0% "	32.0	75	0.79	1.27	(4)
	16% "	22.0	75	0.74	1.35	(4)
	16% "	32.0	75	0.90	1.11	(4)
RED OAK	0% moisture	38.0	75	0.98	1.02	(4)
	0% "	48.0	75	1.18	0.85	(4)
	16% "	38.0	75	1.07	0.94	(4)
	16% "	48.0	75	1.29	0.78	(4)
SHORTLEAF YELLOW PINE	0% moisture	26.0	75	0.74	1.35	(4)
	0% "	36.0	75	0.91	1.10	(4)
	16% "	26.0	75	0.84	1.19	(4)
	16% "	36.0	75	1.04	0.96	(4)
SOFT ELM	0% moisture	28.0	75	0.73	1.37	(4)
	0% "	34.0	75	0.88	1.14	(4)
	16% "	28.0	75	0.81	1.24	(4)
	16% "	34.0	75	0.97	1.03	(4)
SOFT MAPLE	0% moisture	36.0	75	0.89	1.12	(4)
	0% "	42.0	75	0.95	1.05	(4)
	16% "	36.0	75	1.01	0.99	(4)
	16% "	42.0	75	1.09	0.92	(4)
SUGAR PINE	0% moisture	22.0	75	0.54	1.85	(4)
	0% "	28.0	75	0.64	1.56	(4)
	16% "	22.0	75	0.65	1.54	(4)
	16% "	28.0	75	0.78	1.28	(4)
VIRGINIA PINE		34.3	86	0.96	1.04	(1)
WEST COAST HEMLOCK	0% moisture	22.0	75	0.68	1.47	(4)
	0% "	30.0	75	0.79	1.27	(4)
	16% "	22.0	75	0.78	1.28	(4)
	16% "	30.0	75	0.91	1.10	(4)
WHITE PINE		31.2	86	0.78	1.28	(1)
YELLOW PINE		—	—	1.00	1.00	(3)
YELLOW PINE OR FIR		—	—	0.80*	1.25	—

For notes see Page 93.

TABLE 3. COEFFICIENTS OF TRANSMISSION (U) OF MASONRY WALLS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

TYPICAL CONSTRUCTION	TYPE OF WALL	THICKNESS OF MASONRY (INCHES)	WALL No
	**Solid Brick** Based on 4-in hard brick and the remainder common brick.	8 12 16	**1** **2** **3**
	**Hollow Tile** Stucco Exterior Finish. The 8-in. and 10-in. tile figures are based on two cells in the direction of flow of heat. The 12-in. tile is based on three cells in the direction of flow of heat. The 16-in. tile consists of one 10-in. tile and one 6-in tile each having two cells in the direction of heat flow.	8 10 12 16	**4** **5** **6** **7**
	**Limestone or Sandstone**	8 12 16 24	**8** **9** **10** **11**
	**Concrete (Monolithic)** These figures may be used with sufficient accuracy for concrete walls with stucco exterior finish.	6 10 16 20	**12** **13** **14** **15**
	Cinder (Monolithic) Conductivity k = 4.36	6 10 16 20	**16** **17** **18** **19**
	Burned Clay aggregate (Monolithic) Conductivity k = 3.96	6 10 16 20	**20** **21** **22** **23**
	**Cinder Blocks**	8	**24**
	Cores filled with dry cinders, 69.7 lb per cu ft.	8	**25**
	Cores filled with granulated cork, 5.12 lb per cu ft.	8	**26**
	Cores filled with rock wool, 14.2 lb per cu ft	8	**27**
	Based on one air cell in direction of heat flow.	12	**28**
	Cores filled with granulated cork, 5.24 lb per cu ft.	12	**29**
	Concrete Blocks	8	**30**
	Cores filled with granulated cork, 5.14 lb per cu ft.	8	**31**
	Based on one air cell in direction of heat flow.	12	**32**
	Burned Clay aggregate Blocks	8	**33**
	Cores filled with granulated cork, 5.06 lb per cu ft.	8	**34**
	Burned Clay aggregate Blocks	12	**35**
	Cores filled with granulated cork, 5.6 lb per cu ft.	12	**36**

[a]Computed from factors marked by * in Table 2.

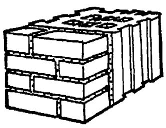
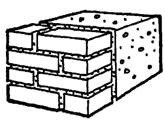
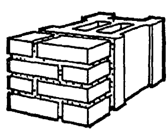
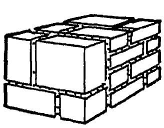
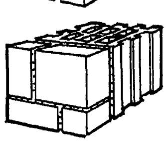
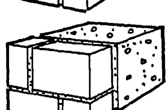
[b]Based on the actual thickness of 2 in furring strips.

INTERIOR FINISH											
UNINSULATED WALLS						INSULATED WALLS					
Plain walls—no interior finish	Plaster (½ in.) on walls	Plaster on wood lath—furred	Plaster (¾ in.) on metal lath—furred	Plaster (½ in.) on plasterboard (⅜ in.)—furred	Decorated building board (½ in.) without plaster—furred	Plaster (½ in.) on rigid insulation (½ in.)—furred	Plaster (½ in.) on rigid insulation (1 in.)—furred	Plaster (½ in.) on corkboard (1½ in.) set in cement mortar (½ in.)	Plaster (½ in.) on plasterboard (⅜ in.)—airspace faced on one side with bright aluminum foil cemented to plasterboard	Plaster on metal lath attached to furring strips (2 in.[b])—rock wool fill (1⅝ in.[b])[c]	Plaster (¾ in.) on metal lath attached to furring strips (2 in.[b]) — flexible insulation (½ in.) between furring strips (one air space)
A	B	C	D	E	F	G	H	I	J	K	L
0.50	0.46	0.30	0.32	0.30	0.23	0.22	0.16	0.14	0.22	0.12	0.20
0.36	0.34	0.24	0.25	0.24	0.19	0.19	0.14	0.12	0.18	0.11	0.17
0.28	0.27	0.20	0.21	0.20	0.17	0.16	0.13	0.11	0.16	0.10	0.15
0.40	0.38	0.26	0.28	0.26	0.20	0.20	0.15	0.13	0.20	0.11	0.18
0.39	0.37	0.26	0.27	0.26	0.20	0.19	0.15	0.13	0.20	0.11	0.18
0.30	0.29	0.22	0.22	0.22	0.17	0.17	0.14	0.12	0.17	0.10	0.16
0.25	0.24	0.19	0.19	0.19	0.16	0.15	0.12	0.11	0.16	0.097	0.14
0.71	0.64	0.37	0.39	0.37	0.26	0.25	0.18	0.15	0.25	0.13	0.23
0.58	0.53	0.33	0.34	0.33	0.24	0.23	0.17	0.14	0.23	0.13	0.21
0.49	0.45	0.30	0.31	0.30	0.22	0.22	0.16	0.14	0 22	0.12	0.20
0.37	0.35	0.25	0.26	0.25	0.20	0.19	0.15	0.13	0.19	0.11	0.18
0.79	0.70	0.39	0.42	0.39	0.27	0.26	0.19	0.16	0.26	0.13	0.23
0.62	0.57	0.34	0.37	0.34	0.25	0.24	0.18	0.15	0.24	0.13	0.22
0.48	0.44	0.29	0.31	0.29	0.22	0.21	0.16	0.14	0.21	0.12	0.20
0.41	0.39	0.27	0.28	0.27	0.21	0.20	0.15	0.13	0.20	0.12	0.18
0.46	0.43	0.29	0.30	0.29	0.22	0.21	0.16	0.14	0.21	0.12	0.19
0.33	0.31	0.23	0.24	0.23	0.18	0.18	0.14	0.12	0.18	0.11	0.16
0.22	0.22	0.17	0.18	0.17	0.15	0.14	0.12	0.10	0.15	0.09	0.13
0.19	0.18	0.15	0.15	0.15	0.13	0.13	0.11	0.09	0.13	0.09	0.12
0.44	0.41	0.28	0.29	0.28	0.21	0.21	0.16	0.13	0.21	0.12	0.19
0.30	0.29	0.22	0.23	0.22	0.17	0.17	0.14	0.12	0.17	0.10	0.16
0.21	0.20	0.16	0.17	0.16	0.14	0.14	0.11	0.10	0.13	0.09	0.13
0.17	0.17	0.14	0.14	0.14	0.12	0.12	0.10	0.09	0.12	0.08	0.11
0.42	0.39	0.27	0.28	0.27	0.21	0.20	0.16	0.13	0.21	0.12	0.19
0.31	0.29	0.23	0.23	0.22	0.18	0.17	0.14	0.12	0.17	0.11	0.16
0.22	0.21	0.17	0.18	0.17	0.14	0.14	0.12	0.11	0.14	0.09	0.13
0.23	0.22	0.19	0.18	0.18	0.15	0.14	0.12	0.10	0.15	0.09	0.14
0.37	0.35	0.25	0.26	0.25	0.19	0.19	0.15	0.13	0.18	0.11	0.17
0.20	0.19	0.17	0.16	0.16	0.13	0.13	0.11	0.10	0.14	0.09	0.13
0.56	0.52	0.32	0.34	0.32	0.24	0.23	0.17	0.14	0.23	0.12	0.21
0.41	0.39	0.27	0.28	0.27	0.21	0.20	0.15	0.13	0.21	0.12	0.18
0.49	0.46	0.30	0.32	0.30	0.23	0.22	0.16	0.14	0.22	0.12	0.20
0.36	0.34	0.26	0.26	0.24	0.19	0.19	0.15	0.13	0.18	0.11	0.17
0.18	0.17	0.15	0.15	0.14	0.13	0.12	0.10	0.09	0.13	0.08	0.12
0.34	0.32	0.25	0.25	0.24	0.19	0.18	0.14	0.12	0.18	0.11	0.17
0.15	0.14	0.13	0.13	0.12	0 11	0.11	0.09	0.08	0.11	0.08	0.10

[c]A waterproof (not vaporproof) membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

Table 4. Coefficients of Transmission (U) of Masonry Walls with Various Types of Veneers[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph

Typical Construction	Type of Wall: Facing	Type of Wall: Backing	Wall No.
	4 in. Brick Veneer[d]	6 in. Hollow Tile[c] 8 in. Hollow Tile[c] 10 in. Hollow Tile[c] 12 in. Hollow Tile[c]	37 38 39 40
	4 in. Brick Veneer[d]	6 in. Concrete 10 in. Concrete 16 in. Concrete	41 42 43
	4 in. Brick Veneer[d]	8 in. **Cinder Blocks**	44
		8 in. **Cinder Blocks** — Cores filled with granulated cork, 5.12 lb per cu ft.	45
		12 in. **Cinder Blocks**	46
		12 in. **Cinder Blocks** — Cores filled with granulated cork, 5.24 lb per cu ft.	47
		8 in. **Concrete Blocks**	48
		8 in. **Concrete Blocks**—Cores filled with granulated cork, 5.14 lb per cu ft.	49
		12 in. **Concrete Blocks**	50
		8 in. **Burned Clay aggregate Block**	51
		8 in. **Burned Clay aggregate Block**—Cores filled with granulated cork, 5.06 lb per cu ft.	52
		12 in. **Burned Clay aggregate Block**	53
		12 in. **Burned Clay aggregate Block**—Cores filled with granulated cork, 5.6 lb per cu ft.	54
	4 in. Cut-Stone Veneer[d]	8 in. Common Brick 12 in. Common Brick 16 in. Common Brick	55 56 57
	4 in. Cut-Stone Veneer[d]	6 in. Hollow Tile[c] 8 in. Hollow Tile[c] 10 in. Hollow Tile[c] 12 in. Hollow Tile[c]	58 59 60 61
	4 in. Cut-Stone Veneer[d]	6 in. Concrete 10 in. Concrete 16 in. Concrete	62 63 64

[a]Computed from factors marked by * in Table 2.
[b]Based on the actual thickness of 2-in. furring strips.
[c]The 6-in., 8-in. and 10-in. tile figures are based on two cells in the direction of heat flow. The 12-in. tile is based on three cells in the direction of heat flow.

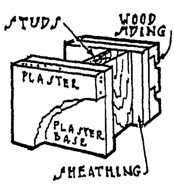
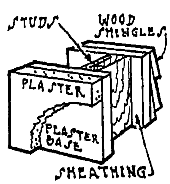
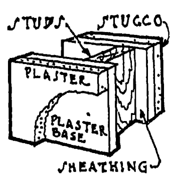
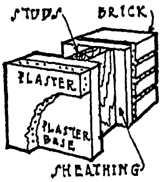
INTERIOR FINISH

Uninsulated Walls						Insulated Walls					
Plain walls—no interior finish	Plaster (½ in.) on walls	Plaster on wood lath—furred	Plaster (¾ in.) on metal lath—furred	Plaster (½ in.) on plasterboard (⅜ in.)—furred	No plaster—decorated rigid or building board interior finish (½ in.)—furred	Plaster (½ in.) on rigid insulation (½ in.)—furred	Plaster (½ in.) on rigid insulation (1 in.)—furred	Plaster (½ in.) on cork board (1½ in.) set in cement mortar (½ in.)	Plaster (½ in.) on plasterboard (⅜ in.)—air space faced on one side with bright aluminum foil cemented to plasterboard	Plaster (¾ in.) on metal lath attached to furring strips (2 in.[b])—rock wool fill (1⅝ in.[b])[f]	Plaster (¾ in.) on metal lath attached to furring strips (2 in.[b])—flexible insulation (½ in.) between furring strips (one air space)
A	B	C	D	E	F	G	H	I	J	K	L
0.36	0.34	0.24	0.25	0.24	0.19	0.19	0.15	0.13	0.18	0.11	0.17
0.34	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.18	0.11	0.17
0.34	0.32	0.23	0.24	0.23	0.19	0.18	0.14	0.12	0.18	0.11	0.17
0.27	0.26	0.20	0.21	0.20	0.16	0.16	0.13	0.11	0.16	0.10	0.15
0.57	0.53	0.33	0.35	0.33	0.24	0.23	0.17	0.14	0.23	0.13	0.21
0.48	0.45	0.30	0.31	0.30	0.22	0.22	0.16	0.14	0.22	0.12	0.20
0.39	0.37	0.26	0.27	0.26	0.20	0.19	0.15	0.13	0.20	0.11	0.18
0.35	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.18	0.11	0.17
0.20	0.19	0.16	0.16	0.16	0.13	0.13	0.11	0.10	0.13	0.09	0.12
0.31	0.30	0.22	0.23	0.22	0.18	0.17	0.14	0.12	0.17	0.11	0.16
0.18	0.18	0.15	0.15	0.15	0.13	0.12	0.10	0.09	0.13	0.08	0.12
0.44	0.42	0.28	0.30	0.28	0.21	0.21	0.16	0.13	0.21	0.12	0.19
0.34	0.32	0.24	0.25	0.23	0.19	0.18	0.14	0.12	0.18	0.11	0.17
0.40	0.38	0.26	0.28	0.26	0.20	0.20	0.15	0.13	0.20	0.11	0.18
0.31	0.29	0.23	0.23	0.22	0.18	0.17	0.14	0.12	0.17	0.11	0.16
0.17	0.16	0.14	0.14	0.14	0.12	0.12	0.10	0.09	0.12	0.08	0.11
0.29	0.28	0.21	0.22	0.21	0.17	0.17	0.13	0.12	0.17	0.10	0.16
0.14	0.14	0.12	0.12	0.12	0.10	0.10	0.09	0.08	0.10	0.07	0.10
0.37	0.35	0.25	0.26	0.25	0.19	0.19	0.15	0.13	0.19	0.11	0.17
0.28	0.27	0.21	0.21	0.21	0.17	0.16	0.13	0.12	0.17	0.10	0.15
0.23	0.22	0.18	0.18	0.18	0.15	0.14	0.12	0.11	0.15	0.095	0.14
0.37	0.35	0.25	0.26	0.25	0.20	0.19	0.15	0.13	0.19	0.11	0.18
0.36	0.34	0.24	0.25	0.24	0.19	0.19	0.15	0.13	0.18	0.11	0.17
0.35	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.18	0.11	0.17
0.28	0.26	0.20	0.21	0.20	0.17	0.16	0.13	0.11	0.17	0.10	0.15
0.61	0.56	0.34	0.36	0.34	0.25	0.24	0.18	0.15	0.24	0.13	0.22
0.51	0.47	0.31	0.32	0.31	0.23	0.22	0.17	0.14	0.22	0.12	0.20
0.41	0.38	0.26	0.28	0.26	0.20	0.20	0.15	0.13	0.20	0.11	0.18

[d]Calculations include cement mortar (½ in.) between veneer or facing and backing.
[e]Based on one air cell in direction of heat flow.
[f]A waterproof (not vaporproof) membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

TABLE 5. COEFFICIENTS OF TRANSMISSION (U) OF VARIOUS TYPES OF FRAME CONSTRUCTION[a]

These coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

TYPICAL CONSTRUCTION	EXTERIOR FINISH	TYPE OF SHEATHING	WALL No
	Wood Siding or Clapboard	1 in. Wood[d]	65
		25/32 in Rigid Insulation	66
		½ in. Plasterboard	67
	Wood Shingles	1 in Wood[d]	68
		25/32 in Rigid Insulation[e]	69
		½ in Plasterboard[e]	70
	Stucco	1 in Wood[d]	71
		25/32 in. Rigid Insulation	72
		½ in. Plasterboard	73
	Brick[f] Veneer	1 in. Wood[d]	74
		25/32 in. Rigid Insulation	75
		½ in. Plasterboard	76

[a]Computed from factors marked by * in Table 2.

[b]These coefficients may also be used with sufficient accuracy for plaster on wood lath or plaster on plasterboard.

[c]Based on the actual width of 2 by 4-in. studding, namely, 3⅝ in.

INTERIOR FINISH

No Insulation Between Studding								Insulation Between Studding				
Plaster on wood lath on studding	Plaster (3/4 in.) on metal lath on studding	Plaster (1/2 in.) on plasterboard (3/8 in.) on studding	Plaster (1/2 in.) on rigid insulation (1/2 in.) on studding	Plaster (1/2 in.) on rigid insulation (1 in.) on studding	Plaster (1/2 in.) on corkboard (1 1/2 in.) on studding	No plaster—decorated rigid or building board interior finish (1/2 in.)	1 in. wood sheathing,[d] furring strips, plaster (1/2 in.) on wood lath	Plaster (1/2 in.) on plasterboard (3/8 in.)—air space faced on one side with bright aluminum foil cemented to plasterboard	Plaster (3/4 in.) on metal lath[b] on studding—flexible insulation (1/2 in.) between studding and in contact with sheathing	Plaster (3/4 in.) on metal lath[b] on studding—flexible insulation (1/2 in.) between studding—2 air spaces	Plaster (3/4 in.) on metal lath[b] on studding—flexible insulation (1 in.) between studding—2 air spaces	Plaster (3/4 in.) on metal lath[b] on studding—rock wool fill (3 5/8 in.[c]) between studding[g,h]
A	B	C	D	E	F	G	H	I	J	K	L	M
0.25	0.26	0.25	0.19	0.15	0.11	0.19	0.17	0.19	0.17	0.15	0.12	0.072
0.19	0.20	0.19	0.15	0.13	0.10	0.16	0.14	0.15	0.15	0 13	0.10	0.068
0.31	0.33	0.31	0.22	0.17	0.13	0 23	0.19	0.22	0.20	0.17	0 13	0.076
0.25	0.26	0.25	0.19	0.15	0.11	0.19	0.17	0.20	0.17	0.15	0.12	0.072
0.17	0.17	0.17	0.14	0.11	0.092	0.14	0.14	0.15	0.13	0.11	0.094	0.064
0.24	0.25	0.24	0.19	0.15	0.11	0.19	0.19	0.22	0.17	0.15	0.12	0.071
0.30	0.32	0.30	0.22	0.16	0.12	0.22	0.19	0.23	0.20	0.17	0.13	0.076
0.22	0.23	0.22	0.17	0.14	0.11	0.19	0.15	0.17	0.16	0.14	0.11	0.071
0.40	0.43	0.40	0.26	0.19	0.14	0.28	0.22	0.26	0.24	0.20	0.14	0.081
0.27	0.28	0 27	0.20	0.15	0.12	0.21	0.17	0.21	0.18	0.16	0.12	0.074
0.21	0.21	0.21	0.16	0.14	0.10	0.17	0.15	0.16	0.15	0.13	0.11	0.068
0.35	0.37	0.35	0.24	0.18	0.13	0.25	0.21	0.24	0.22	0 18	0.14	0.079

[d]Yellow pine or fir—actual thickness about 25/32 in.

[e]Furring strips between wood shingles and sheathing.

[f]Small air space and mortar between building paper and brick veneer neglected.

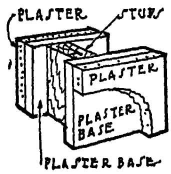
[g]A waterproof (not vaporproof) membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

[h]The coefficients in this column are corrected for the effect of studs.

TABLE 6. COEFFICIENTS OF TRANSMISSION (U) OF FRAME INTERIOR WALLS AND PARTITIONS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION



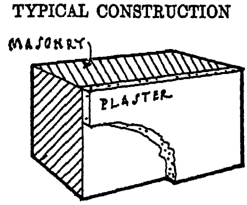
TYPICAL CONSTRUCTION	WALL No.	SINGLE PARTITION (FINISH on ONE SIDE OF STUDDING)	DOUBLE PARTITION (FINISHED ON BOTH SIDES OF STUDDING)			
			Air Space Between Studding	Flaked Gypsum Fill[b] Between Studding	Rock Wool Fill[b] Between Studding	½ in. Flexible Insulation Between Studding (One Air Space)
TYPE OF WALL		A	B	C	D	E
Wood Lath and Plaster On Studding	77	0.62	0.34	0.11	0.076	0.21
Metal Lath and Plaster[c] On Studding	78	0.69	0.39	0.11	0.078	0.23
Plasterboard (⅜ in.) and Plaster[d] On Studding	79	0.61	0.34	0.10	0.075	0.21
Plasterboard (⅜ in.) and Plaster[d] On Studding—bright aluminum foil cemented to plasterboard on surface nailed to studding	80	0.42	0.24			0.16
½ in. Rigid Insulation and Plaster[d] On Studding	81	0.35	0.18	0.083	0.063	0.14
1 in. Rigid Insulation and Plaster[d] On Studding	82	0.23	0.12	0.066	0.054	0.097
1½ in. Corkboard and Plaster[d] On Studding	83	0.16	0.081	0.052	0.044	0.070
2 in. Corkboard and Plaster[d] On Studding	84	0.12	0.063	0.045	0.038	0.057

[a]Computed from factors marked by * in Table 2.
[b]Thickness assumed 3⅝ in.
[c]Plaster on metal lath assumed ¾ in. thick.
[d]Plaster assumed ½ in. thick.

TABLE 7. COEFFICIENTS OF TRANSMISSION (U) OF MASONRY PARTITIONS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION



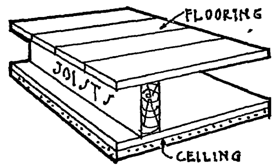
TYPICAL CONSTRUCTION	No.	PLAIN WALLS (NO PLASTER)	WALLS PLASTERED ON ONE SIDE	WALLS PLASTERED ON BOTH SIDES
TYPE OF WALL		A	B	C
4-in. Hollow Clay Tile	85	0.45	0.42	0.40
4-in. Common Brick	86	0.50	0.46	0.43
4-in. Hollow Gypsum Tile	87	0.30	0.28	0.27
2-in. Solid Plaster	88	------	------	0.53

[a]Computed from factors marked by * in Table 2.

TABLE 8. COEFFICIENTS OF TRANSMISSION (U) OF FRAME CONSTRUCTION FLOORS AND CEILINGS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION



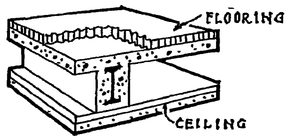
Type of Ceiling	Insulation Between Joists	No.	Type of Flooring: No Flooring (A)	Yellow Pine Flooring[b] on Joists (B)	Yellow Pine Flooring on Rigid Insulation (½ in.) on Joists (C)	Maple or Oak Flooring[c] on Yellow Pine Sub-Flooring[b] on Joists (D)	¼ in. Battleship Linoleum on Yellow Pine Flooring[b] (E)
No Ceiling	None	1	....	0.46	0.27	0.34	0.34
Metal Lath and Plaster (¾ in.)	None	2	0.69	0.30	0.21	0.25	0.25
Wood Lath and Plaster	None	3	0.62	0.28	0.20	0.24	0.24
Plasterboard (⅜ in.) and Plaster (½ in.)	None	4	0.61	0.28	0.20	0.24	0.23
Rigid Insulation (½ in.) and Plaster (½ in.)	None	5	0.35	0.21	0.16	0.18	0.18
Rigid Insulation (1 in.) and Plaster (½ in.)	None	6	0.23	0.16	0.13	0.14	0.14
Plasterboard (⅜ in.) and Plaster (½ in.)	Bright Aluminum Foil[e]	7	0.53	0.21	0.16	0.18	0.18
Metal Lath and Plaster	Flexible[d] Insulation (1 in.)	8	0.17	0.13	0.11	0.12	0.12
Metal Lath and Plaster	Flexible[d] Insulation (2 in.)	9	0.10	0.086	0.076	0.081	0.081
Metal Lath and Plaster	Rock Wool Fill (3⅝ in.)	10	0.079	0.068	0.063	0.066	0.066
Corkboard (1½ in.) and Plaster (½ in.)	None	11	0.16	0.12	0.10	0.11	0.11
Corkboard (2 in.) and Plaster (½ in.)	None	12	0.12	0.10	0.087	0.094	0.094

[a]Computed from factors marked by * in Table 2.
[b]Thickness assumed to be 25/32 in.
[c]Thickness assumed to be 13/16 in
[d]Based on one air space with no flooring, and two air spaces with flooring. The value of U will be the same if insulation is applied to under side of joists and separated from lath and plaster ceiling by 1 in. furring strips.
[e]Bright aluminum foil cemented to plasterboard

TABLE 9. COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE CONSTRUCTION FLOORS AND CEILINGS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION



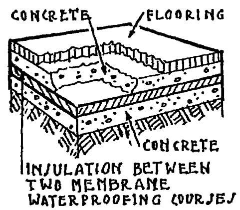
Type of Ceiling	Thickness of Concrete (Inches)	No.	Type of Flooring: No Flooring (Concrete Bare)[b]	Yellow Pine Flooring[c] on Wood Sleepers Embedded in Concrete[d]	Maple or Oak Flooring[e] on Yellow Pine Sub-Flooring[c] on Wood Sleepers Embedded in Concrete	Tile or Terrazzo[f] Flooring on Concrete	1/8 in Battleship Linoleum Directly on Concrete
			A	B	C	D	E
No Ceiling	4	1	0.65	0.40	0.31	0.61	0.44
	6	2	0.59	0.37	0.30	0.56	0.41
	8	3	0.53	0.35	0.28	0.51	0.38
	10	4	0.49	0.33	0.27	0.47	0.36
1/2 in. Plaster Applied Directly to Under Side of Concrete	4	5	0.59	0.38	0.30	0.56	0.41
	6	6	0.54	0.35	0.28	0.52	0.38
	8	7	0.50	0.33	0.27	0.47	0.36
	10	8	0.45	0.32	0.26	0.44	0.34
Suspended or Furred Metal Lath and Plaster (3/4 in.) Ceiling	4	9	0.37	0.28	0.23	0.36	0.29
	6	10	0.35	0.26	0.22	0.34	0.28
	8	11	0.33	0.25	0.21	0.32	0.27
	10	12	0.32	0.24	0.21	0.31	0.25
Suspended or Furred Ceiling of Plasterboard (3/8 in.) and Plaster (1/2 in.)	4	13	0.35	0.26	0.22	0.34	0.28
	6	14	0.33	0.25	0.21	0.32	0.26
	8	15	0.31	0.24	0.21	0.30	0.25
	10	16	0.30	0.23	0.20	0.29	0.24
Suspended or Furred Ceiling of Rigid Insulation (1/2 in.) and Plaster (1/2 in.)	4	17	0.24	0.20	0.17	0.24	0.21
	6	18	0.23	0.19	0.17	0.23	0.20
	8	19	0.22	0.18	0.16	0.22	0.19
	10	20	0.22	0.18	0.16	0.21	0.19
Plaster (1/2 in.) on Corkboard (1 1/2 in.) Set in Cement Mortar (1/2 in.) on Concrete	4	21	0.15	0.13	0.12	0.14	0.14
	6	22	0.14	0.13	0.12	0.14	0.13
	8	23	0.14	0.12	0.11	0.14	0.13
	10	24	0.14	0.12	0.11	0.14	0.13

[a]Computed from factors marked by * in Table 2.
[b]The figures in COLUMN A may be used with sufficient accuracy for concrete floors covered with carpet.
[c]Thickness of yellow pine flooring assumed to be 25/32 in.
[d]The figures in COLUMN B may be used with sufficient accuracy for maple or oak flooring[e] applied directly over the concrete on wood sleepers.
[e]Thickness of maple or oak flooring assumed to be 13/16 in.
[f]Thickness of tile or terrazzo assumed 1 in.

TABLE 10. COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE FLOORS ON GROUND WITH VARIOUS TYPES OF FINISH FLOORING[a, e]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the ground and the air over the floor, and are based on still air (no wind) conditions.

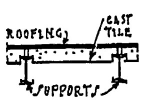
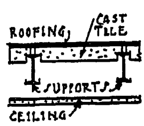
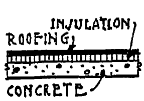
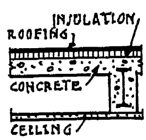
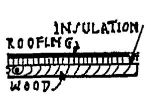
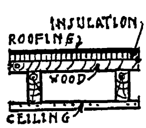
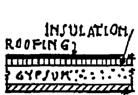
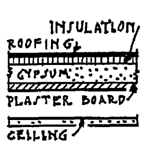
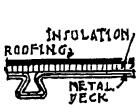
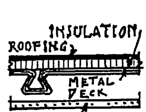
TYPICAL CONSTRUCTION



Type and Thickness of Insulation	Thickness of Concrete (Inches)	No.	TYPE OF FINISH FLOORING				
			No Flooring (Concrete Bare)	Yellow Pine Flooring[b] on Wood Sleepers Resting on Concrete	Maple or Oak Flooring[c] on Yellow Pine Sub-Flooring on Wood Sleepers Resting on Concrete	Tile or Terrazzo[d] on Concrete	¼ in. Battleship Linoleum Directly on Concrete
			A	B	C	D	E
None	4	1	1.07	0.35	0.28	0.98	0.60
	6	2	0.90	0.33	0.27	0.84	0.54
	8	3	0.79	0.32	0.26	0.74	0.50
	10	4	0.70	0.30	0.25	0.66	0.46
None[e]	4	5	0.66	0.29	0.24	0.63	0.44
	8	6	0.54	0.27	0.23	0.52	0.39
1 in. Rigid Insulation[e]	4	7	0.22	0.16	0.14	0.22	0.19
1 in. Rigid Insulation[e]	8	8	0.21	0.15	0.13	0.20	0.18
2 in. Corkboard[e]	4	9	0.12	0.099	0.093	0.12	0.11
2 in. Corkboard[e]	8	10	0.12	0.096	0.090	0.12	0.11

[a]Computed from factors marked by * in Table 2.
[b]Assumed 25/32 in. thick.
[c]Assumed 13/16 in. thick.
[d]Assumed 1 in. thick.
[e]The figures for Nos. 5 to 10, inclusive, include 3 in. cinder concrete placed directly on the ground. The insulation is applied between the cinder concrete and the stone concrete. Usually the insulation is protected on both sides by a waterproof membrane, but this is not considered in the calculations.

Table 11. Coefficients of Transmission (*U*) of Various Types of Flat Roofs Covered with Built-Up Roofing[a]

Typical Construction		Type of Roof Deck	Thickness of Roof Deck (Inches)	No
Without Ceilings	With Metal Lath and Plaster Ceilings[d]			
		**Precast Cement Tile**	1⅝	1
		**Concrete** **Concrete** **Concrete**	2 4 6	2 3 4
		**Wood** **Wood** **Wood** **Wood**	1[b] 1½[b] 2[b] 4[b]	5 6 7 8
		**Gypsum Fiber Concrete[c] (2 in.) on Plasterboard (⅜ in.)** **Gypsum Fiber Concrete[c] (3 in.) on Plasterboard (⅜ in.)** **Gypsum Fiber Concrete[c] (2 in.) on Rigid Insulation Board (½ in.)** **Gypsum Fiber Concrete[c] (2 in.) on Rigid Insulation Board (1 in.)**	2⅜ 3⅜ 2½ 3	9 10 11 12
		**Flat Metal Roofs** Coefficient of transmission of bare corrugated iron (no roofing) is 1.50 Btu per hour per square foot of projected area per degree Fahrenheit difference in temperature, based on an outside wind velocity of 15 mph.	—	13

[a]Computed from factors marked by * in Table 2.
[b]Nominal thicknesses specified—actual thicknesses used in calculations.
[c]Gypsum fiber concrete—87½ per cent gypsum, 12½ per cent wood fiber.

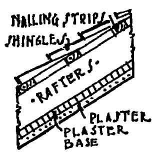
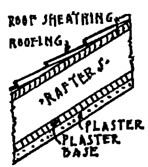
Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 mph.

WITHOUT CEILING—UNDER SIDE OF ROOF EXPOSED								WITH METAL LATH AND PLASTER CEILINGS[d]							
No Insulation	Rigid Insulation (½ in.)	Rigid Insulation (1 in.)	Rigid Insulation (1½ in.)	Rigid Insulation (2 in.)	Corkboard (1 in.)	Corkboard (1½ in.)	Corkboard (2 in.)	No Insulation	Rigid Insulation (½ in.)	Rigid Insulation (1 in.)	Rigid Insulation (1½ in.)	Rigid Insulation (2 in.)	Corkboard (1 in.)	Corkboard (1½ in.)	Corkboard (2 in.)
A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P
0.84	0.37	0.24	0.18	0.14	0.22	0.16	0.13	0.43	0.26	0.19	0.15	0.12	0.18	0.14	0.11
0.82	0.37	0.24	0.17	0.14	0.22	0.16	0.13	0.42	0.26	0.19	0.15	0.12	0.18	0.14	0.11
0.72	0.34	0.23	0.17	0.13	0.21	0.16	0.12	0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11
0.64	0.33	0.22	0.16	0.13	0.21	0.15	0.12	0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11
0.49	0.28	0.20	0.15	0.12	0.19	0.14	0.12	0.32	0.21	0.16	0.13	0.11	0.15	0.12	0.10
0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11	0.26	0.19	0.15	0.12	0.10	0.14	0.11	0.095
0.32	0.22	0.16	0.13	0.11	0.16	0.12	0.10	0.24	0.17	0.14	0.11	0.097	0.13	0.11	0.092
0.23	0.17	0.14	0.11	0.096	0.13	0.11	0.091	0.18	0.14	0.12	0.10	0.087	0.11	0.096	0.082
0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11	0.27	0.19	0.15	0.12	0.10	0.14	0.12	0.097
0.32	0.22	0.16	0.13	0.11	0.15	0.12	0.10	0.23	0.17	0.14	0.11	0.097	0.13	0.11	0.091
0.26	0.19	0.15	0.12	0.10	0.14	0.11	0.10	0.20	0.16	0.13	0.11	0.09	0.12	0.10	0.087
0.19	0.15	0.12	0.10	0.09	0.12	0.10	0.08	0.16	0.13	0.11	0.09	0.08	0.10	0.09	0.077
0.95	0.39	0.25	0.18	0.14	0.23	0.17	0.13	0.46	0.27	0.19	0.15	0.12	0.18	0.14	0.11

[d]These coefficients may be used with sufficient accuracy for wood lath and plaster, or plasterboard and plaster ceilings. It is assumed that there is an air space between the under side of the roof deck and the upper side of the ceiling.

TABLE 12. COEFFICIENTS OF TRANSMISSION (U) OF PITCHED ROOFS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 mph.

TYPICAL CONSTRUCTION	TYPE OF ROOFING AND ROOF SHEATHING	INSULATION BETWEEN ROOF RAFTERS	No	TYPE OF CEILING (APPLIED DIRECTLY TO ROOF RAFTERS)								
				No Ceiling (Rafters Exposed)	Metal Lath and Plaster (¾ in.)	Plasterboard (⅜ in.) and Plaster (½ in.)	Wood Lath and Plaster	Rigid Insulation (½ in.)	Rigid Insulation (½ in.) and Plaster (½ in.)	Rigid Insulation (1 in.) and Plaster (½ in.)	Corkboard (1½ in.) and Plaster (½ in.)	Corkboard (2 in.) and Plaster (½ in.)
				A	B	C	D	E	F	G	H	I
	Wood Shingles on Wood Strips[b]	None	1	0.46	0.30	0.29	0.29	0.22	0.21	0.16	0.12	0.10
		Bright Aluminum Foil[g]	2	....	....	0.21	....	....	....	....	....	....
		1 in Flexible[c]	3	....	0.13	0.12	0.12	0.11	0.11	0.092	0.078	0.069
		2 in. Flexible[c]	4	....	0.086	0.083	0.083	0.076	0.075	0.068	0.060	0.054
		3⅝ in. Rock Wool[e]	5	....	0.070	0.070	0.070	0.064	0.064	0.057	0.052	0.047
	Asphalt Shingles, Rigid Asbestos Shingles, Composition Roofing, or Slate or Tile Roofing[d] on Wood Sheathing[f]	None	6	0.56	0.34	0.32	0.32	0.24	0.23	0.17	0.13	0.11
		Bright Aluminum Foil[g]	7	....	....	0.24	....	....	....	....	....	....
		1 in. Flexible[c]	8	....	0.13	0.13	0.13	0.11	0.11	0.095	0.080	0.071
		2 in. Flexible[c]	9	....	0.088	0.087	0.087	0.079	0.078	0.070	0.062	0.056
		3⅝ in. Rock Wool[e]	10	....	0.073	0.072	0.072	0.067	0.065	0.059	0.053	0.048

[a]Computed from factors marked by * in Table 2. Nos 6 to 10, inclusive, based on ½ in. thick slate.
[b]Based on 1 in. by 4 in. strips spaced 2 in.
[c]Figures based on two air spaces. Insulation may also be applied to under side of roof rafters with furring strips between.
[d]Roofing felt between roof sheathing and slate or tile neglected in calculations.
[e]Assumed 3⅝ in. thick based on the actual width of 2 in. by 4 in. rafters. These coefficients are corrected for the effect of studs.
[f]Sheathing assumed 25/32 in. thick.
[g]Air space faced on one side with bright aluminum foil applied to plasterboard.

TABLE 13. COEFFICIENTS OF TRANSMISSION (U) OF DOORS, WINDOWS, SKYLIGHTS AND GLASS BLOCK WALLS

Coefficients are based on a wind velocity of 15 mph, and are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air inside and outside of the door, window, skylight or wall

Section A. Windows and Skylights

DESCRIPTION	U
Single	1.13[a, c]
Double	0.45[a, e]
Triple	0.281[a, e]

Section B. Solid Wood Doors[b, c]

NOMINAL THICKNESS INCHES	ACTUAL THICKNESS INCHES	U EXPOSED DOOR	U[d] WITH GLASS STORM DOOR
1	25/32	0.69	0.42
1¼	1 1/16	0.59	0.38
1½	1 5/16	0.52	0.35
1¾	1⅜	0.51	0.35
2	1⅝	0.46	0.32
2½	2⅛	0.38	0.28
3	2⅝	0.33	0.25

Section C. Hollow Glass Block Walls

DESCRIPTION	U STILL AIR BOTH SIDES	U STILL AIR INSIDE, 15 MPH OUTSIDE
Smooth surface glass blocks 7¾ x 7¾ x 3⅞ in. thick	0.40	0.49
Ribbed surface glass blocks 7¾ x 7¾ x 3⅞ in. thick	0.38	0.46

[a]See Heating, Ventilating and Air Conditioning, by Harding and Willard, revised edition, 1932.
[b]Computed using $C = 1.15$ for wood; $f_i = 1.65$ and $f_o = 6.0$.
[c]It is sufficiently accurate to use the same coefficient of transmission for doors containing thin wood panels as that of single panes of glass, namely, 1.13 Btu per hour per square foot per degree difference between inside and outside air temperatures.
[d]These values may also be used with sufficient accuracy for wood storm doors. Neglect storm doors if loose and use values for exposed doors.
[e]Air spaces assumed to be ¾ in. or more in width.

coefficient of transmission of a top floor ceiling, unheated attic space, and pitched roof, per square foot of ceiling area, is as follows:

$$U = \frac{U_r \times U_{ce}}{U_r + \frac{U_{ce}}{n}} \quad (6)$$

where

U = combined coefficient to be used with ceiling area.
U_r = coefficient of transmission of the roof.
U_{ce} = coefficient of transmission of the ceiling.
n = the ratio of the area of the roof to the area of the ceiling.

In selecting the values to be used for U_r and U_{ce} it should be noted that the under surface of the roof and the upper surface of the ceiling are more nearly equivalent to the boundary surfaces of an internal air space than they are to the external surfaces of a wall. It would be more nearly correct to use a value of 2.2 rather than the usual value of 1.65 as coefficients for these surfaces. In most cases this would make only a minor

change in U. It should be noted that the over-all coefficient should be multiplied by the ceiling and not the roof area.

If the unheated attic space between the roof and ceiling has no dormers, windows or vertical wall spaces the combined coefficient may be used for determining the heat loss through the roof construction, attic and top floor ceiling. If the unheated attic contains windows and vertical wall spaces these must be taken into consideration in calculating the roof area and also its coefficient U_r. In this case an approximate value of U_r may be obtained as the summation of the coefficient of each individual section such as the roof, vertical walls or windows times its percentage of total area. This coefficient may then be used with reasonable accuracy in Equation 6. If there are large vertical wall areas, the most accurate procedure is to estimate the attic temperature by means of Equation 1, Chapter 6 and to calculate the heat loss by using the ceiling coefficient only, and the attic temperature instead of the outside temperature.

Effect of Attic Ventilation on Ceiling Heat Loss

Neither the combined coefficient equation in this chapter nor the attic temperature equation (Chapter 6) makes allowance for attic ventilation. The effect of winter attic ventilation is to reduce the attic temperature, thereby increasing the ceiling heat loss. Obviously, if the amount of ventilation were such that the attic temperature would be substantially at outside temperature, then the roof should be neglected and only the top floor ceiling coefficient used. On the other hand, according to recently conducted tests[5], ordinary venting to preclude attic condensation has only a minor effect on the attic temperature, in which case the full value of the roof may be taken into consideration without appreciable error.

Basements and Unheated Rooms

The heat loss through floors into basements and into unheated rooms kept closed may be computed by assuming a temperature for these rooms of 32 F. The coefficients of transmission for concrete floors on ground (Table 10) are based on the assumption that the heat-resisting value of the floor extends downward and stops at the under side of the concrete. It is probable, however, that the dirt underneath has some heat-resistance value extending to a considerable depth, which would result in substantially lower heat transmission coefficients than given in Table 10. This problem is now the subject of research. Additional information on the inside and outside temperatures to be used in heat loss calculations is given in Chapter 6.

CONDENSATION IN BUILDINGS

The water vapor or moisture mixed with the air in buildings will be transmitted through many types of building construction if there is a difference in the vapor pressures on the two sides of the structure. Such water vapor will also condense whenever it comes in contact with surfaces

[5]Methods of Moisture Control and Their Application to Building Construction, by F. B. Rowley, A. B. Algren and C. E. Lund. (University of Minnesota *Engineering Experiment Station Bulletin* No. 17).

or objects at or below the dew-point temperature. Thus two types of condensation problems are encountered in building practice, namely (1) *Surface condensation* or condensation on the interior building surfaces including the walls, ceiling (or roof) and glass, and (2) *Interstitial condensation* or the transmittance of the vapor through the building materials and condensation of the moisture on surfaces or voids within the materials of construction.

Condensation within the construction as well as condensation on the interior surfaces does not necessarily occur in all buildings but only in isolated cases when conditions conducive to such condensation exist. The probability of condensation increases with the relative humidity or vapor pressure and with the temperature difference and, in the case of inter-

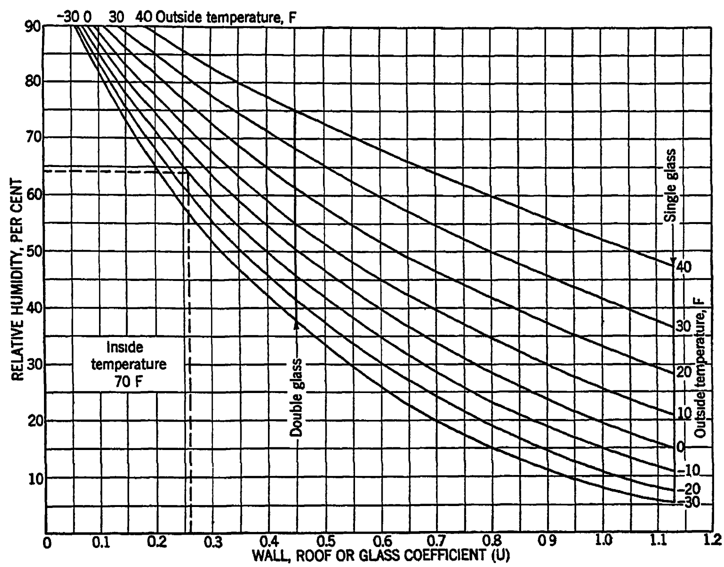


FIG. 2. PERMISSIBLE RELATIVE HUMIDITIES FOR VARIOUS TRANSMISSION COEFFICIENTS

stitial condensation, decreases with the vapor resistance on the warm side of the wall.

Condensation on interior building surfaces[6] (surface condensation) may be eliminated by either reducing the relative humidity or by maintaining the interior surfaces at or above the dew-point temperature. Permissible relative humidities for various wall, roof or glass coefficients and temperature differences may be determined from Fig. 2. The permissible relative humidity for any specific type of construction may be determined by first ascertaining the coefficient of transmission (U) of the construction and then locating this coefficient on the horizontal scale of Fig. 2. A vertical line drawn to the proper outside temperature curve and then to the left hand scale will indicate the permissible relative humidity for the conditions involved. The dotted line shown in Fig. 2 indicates the per-

[6]Permissible Relative Humidities in Humidified Buildings, by Paul D. Close (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1939, p. 766).

missible relative humidity (64 per cent) if surface condensation is to be avoided, for a frame wall having a coefficient of 0.26 and for an outside temperature of −10 F.

Condensation within the construction may likewise be prevented by eliminating the moisture at the source or by providing a barrier on the warm side of the insulation construction. A good vapor barrier construction may be obtained with a vapor-proof paper properly applied under the plaster or a vapor-proof finish on the interior surface of the wall[7]. In the case of attics, the greater the heat resistance in the top floor ceiling, the lower the attic temperature and consequently the greater the tendency for condensation to take place on the under side of the roof boards which moisture will drop on to the ceiling. Thus where thick insulations are installed between ceiling joists, it is desirable to allow openings for outside air circulation through attic space as a precaution against condensation on the underside of the roof even though barriers are used in the ceiling below.

[7]Condensation within Walls, by F. B. Rowley, A. B. Algren and C. E. Lund (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 95).

REFERENCES

A.S.H.V.E. RESEARCH REPORTS:

No. 852—Effects of Air Velocities on Surface Coefficients, by F. B. Rowley, A. B. Algren and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 123).

No. 895—Wind Velocity Gradients Near a Surface and Their Effect on Film Conductance, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 301).

No. 914—Surface Coefficients as Affected by Direction of Wind, by F. B. Rowley and W. A. Eckley (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 33).

No. 915—Conductivity of Concrete, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 47).

No. 964—The Heat Conductivity of Wood at Climatic Temperature Differences, by F. B. Rowley (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 329).

No. 996—Insulating Value of Bright Metallic Surfaces, by F. B. Rowley (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 413).

No. 1026—Thermal Properties of Concrete Construction, by F. B. Rowley, A. B. Algren and Clifford Carlson (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 33).

No. 1048—Thermal Properties of Concrete Construction, by F. B. Rowley, A. B. Algren and Robert Lander (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 33).

Insulating Effect of Successive Air Spaces Bounded by Bright Metallic Surfaces, by L. W. Schad (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 285).

Condensation of Moisture and Its Relation to Building Construction and Operation, by F. B. Rowley, A. B. Algren and C. E. Lund (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 231).

A Theory Covering the Transfer of Vapor Through Materials, by F. B. Rowley (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 545).

Heat Transmission Through Building Materials, by F. B. Rowley and A. B. Algren, *University of Minnesota Engineering Experiment Station Bulletin* No. 8.

Aluminum Foil Insulation (*National Bureau of Standards Letter Circular* No. LC535, October 15, 1938).

Effect of Ceiling Insulation upon Summer Comfort, by T. D. Phillips (*National Bureau of Standards*, Report BMS52, July 1, 1940).

Chapter 5

AIR LEAKAGE

Nature of Air Infiltration, Infiltration Through Walls, Window Leakage, Door Leakage, Selection of Wind Velocity, Crack Length used for Computations, Multi-Story Buildings, Heat Equivalent of Air Infiltration

AIR leakage losses are those resulting from the displacement of heated air in a building by unheated outside air, the interchange taking place through various apertures in the building, such as cracks around doors and windows, fireplaces and chimneys. This leakage of air must be considered in heating and cooling calculations. (See Chapters 6 and 7.)

NATURE OF AIR INFILTRATION

The natural movement of air through building construction is due to two causes. One is the pressure exerted by the wind; the other is the difference in density of outside and inside air because of differences in temperature.

The wind causes a pressure to be exerted on one or two sides of a building. As a result, air comes into the building on the windward side through cracks or porous construction, and a similar quantity of air leaves on the leeward side through like openings. In general the resistance to air movement is similar on the windward to that on the leeward side. This causes a building up of pressure within the building and a lesser air leakage than that experienced in single wall tests as determined in the laboratory. It is assumed that actual building leakages owing to this building up of pressure will be 80 per cent of laboratory test values. While there are cases where this is not true, tests in actual buildings substantiate the factor for the general case. Mechanical ventilating systems are frequently designed to produce positive or negative pressures in an enclosure which are greater or lower than prevalent wind pressures. In such designs, if the rate at which air is specified to be introduced to or removed from the enclosure by positive means exceeds the infiltration rate, it is common practice to use the greater value in determining the heating capacity to warm the outside air.

The air exchange owing to temperature difference, inside to outside, is not appreciable in low buildings. In tall, single story buildings with openings near the ground level and near the ceiling, this loss must be considered. Also in multi-story buildings it is a large item unless the sealing between various floors and rooms is quite perfect. This temperature effect is a *chimney action*, causing air to enter through openings at lower levels and to leave at higher levels.

A complete study of all of the factors involved in air movement through building constructions would be very complex. Some of the complicating factors are: the variations in wind velocity and direction; the exposure of the building with respect to air leakage openings and with respect to adjoining buildings; the variations in outside temperatures as influencing the chimney effect; the relative area and resistance of openings on the windward and leeward sides and on the lower floors and on the upper floors; the influence of a planned air supply and the related outlet vents; and the variation from the average of individual building units. A study of infiltration points to the need for care in the obtaining of good building construction, or unnecessarily large heat losses will result.

INFILTRATION THROUGH WALLS

Table 1 gives data on infiltration through brick and frame walls. The brick walls listed in this table are walls which show poor workmanship and which are constructed of porous brick and lime mortar. For good workmanship, the leakage through hard brick walls with cement-lime mortar does not exceed one-third the values given. These tests indicate that plastering reduces the leakage by about 96 per cent; a heavy coat of cold water paint, 50 per cent; and 3 coats of oil paint carefully applied, 28 per cent. The infiltration through walls ranges from 6 to 25 per cent of that through windows and doors in a 10-story office building, with imperfect sealing of plaster at the baseboards of the rooms. With perfect sealing the range is from 0.5 to 2.7 per cent or a practically negligible quantity, which indicates the importance of good workmanship in proper sealing at the baseboard. It will be noted from Table 1, that the infiltration through properly plastered walls can be neglected.

The value of building paper when applied between sheathing and shingles is indicated by Fig. 1, which represents the effect on outside construction only, without lath and plaster. The effectiveness of plaster properly applied is no justification for the use of low grade building paper or of the poor construction of the wall containing it. Not only is it

TABLE 1. INFILTRATION THROUGH WALLS[a]

Expressed in cubic feet per square foot per hour

Type of Wall		Wind Velocity, Miles per Hour					
		5	10	15	20	25	30
8½ in. Brick Wall	Plain	1.75	4.20	7.85	12.2	18.6	22.9
	Plastered	0.017	0.037	0.066	0.107	0.161	0.236
13 in. Brick Wall	Plain	1.44	3.92	7.48	11.6	16.3	21.2
	Plastered	0.005	0.013	0.025	0.043	0.067	0.097
Frame Wall, with lath and plaster[b]		0.03	0.07	0.13	0.18	0.23	0.26

[a]The values given in this table are 20 per cent less than test values to allow for building up of pressure in rooms and are based on test data reported in the papers listed at the end of this chapter.

[b]Wall construction: Bevel siding painted or cedar shingles, sheathing, building paper, wood lath and 3 coats gypsum plaster.

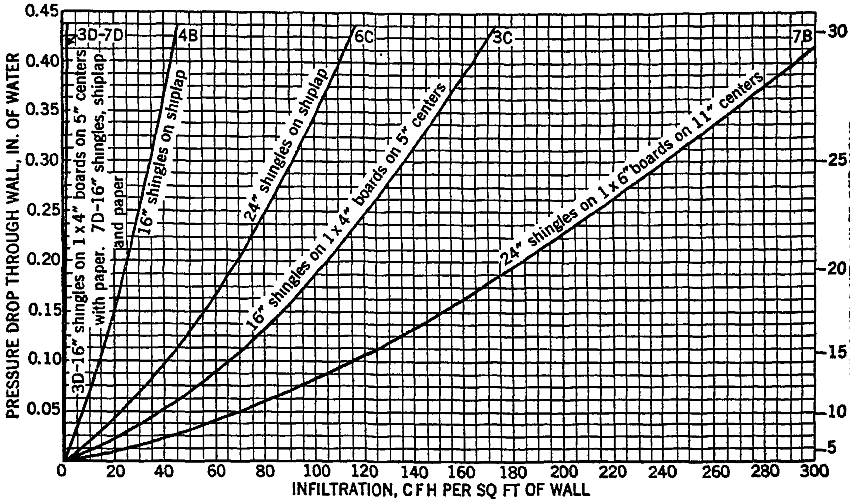


FIG. 1. INFILTRATION THROUGH VARIOUS TYPES OF SHINGLE CONSTRUCTION

difficult to secure and maintain the full effectiveness of the plaster but also it is highly desirable to have two points of high resistance to air flow with an air space between them.

The amount of infiltration that may be expected through single walls used in farm and other shelter buildings, is shown in Fig. 2. The infiltration indicated in Figs. 1 and 2 is that determined in the laboratory and should be multiplied by the factor 0.80 to give proper working values.

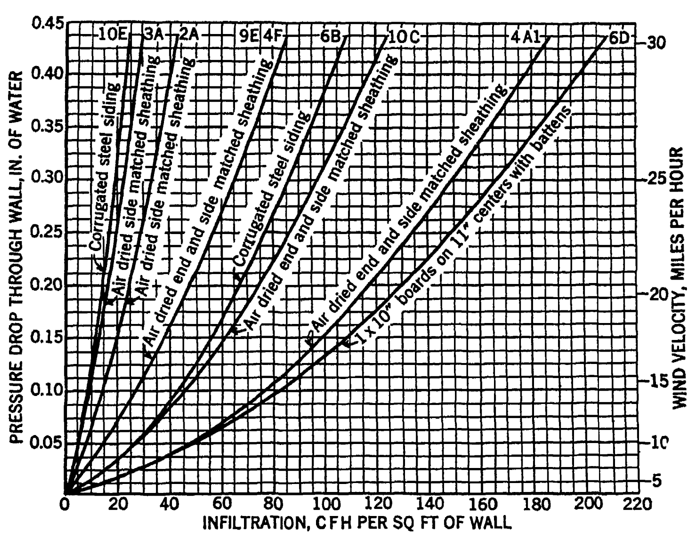


FIG. 2. INFILTRATION THROUGH SINGLE SURFACE WALLS USED IN FARM AND OTHER SHELTER BUILDINGS

Table 2. Infiltration Through Windows

Expressed in Cubic Feet per Foot of Crack per Hour[a]

Type of Window	Remarks	Wind Velocity, Miles per Hour 5	10	15	20	25	30
Double-Hung Wood Sash Windows (Unlocked)	Around frame in masonry wall—not calked[b]	3.3	8.2	14.0	20.2	27.2	34.6
	Around frame in masonry wall—calked[b]	0.5	1.5	2.6	3.8	4.8	5.8
	Around frame in wood frame construction[b]	2.2	6.2	10.8	16.6	23.0	30.3
	Total for average window, non-weatherstripped, 1/16-in. crack and 3/64-in. clearance.[c] Includes wood frame leakage[d]	6.6	21.4	39.3	59.3	80.0	103.7
	Ditto, weatherstripped[d]	4.3	13.0	23.6	35.5	48.6	63.4
	Total for poorly fitted window, non-weatherstripped, 3/32-in. crack and 3/32-in. clearance[e] Includes wood frame leakage[d]	26.9	69.0	110.5	153.9	199.2	249.4
	Ditto, weatherstripped[d]	5.9	18.9	34.1	51.4	70.5	91.5
Double-Hung Metal Windows[f]	Non-weatherstripped, locked	20	45	70	96	125	154
	Non-weatherstripped, unlocked	20	47	74	104	137	170
	Weatherstripped, unlocked	6	19	32	46	60	76
Rolled Section Steel Sash Windows[k]	Industrial pivoted, 1/16-in. crack[g]	52	108	176	244	304	372
	Architectural projected, 1/32-in. crack[h]	15	36	62	86	112	139
	Architectural projected, 3/64-in. crack[h]	20	52	88	116	152	182
	Residential casement, 1/64-in. crack[i]	6	18	33	47	60	74
	Residential casement, 1/32-in. crack[i]	14	32	52	76	100	128
	Heavy casement section, projected, 1/64-in. crack[j]	3	10	18	26	36	48
	Heavy casement section, projected 1/32-in. crack[j]	8	24	38	54	72	92
Hollow Metal, vertically pivoted window[f]		30	88	145	186	221	242

[a]The values given in this table, with the exception of those for double-hung and hollow metal windows, are 20 per cent less than test values to allow for building up of pressure in rooms, and are based on test data reported in the papers listed at the end of this chapter.

[b]The values given for frame leakage are per foot of sash perimeter as determined for double-hung wood windows. Some of the frame leakage in masonry walls originates in the brick wall itself and cannot be prevented by calking. For the additional reason that calking is not done perfectly and deteriorates with time, it is considered advisable to choose the masonry frame leakage values for calked frames as the average determined by the calked and not-calked tests.

[c]The fit of the average double-hung wood window was determined as 1/16-in. crack and 3/64-in. clearance by measurements on approximately 600 windows under heating season conditions.

[d]The values given are the totals for the window opening per foot of sash perimeter and include frame leakage and so-called *elsewhere* leakage. The frame leakage values included are for wood frame construction but apply as well to masonry construction assuming a 50 per cent efficiency of frame calking.

[e]A 3/32-in. crack and clearance represents a poorly fitted window, much poorer than average.

[f]Windows tested in place in building.

[g]Industrial pivoted window generally used in industrial buildings. Ventilators horizontally pivoted at center or slightly above, lower part swinging out.

[h]Architectural projected made of same sections as industrial pivoted except that outside framing member is heavier, and it has refinements in weathering and hardware. Used in semi-monumental buildings such as schools. Ventilators swing in or out and are balanced on side arms 1/32-in crack is obtainable in the best practice of manufacture and installation, 3/64-in. crack considered to represent average practice.

[i]Of same design and section shapes as so-called *heavy section casement* but of lighter weight. 1/64-in. crack is obtainable in the best practice of manufacture and installation, 1/32-in. crack considered to represent average practice.

[j]Made of heavy sections. Ventilators swing in or out and stay set at any degree of opening. 1/64-in. crack is obtainable in the best practice of manufacture and installation, 1/32-in. crack considered to represent average practice.

[k]With reasonable care in installation, leakage at contacts where windows are attached to steel framework and at mullions is negligible. With 3/64-in. crack, representing poor installation, leakage at contact with steel framework is about one-third, and at mullions about one-sixth of that given for industrial pivoted windows in the table.

WINDOW LEAKAGE

The amount of infiltration for various types of windows is given in Table 2. The fit of double-hung wood windows is determined by crack and clearance. Crack thickness is equivalent to one-half the difference between the inside window frame dimension and the outside sash width. The difference between the width of the window frame guide and the sash thickness is considered as the clearance. The length of the perimeter opening or crack for a double-hung window is equal to three times the width plus two times the height, or in other words, it is the outer sash perimeter length plus the meeting rail length. Values of leakage shown in Table 2 for the average double-hung wood window were determined by setting the average measured crack and clearance found in a field survey of a large number of windows on nine windows tested in the laboratory. In addition, the table gives figures for a poorly fitted window. All of the figures for double-hung wood windows are for the *unlocked* condition. Just how a window is closed, or fits when it is closed, has considerable influence on the leakage. The leakage will be high if the sash are short, if the meeting rail members are warped, or if the frame and sash are not fitted squarely to each other. It is possible to have a window with approximately the average crack and clearance that will have a leakage at least double that of the figures shown. Values for the average double-hung wood window in Table 2 are considered to be easily obtainable figures provided the workmanship on the window is good. Should it be known that the windows under consideration are poorly fitted, the larger leakage values should be used. Locking a window generally decreases its leakage, but in some cases may push the meeting rail members apart and increase the leakage. On windows with large clearances, locking will usually reduce the leakage.

Wood casement windows may be assumed to have the same unit leakage as for the average double-hung wood window when properly fitted. Locking, a normal operation in the closing of this type of window, maintains the crack at a low value.

For metal pivoted sash, the length of crack is the total perimeter of the movable or ventilating sections. Frame leakage on steel windows may be neglected when they are properly grouted with cement mortar into brick work or concrete. When they are not properly sealed, the linear feet of sash section in contact with steel work at mullions should be figured at 25 per cent of the values for industrial pivoted windows as given in Table 2.

When storm sash are applied to well fitted windows, very little reduction in infiltration is secured, but the application of the sash does give an air space which reduces the heat transmission and helps prevent the frosting of the windows. When storm sash are applied to poorly fitted windows, a reduction in leakage of 50 per cent may be secured.

DOOR LEAKAGE

Doors vary greatly in fit because of their large size and tendency to warp. For a well fitted door, the leakage values for a poorly fitted double-hung wood window may be used. If poorly fitted, twice this figure should

be used. If weatherstripped, the values may be reduced one-half. A single door which is frequently opened, such as might be found in a store, should have a value applied which is three times that for a well fitted door. This extra allowance is for opening and closing losses and is kept from being greater by the fact that doors are not used as much in the coldest and windiest weather.

The infiltration rate through swinging and revolving doors is generally a matter of judgment by the engineer making cooling load determinations and in the absence of adequate research data the values given in Table 3 represent current engineering practice. These values are based on the average number of persons in a room at a specified time, which may also be the same occupancy assumed for determining the outside ventilation requirements outlined in Chapters 2 and 7.

TABLE 3. INFILTRATION THROUGH OUTSIDE DOORS FOR COOLING LOADS[a]
Expressed in Cubic Feet per Minute per Person in Room

Application	Pair 36 in. Swinging Doors, Single Entrance[b]
Bank	7.5
Barber Shop	4.5
Broker's Office	7.0
Candy and Soda	6.0
Cigar Store	25.0
Department Store	8.0
Dress Shop	2.5
Drug Store	7.0
Furrier	2.5
Hospital Room	3.5
Lunch Room	5.0
Men's Shop	3.5
Office	3.0
Office Building	2.0
Public Building	2.5
Restaurant	2.5
Shoe Store	3.5

[a]For doors located in only one wall or where doors in other walls are of revolving type.

[b]Vestibules with double pair swinging doors, infiltration may be assumed 75 per cent of swinging door values.

Infiltration for 72 in. revolving doors may be assumed 60 per cent of swinging door values.

SELECTION OF WIND VELOCITY

Although all authorities do not agree upon the value of the wind velocity that should be chosen for any given locality, it is common engineering practice to use the average wind velocity during the three coldest months of the year. Average wind velocities for the months of December, January and February for various cities in the United States and Canada are given in Table 2, Chapter 6.

In considering both the transmission and infiltration losses, the more exact procedure would be to select the outside temperature and the wind velocity corresponding thereto, based on Weather Bureau records, which would result in the maximum heat demand. Since the proportion of transmission and infiltration losses varies with the construction and is

different for every building, the proper combination of temperature and wind velocity to be selected would be different for every type of building, even in the same locality. Furthermore, such a procedure would necessitate a laborious cut-and-try process in every case in order to determine the worst combination of conditions for the building under consideration. It would also be necessary to consider heat lag due to heat capacity in the case of heavy masonry walls, and other factors, to arrive at the most accurate solution of the problem. Although heat capacity should be considered wherever possible, it is seldom possible to accurately determine the worst combination of outside temperature and wind velocity for a given building and locality. The usual procedure, with modification explained in Chapter 6, is to select an outside temperature which is not more than 15 F above the lowest recorded, and the average wind velocity during the months of December, January and February.

The direction of prevailing winds may usually be included within an angle of about 90 deg. The windows that are to be figured for prevailing and non-prevailing winds will ordinarily each occupy about one-half the perimeter of the structure, the proportion varying to a considerable extent with the plan of the structure. (See discussion of wind movement in Chapter 42 and Table 2 in Chapter 6.)

CRACK LENGTH USED FOR COMPUTATIONS

In no case should the amount of crack used for computation be less than half of the total crack in the outside walls of the room. Thus, in a room with one exposed wall, take all the crack; with two exposed walls, take the wall having the most crack; and with three or four exposed walls, take the wall having the most crack; but in no case take less than half the total crack. For a building having no partitions, whatever wind enters through the cracks on the windward side must leave through the cracks on the leeward side. Therefore, take one-half the total crack for computing each side and end of the building.

The amount of air leakage is sometimes roughly estimated by assuming a certain number of air changes per hour for each room, the number of changes assumed being dependent upon the type, use and location of the room, as indicated in Table 4. This method may be used to advantage as a check on the calculations made in the more exact manner.

MULTI-STORY BUILDINGS

In tall buildings, infiltration may be considerably influenced by temperature difference or chimney effect which will operate to produce a head that will add to the effect of the wind at lower levels and subtract from it at higher levels. On the other hand, the wind velocity at lower levels may be somewhat abated by surrounding obstructions. Furthermore, the chimney effect is reduced in multi-story buildings by the partial isolation of floors preventing free upward movement, so that wind and temperature difference may seldom cooperate to the fullest extent. Making the rough assumption that the *neutral zone* is located at mid-height of a building, and that the temperature difference is 70 F, the following formulae may be used to determine an equivalent wind velocity

TABLE 4. AIR CHANGES TAKING PLACE UNDER AVERAGE CONDITIONS EXCLUSIVE OF AIR PROVIDED FOR VENTILATION

KIND OF ROOM OR BUILDING	NUMBER OF AIR CHANGES TAKING PLACE PER HOUR
Rooms, 1 side exposed	1
Rooms, 2 sides exposed	1½
Rooms, 3 sides exposed	2
Rooms, 4 sides exposed	2
Rooms with no windows or outside doors	½ to ¾
Entrance Halls	2 to 3
Reception Halls	2
Living Rooms	1 to 2
Dining Rooms	1 to 2
Bath Rooms	2
Drug Stores	2 to 3
Clothing Stores	1
Churches, Factories, Lofts, etc.	½ to 3

to be used in connection with Tables 1 and 2 that will allow for both wind velocity and temperature difference:

$$M_e = \sqrt{M^2 - 1.75\,a} \qquad (1)$$

$$M_e = \sqrt{M^2 + 1.75\,b} \qquad (2)$$

where

M_e = equivalent wind velocity to be used in conjunction with Tables 1 and 2.
M = wind velocity upon which infiltration would be determined if temperature difference were disregarded.
a = distance of windows under consideration from mid-height of building if *above* mid-height.
b = distance if *below* mid-height.

The coefficient 1.75 allows for about one-half the temperature difference head.

For buildings of unusual height, Equation 1 would indicate negative infiltration at the highest stories, which condition may, at times, actually exist.

Sealing of Vertical Openings

In tall, multi-story buildings, every effort should be made to seal off vertical openings such as stair-wells and elevator shafts from the remainder of the building. Stair-wells should be equipped with self-closing doors, and in exceptionally high buildings, should be closed off into sections of not over 10 floors each. Plaster cracks should be filled. Elevator enclosures should be tight and solid doors should be used.

If the sealing of the vertical openings is made effective, no allowance need be made for the chimney effect. Instead, the greater wind movement at the greater heights makes it advisable to install additional heating surface on the upper floors above the level of neighboring buildings, this additional surface being increased as the height is increased. One arbitrary rule is to increase the heating surface on floors above neighboring buildings by an amount ranging from 5 per cent to 20 per cent. This extra heating surface is required only on the windward side and on windy days, and hence automatic temperature control is especially desirable with such installations.

In stair-wells that are open through many floor levels although closed off from the remainder of each floor by doors and partitions, the stratification of air makes it advisable to increase the amount of heating surface at the lower levels and to decrease the amount at higher levels even to the point of omitting all heating surface on the top several floor levels. One rule is to calculate the heating surface of the entire stair-well in the usual way and to place 50 per cent of this in the bottom third, the normal amount in the middle third and the balance in the top third.

HEAT EQUIVALENT OF AIR INFILTRATION

Sensible Heat Loss

The heat required to warm cold outside air, which enters a room by infiltration, to the temperature of the room is given by the equation:

$$H_s = 0.24\, Q\, d\, (t_i - t_o) \tag{3}$$

where

H_s = heat required to raise temperature of air leaking into building from t_o to t_i Btu per hour.
0.24 = specific heat of air.
Q = volume of outside air entering building, cubic feet per hour.
d = density of air at temperature t_o, pounds per cubic foot.
t_i = room air temperature, degrees Fahrenheit.
t_o = outside air temperature, degrees Fahrenheit.

Latent Heat Loss

When it is intended to add moisture to air leaking into a room for the maintenance of proper winter comfort conditions, it is necessary to determine the heat equivalent to evaporate the required amount of water vapor, which may be calculated by the equation:

$$H_l = Q\, d \left(\frac{M_i - M_o}{7000}\right) L \tag{4}$$

where

H_l = heat required to increase moisture content of air leaking into building from M_o to M_i, Btu per hour.
Q = volume of outside air entering building, cubic feet per hour.
d = density of air at temperature t_i, pounds per cubic foot.
M_i = vapor density of inside air, grains per pound of dry air.
M_o = vapor density of outside air, grains per pound of dry air.
L = latent heat of vapor at M_i, Btu per pound.

It is sufficiently accurate to use d = 0.075 lb, in which case Equation 3 reduces to 5 and if the latent heat of vapor is assumed for general conditions as 1060 Btu per pound Equation 4 reduces to 6.

$$H_s = 0.018\, Q\, (t_i - t_o) \tag{5}$$

$$H_l = 0.0114\, Q\, (M_i - M_o) \tag{6}$$

Changing the temperature and vapor subscripts in Equations 5 and 6 to $(t_o - t_i)$ and $(M_o - M_i)$ permits the use of these same formulae for determining the sensible and latent heat gains due to infiltration in cooling load computations.

If a building has more than one room which is divided by interior walls or partitions, it is sufficiently accurate to use half of the total infiltration losses for determining the total heat requirements. Where buildings

have no interior walls, the infiltration losses are calculated by using one-half of the total crack, in which case the entire infiltration loss should be considered.

REFERENCES

A.S.H.V.E. RESEARCH REPORTS:

No. 686—Air Leakage, by F. C. Houghten and C. C. Schrader (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 105).

No. 704—Air Leakage Around Window Openings, by C. C. Schrader (A.S H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 313).

No. 786—Infiltration Through Plastered and Unplastered Brick Walls, by F. C. Houghten and Margaret Ingels (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 377).

No. 803—Air Leakage on Metal Windows in a Modern Office Building, by F. C. Houghten and M. E. O'Connell (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 321).

No. 815—Air Leakage Through a Pivoted Metal Window, by F. C. Houghten and M. E. O'Connell (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 519).

No. 826—Air Infiltration Through Various Types of Brick Wall Construction, by G. L. Larson, D. W. Nelson and C. Braatz (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 183).

No. 851—Air Infiltration Through Various Types of Wood Frame Construction, by G. L. Larson, D. W. Nelson and C. Braatz (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 99).

No. 909—Air Infiltration Through Double-Hung Wood Windows, by G. L. Larson, D. W. Nelson and R. W. Kubasta (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 571).

No. 936—Investigation of Air Outlets in Class Room Ventilation, by G. L. Larson, D. W. Nelson and R. W. Kubasta (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 463).

No. 994—Wind Velocities Near a Building and Their Effect on Heat Loss, by F. C. Houghten, J. L. Blackshaw and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 387).

No. 1069—Heating Requirements of an Office Building as Influenced by the Stack Effect, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 437).

Neutral Zone in Ventilating, by J. E. Emswiler (A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926, p. 59).

Effect of Frame Calking and Storm Sash on Infiltration Around and Through Windows, by W. M. Richtmann and C. Braatz (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 547).

The Weathertightness of Rolled Section Steel Windows, by J. E. Emswiler and W. C. Randall (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 527).

Pressure Differences Across Windows in Relation to Wind Velocity, by J. E. Emswiler and W. C. Randall (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 83).

Flue Action in Tall Buildings, by H. L. Alt (*Heating, Piping and Air Conditioning*, May, 1932).

Air Infiltration Through Steel Framed Windows, by D. O. Rusk, V. H. Cherry and L. Boelter (*Heating, Piping and Air Conditioning*, October, 1932).

Influence of Stack Effect on the Heat Loss in Tall Buildings, by Axel Marin (A.S.H. V.E. TRANSACTIONS, Vol. 40, 1934, p. 377).

Fuel Saving Resulting from the Use of Storm Windows and Doors, by A. P. Kratz and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 87).

The Infiltration Problem of Multiple Entrances, by A. M. Simpson and K. B. Atkinson (A.S.H.V.E. JOURNAL SECTION, *Heating. Piping and Air Conditioning*, June, 1936).

Infiltration Characteristics of Entrance Doors, by A. M. Simpson (*Refrigerating Engineering*, June, 1936).

Chapter 6

HEATING LOAD

Heat Demand Design Factors, Method of Procedure, Inside and Outside Temperatures, Wind Velocity Effects, Auxiliary Heat Sources, Wall Condensation, Heat Loss Computation

TO design any system of heating, the maximum probable heat demand must be accurately estimated in order that the apparatus installed shall be capable of maintaining the desired temperature at all times. The factors which govern this maximum heat demand—most of which are seldom, if ever, in equilibrium—include the following:

1. Outside temperature. 2. Rain or snow. 3. Sunshine or cloudiness. 4. Wind velocity.	*Outside Conditions (The Weather)*
5. Heat transmission of exposed parts of building. 6. Infiltration of air through cracks, crevices and open doors and windows. 7. Heat capacity of materials. 8. Rate of absorption of solar radiation by exposed materials.	*Building Construction*
9. Inside temperatures. 10. Stratification of air. 11. Type of heating system. 12. Ventilation requirements. 13. Period and nature of occupancy. 14. Temperature regulation.	*Inside Conditions*

The *inside conditions* vary from time to time, the physical properties of the *building construction* may change with age, and the *outside conditions* are changing constantly. Just what the worst combination of all of these variable factors is likely to be in any particular case is therefore conjectural. Because of the nature of the problem, extreme precision in estimating heat losses at any time, while desirable, is hard to attain.

The procedure to be followed in determining the heat loss from any building can be divided into seven consecutive steps, as follows:

1. Determine on the inside air temperature, at the breathing line or the 30-in. line, which is to be maintained in the building during the coldest weather. (See Table 1.)

2. Determine on an outside air temperature for design purposes, based on the minimum temperatures recorded in the locality in question, which will provide for all but the most severe weather conditions. Such conditions as may exist for only a few consecutive hours are readily taken care of by the heat capacity of the building itself. (See Table 2.)

3. Select or compute the heat transmission coefficients for outside walls and glass; also for inside walls, floors, or top-floor ceilings, if these are next to unheated space; include roof if next to heated space. (See Chapter 4.)

4. Measure up net outside wall, glass and roof next to heated spaces, as well as any cold walls, floors or ceilings next to unheated space. Such measurements are made from building plans, or from the actual building.

5. Compute the heat transmission losses for each kind of wall, glass, floor, ceiling and roof in the building by multiplying the heat transmission coefficient in each case by the area of the surface in square feet and the temperature difference between the inside and outside air. (See Items 1 and 2.)

6. Select unit values and compute the heat equivalent of the infiltration of cold air taking place around outside doors and windows. These unit values depend on the kind or width of crack and wind velocity, and when multiplied by the length of crack and the temperature difference between the inside and outside air, the result expresses the heat required to warm up the cold air leaking into the building per hour. (See Chapter 5.)

7. The sum of the heat losses by transmission (Item 5) through the outside wall and glass, as well as through any cold floors, ceilings or roof, plus the heat equivalent (Item 6) of the cold air entering by infiltration represents the total heat loss equivalent for any building.

Item 7 represents the heat losses after the building is heated and under stable operating conditions in coldest weather. Additional heat is required for raising the temperature of the air, the building materials and the material contents of the building to the specified standard inside temperature.

The rate at which this additional heat is required depends upon the heat capacity of the structure and its material contents and upon the time in which these are to be heated.

This additional heat may be figured and allowed for as conditions require, but inasmuch as the heating system proportioned for taking care of the heat losses will usually have a capacity about 100 per cent greater than that required for average winter weather, and inasmuch as most buildings may either be continuously heated or have more time allowed

Table 1. Winter Inside Dry-Bulb Temperatures Usually Specified[a]

Type of Building	Deg Fahr	Type of Building	Deg Fahr
Schools		Theaters—	
Class rooms	70–72	Seating space	68–72
Assembly rooms	68–72	Lounge rooms	68–72
Gymnasiums	55–65	Toilets	68
Toilets and baths	70		
Wardrobe and locker rooms	65–68	Hotels—	
Kitchens	66	Bedrooms and baths	70
Dining and lunch rooms	65–70	Dining rooms	70
Playrooms	60–65	Kitchens and laundries	66
Natatoriums	75	Ballrooms	65–68
		Toilets and service rooms	68
Hospitals—			
Private rooms	70–72	Homes	70–72
Private rooms (surgical)	70–80	Stores	65–68
Operating rooms	70–95	Public buildings	68–72
Wards	68	Warm air baths	120
Kitchens and laundries	66	Steam baths	110
Toilets	68	Factories and machine shops	60–65
Bathrooms	70–80	Foundries and boiler shops	50–60
		Paint shops	80

[a]The most comfortable dry-bulb temperature to be maintained depends on the relative humidity and air motion. These three factors considered together constitute what is termed the *effective temperature*. (See Chapter 2.)

for heating-up during the few minimum temperature days, no allowance is made except in the size of boilers or furnaces.

INSIDE TEMPERATURES

The inside air temperature which must be maintained within a building and which should always be stated in the heating specifications is understood to be the dry-bulb temperature at the breathing line, 5 ft above the floor, or the 30-in. line, and not less than 3 ft from the outside walls. Inside air temperatures, usually specified, vary in accordance with the use to which the building is to be put and Table 1 presents values which conform with good practice.

The proper dry-bulb temperature to be maintained depends upon the relative humidity and air motion, as explained in Chapter 2. In other words, a person may feel warm or cool at the same dry-bulb temperature, depending on the relative humidity and air motion. The optimum winter *effective temperature* for sedentary persons, as determined at the A.S.H. V.E. Research Laboratory, is 66 deg.

According to Fig. 6, Chapter 2, for so-called still air conditions, a relative humidity of approximately 50 per cent is required to produce an effective temperature of 66 deg when the dry-bulb temperature is 70 F. However, even where provision is made for artificial humidification, the relative humidity is seldom maintained higher than 40 per cent during the extremely cold weather, and where no provision is made for humidification, the relative humidity may be 20 per cent or less. Consequently, in using the figures listed in Table 1, consideration should be given to whether provision is to be made for humidification, and if so, the actual relative humidity to be maintained.

Temperature at Proper Level: In making the actual heat loss computations, however, for the various rooms in a building it is often necessary to modify the temperatures given in Table 1 so that the air temperature at the proper level will be used. By *air temperature at the proper level* is meant, in the case of walls, the air temperature at the mean height between floor and ceiling; in the case of glass, the air temperature at the mean height of the glass; in the case of roof or ceiling, the air temperature at the mean height of the roof or ceiling above the floor of the heated room; and in the case of floors, the air temperature at the floor level. In the case of heated spaces adjacent to unheated spaces, it will usually be sufficient to assume the temperature in such spaces as the mean between the temperature of the inside heated spaces and the outside air temperature, excepting where attic temperature may be calculated as discussed later or where the combined heat transmission coefficient of the roof and ceiling can be used, in which case the usual inside and outside temperatures should be applied. (See discussion regarding the use of combined coefficients of roofs, attics and top-floor ceilings Chapter 4.

Attic Temperature: It is the practice in many cases to estimate the heat loss through top-floor ceilings by assuming the attic temperature to be the mean between the inside and outside temperatures. In the case of attics with thick insulations between the ceiling joists, the attic temperature will ordinarily be only a few degrees above the outside temperatures rather than the mean between the inside and outside temperatures.

Therefore, calculations based on the latter assumption are likely to be somewhat in error. A formula for calculating attic temperatures is:

$$t_a = \frac{A_c U_c t_i + t_o (A_r U_r + A_w U_w + A_g U_g)}{A_r U_r + A_w U_w + A_g U_g + A_c U_c} \quad (1)$$

where

t_a = attic temperature, degrees Fahrenheit.
t_i = inside temperature near top floor ceiling, degrees Fahrenheit.
t_o = outside temperature, degrees Fahrenheit.
A_c = area of ceiling, square feet.
A_r = area of roof, square feet.
A_w = area of net vertical wall surface, square feet.
A_g = area of glass, square feet.
U_c = coefficient of transmission of ceiling, based on surface coefficient of 2.20 (upper surface, see Chapter 4).
U_r = coefficient of transmission of roof, based on surface coefficient of 2.20 (lower surface, see Chapter 4).
U_w = coefficient of transmission of vertical wall surface.
U_g = coefficient of transmission of glass.

Example 1. Calculate the temperature in an unheated attic, assuming the following conditions: $t_i = 70$; $t_o = 10$; $A_c = 1000$; $A_r = 1200$; $A_w = 100$; $A_g = 10$; $U_r = 0.50$; $U_c = 0.40$; $U_w = 0.30$; $U_g = 1.13$.

Solution: Substituting these values in Equation 1:

$$t_a = \frac{(1000 \times 0.40 \times 70) + 10\,[(1200 \times 0.50) + (100 \times 0.30) + (10 \times 1.13)]}{(1200 \times 0.50) + (100 \times 0.30) + (10 \times 1.13) + (1000 \times 0.40)}$$

$$t_a = \frac{34{,}413}{1041} = 33.1 \text{ F.}$$

High Ceilings: Research data concerning stratification of air in buildings are lacking, but in general it may be said that where the increase in temperature is due to the natural tendency of the warmer or less dense air to rise, as where a direct radiation system is installed, the temperature of the air at the ceiling increases with the ceiling height. The relation, however, is not a straight line function, as the amount of increase per foot of height apparently decreases as the height of the ceiling increases, according to present available information[1].

Where ceiling heights are under 20 ft, it is common engineering practice to consider that the Fahrenheit temperature increases 2 per cent for each foot of height above the breathing line. This rule, sufficiently accurate for most cases, will give the probable air temperature at any given level for a room heated by direct radiation. Thus, the probable temperature in a room at a point 3 ft above the breathing line, if the breathing line temperature is 70 F, will be $[1.00 + (3 \times 0.02)]\ 70 = 74.2$ F.

With certain types of heating and ventilating systems, which tend to oppose the natural tendency of warm air to rise, the temperature differential between floor and ceiling can be greatly reduced. These include unit heaters, fan-furnace heaters, and the various types of mechanical ventilating systems. The amount of reduction is problematical in certain instances, as it depends upon many factors such as location of heaters,

[1]A.S.H.V.E. RESEARCH REPORT No. 958—Temperature Gradient Observations in a Large Heated Space, by G. L. Larson, D. W. Nelson and O. C. Cromer (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 243). A.S.H.V.E. RESEARCH REPORT No 1011—Tests of Three Heating Systems in an Industrial Type of Building, by G. L. Larson, D. W. Nelson and John James (A.S.H V.E. TRANSACTIONS, Vol. 41, 1935, p. 185).

air temperature, and direction and velocity of air discharge. In some cases it has been possible to reduce the temperature between the floor and ceiling by a few degrees, whereas, in other cases, the temperature at the ceiling has actually been increased because of improper design, installation or operation of equipment. So much depends upon the factors enumerated that it is not advisable to allow less than 1 per cent per foot (and usually more) above the breathing line in arriving at the air temperature at any given level for any of these types of heating and ventilating systems, unless the manufacturers are willing to guarantee that the particular type of equipment under consideration will maintain a smaller temperature differential for the specific conditions involved.

Temperature at Floor Level: In determining mean air temperatures just above floors which are next to ground or unheated spaces, a temperature 5 deg lower than the breathing line temperature may be used, provided the breathing line temperature is not less than 55 F.

OUTSIDE TEMPERATURES

The outside temperature used in computing the heat loss from a building is seldom taken as the lowest temperature ever recorded in a given locality. Such temperatures are usually of short duration and are rarely repeated in successive years. It is therefore evident that a temperature somewhat higher than the lowest on record may be properly assumed in making the heat loss computations.

The outside temperature to be assumed in the design of any heating system is ordinarily not more than 15 deg above the lowest recorded temperature as reported by the Weather Bureau during the preceding 10 years for the locality in which the heating system is to be installed. In the case of massive and well insulated buildings in localities where the minimum does not prevail for more than a few hours, or where the lowest recorded temperature is extremely unusual, more than 15 deg above the minimum may be allowed, due primarily to the *fly-wheel* effect of the heat capacity of the structure. The outside temperature assumed and used in the design should always be stated in the heating specifications. Table 2 lists the coldest dry-bulb temperatures ever recorded by the Weather Bureau at the places listed. Recommended design temperatures are given in Column E.

If Weather Bureau reports are not available for the locality in question, then the reports for the station nearest to this locality are to be used, unless some other temperature is specifically stated in the specifications. In computing the average heat transmission losses for the heating season in the United States the average outside temperature from October 1 to May 1 should be used.

WIND VELOCITY EFFECTS

The effect of wind on the heating requirements of any building should be given consideration under two heads:

1. Wind movement increases the heat transmission of walls, glass, and roof, affecting poor walls to a much greater extent than good walls.

2. Wind movement materially increases the infiltration (inleakage) of cold air through the cracks around doors and windows, and even through the building materials themselves, if such materials are at all porous.

TABLE 2. CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS[a]

COL. A	COL. B	COL. C	COL. D	COL. E	COL. F	COL. G
State	City	Average Temperature, Oct. 1-May 1	Lowest Temperature Ever Reported	Recommended Design Temperature	Average Wind Velocity Dec., Jan., Feb., Miles per Hour	Direction of Prevailing Wind, Dec., Jan., Feb.
Ala.	Birmingham	53.8	−10	10	8.5	N
	Mobile	58.9	−1	20	10.4	N
Ariz.	Flagstaff	35.8	−25	−10	7.8	SW
	Phoenix	59.5	12	25	6.4	E
Ark.	Fort Smith	50.4	−15	5	8.1	E
	Little Rock	51.6	−12	10	8.7	NW
Calif.	Los Angeles	58.5	28	30	6.3	NE
	San Francisco	54.2	27	30	7.6	N
Colo.	Denver	38.9	−29	−15	7.5	S
	Grand Junction	38.9	−21	−10	5.3	NW
Conn.	New Haven	38.4	−15	0	9.7	N
D. C.	Washington	43.4	−15	0	7.1	NW
Fla.	Jacksonville	62.0	10	25	9.2	NE
Ga.	Atlanta	51.5	−8	10	12.1	NW
	Savannah	58.5	8	15	9.5	NW
Idaho	Lewiston	42.3	−23	−5	5.3	E
	Pocatello	35.7	−28	−10	9.6	SE
Ill.	Chicago	36.4	−23	−10	12.5	W
	Springfield	39.8	−24	−10	10.1	NW
Ind.	Evansville	45.1	−16	0	9.8	S
	Indianapolis	40.3	−25	−10	11.5	SW
Iowa	Dubuque	33.9	−32	−20	7.1	NW
	Sioux City	32.6	−35	−20	11.6	NW
Kans.	Concordia	39.8	−25	−10	8.1	S
	Dodge City	41.4	−26	−10	9.8	NW
Ky.	Louisville	45.3	−20	−5	9.9	SW
La.	New Orleans	61.6	7	20	8.8	N
	Shreveport	56.2	−5	10	8.9	SE
Me.	Eastport	31.5	−23	−10	12.0	W
	Portland	33.8	−21	−10	9.2	NW
Md.	Baltimore	43.8	−7	10	7.8	NW
Mass.	Boston	38.1	−18	0	11.2	W
Mich.	Alpena	29.6	−28	−10	12.4	W
	Detroit	35.8	−24	−10	12.7	SW
	Marquette	28.3	−27	−10	11.1	NW
Minn.	Duluth	24.3	−41	−30	12.6	SW
	Minneapolis	29.4	−33	−20	11.3	NW
Miss.	Vicksburg	56.8	−1	15	8.3	SE
Mo.	St. Joseph	40.7	−24	−10	9.3	NW
	St. Louis	43.6	−22	−5	11 6	S
	Springfield	44.3	−29	−10	10.8	SE
Mont.	Billings	34.0	−49	−30	....	W
	Havre	27.6	−57	−30	9.5	SW
Nebr.	Lincoln	37.0	−29	−15	10.5	S
	North Platte	35.4	−35	−20	8.5	W
Nev.	Tonopah	39.4	−10	5	10.0	SE
	Winnemucca	37.9	−28	−15	8.7	NE
N. H.	Concord	33.3	−35	−20	6.6	NW
N. J.	Atlantic City	41.6	−9	5	15.9	NW
N. Y.	Albany	35.2	−24	−5	8.1	S
	Buffalo	34.8	−20	0	17.2	W
	New York	40.7	−14	0	17.1	NW
N. M.	Santa Fe	38.3	−13	0	7.8	NE

[a]United States data from U. S. Weather Bureau.
Canadian data from Meteorological Service of Canada.

TABLE 2 CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS[a]—(Concluded)

COL. A	COL. B	COL. C	COL. D	COL E	COL. F	COL. G
State or Province	City	Average Temperature, Oct. 1-May 1	Lowest Temperature Ever Reported	Recommended Design Temperature	Average Wind Velocity Dec., Jan., Feb., Miles per Hour	Direction of Prevailing Wind, Dec., Jan., Feb.
N. C.	Raleigh	50.0	−2	15	8.2	SW
	Wilmington	54.2	5	20	8.5	SW
N. D.	Bismarck	24.6	−45	−30	9.1	NW
	Devils Lake	20.3	−44	−30	10.6	W
Ohio	Cleveland	37.2	−17	−5	13.0	SW
	Columbus	39.9	−20	−10	12.0	SW
Okla.	Oklahoma City	47.9	−17	0	12.0	N
Ore.	Baker	35.2	−24	−15	6.9	SE
	Portland	46.1	−2	10	7.5	S
Pa.	Philadelphia	42.7	−6	0	11.0	NW
	Pittsburgh	41.0	−20	−5	11.7	W
R. I.	Providence	37.2	−17	0	12.8	NW
S. C.	Charleston	57.4	7	15	10.6	SW
	Columbia	54.0	−2	10	8.1	NE
S. D.	Huron	28.2	−43	−25	10.6	NW
	Rapid City	33.4	−34	−20	8.2	W
Tenn.	Knoxville	47.9	−16	0	7.8	SW
	Memphis	51.1	−9	0	9.7	S
Texas	El Paso	53.5	−5	0	10.4	NW
	Ft. Worth	55.2	−8	0	10.4	NW
	San Antonio	60.6	4	10	8.0	NE
Utah	Modena	36.3	−24	−15	8.8	W
	Salt Lake City	40.0	−20	−10	6.7	SE
Vt.	Burlington	31.5	−29	−20	11.8	S
Va.	Lynchburg	46.8	−7	10	7.1	NW
	Norfolk	49.3	2	15	12.5	N
	Richmond	47.0	−3	10	7.9	SW
Wash.	Seattle	44.8	3	15	11.3	SE
	Spokane	37.7	−30	−15	7.1	SW
W. Va.	Elkins	39.4	−28	−10	6.6	W
	Parkersburg	42.6	−27	−10	7.5	SW
Wis.	Green Bay	30.0	−36	−20	10.4	SW
	LaCrosse	31.7	−43	−25	7.3	S
	Milwaukee	33.4	−25	−10	11.5	W
Wyo.	Lander	30.0	−40	−25	5.0	SW
	Sheridan	30.7	−41	−25	6.0	NW
Alta.	Edmonton	23.0	−57	−20	6.5	SW
B. C.	Vancouver	42.0	2	15	4.5	E
	Victoria	43.9	−1.5	15	12.5	N
Man.	Winnipeg	17.5	−47	−30	10.0	NW
N. B.	Fredericton	27.0	−35	−10	9.6	NW
N. S.	Yarmouth	35.0	−12	0	14.2	NW
Ont.	London	32.6	−27	0	10.3	SW
	Ottawa	26.5	−34	−10	8.4	NW
	Port Arthur	22.4	−37	−15	7.8	NW
	Toronto	32.9	−26.5	−5	13.0	SW
P. E. I.	Charlottetown	29.0	−27	−5	9.4	SW
Que.	Montreal	27.8	−29	−10	14.3	SW
	Quebec	24.2	−34	−10	13.6	SW
Sask.	Prince Albert	15.8	−70	−55	5.1	W
Yukon	Dawson	2.1	−68	−50	3.7	S

[a]United States data from U. S. Weather Bureau.
Canadian data from Meteorological Service of Canada.

Theoretically as a basis for design, the most unfavorable combination of temperature and wind velocity should be chosen. It is entirely possible that a building might require more heat on a windy day with a moderately low outside temperature than on a quiet day with a much lower outside temperature. However, the combination of wind and temperature which is the worst would differ with different buildings, because wind velocity has a greater effect on buildings which have relatively high infiltration losses. It would be possible to work out the heating load for a building for several different combinations of temperature and wind velocity which records show to have occurred and to select the worst combination; but designers generally do not feel that such a degree of refinement is justified. Therefore, pending further studies of actual buildings, it is recommended that the average wind movement in any locality during December, January and February be provided for in computing (1) the heat transmission of a building, and (2) the heat required to take care of the infiltration of outside air.

The first condition is readily taken care of, as explained in Chapter 4, by using a surface coefficient f_o for the outside wall surface which is based on the proper wind velocity. In case specific data are lacking for any given locality, it is sufficiently accurate to use an average wind velocity of approximately 15 mph which is the velocity upon which the heat transmission coefficient tables in Chapter 4 are based.

In a similar manner, the heat allowance for infiltration through cracks and walls (Tables 1 and 2, Chapter 5) must be based on the proper wind velocity for a given locality. In the case of *tall buildings* special attention must be given to infiltration factors. (See Chapter 5.)

In the past many designers have used empirical *exposure factors* which were arbitrarily chosen to increase the calculated heat loss on the side or sides of the building exposed to the prevailing winds. It is also possible to differentiate among the various exposures more accurately by calculating the infiltration and transmission losses separately for the different sides of the building, using different assumed wind velocities. Recent investigations show, however, that the wind direction indicated by Weather Bureau instruments does not always correspond with the direction of actual impact on the building walls, due to deflection by surrounding buildings.

The exposure factor, which is still in use by many engineers, is usually taken as 15 per cent, and is added to the calculated heat loss on the side or sides exposed to what is considered the prevailing winter wind. There is a need for actual test data on this point, and pending the time when it can be secured, the question must be left to the judgment of the designing engineer. It should be remembered that the values of U in the tables in Chapter 4 are based on a wind velocity of 15 mph and that the infiltration figures are supposed to be selected from the tables in Chapter 5 to correspond to the wind velocities given in Table 2 of the present chapter.

The *Heating, Piping and Air Conditioning Contractors National Association* has devised a method[2] for calculating the square feet of equivalent direct radiation required in a building. This method makes use of ex-

[2]See Standards of *Heating, Piping and Air Conditioning Contractors National Association.*

posure factors which vary according to the geographical location and the angular situation of the construction in question in reference to prevailing winds and their velocity.

AUXILIARY HEAT SOURCES

The heat supplied by persons, lights, motors and machinery should always be ascertained in the case of theaters, assembly halls, and industrial plants, but allowances for such heat sources must be made only after careful consideration of all local conditions. In many cases, these heat sources should not be allowed to affect the size of the installation at all, although they may have a marked effect on the operation and control of the system. In general, it is safe to say that where audiences are involved, the heating installation must have sufficient capacity to bring the building up to the stipulated inside temperature before the audience arrives. In industrial plants, quite a different condition exists, and heat sources, if they are always available during the period of human occupancy, may be substituted for a portion of the heating installation. In no case should the actual heating installation (exclusive of heat sources) be reduced below that required to maintain at least 40 F in the building.

Electric Motors and Machinery

Motors and the machinery which they drive, if both are located in the room, convert all of the electrical energy supplied into heat, which is retained in the room if the product being manufactured is not removed until its temperature is the same as the room temperature.

If power is transmitted to the machinery from the outside, then only the heat equivalent of the brake horsepower supplied is used. In the first case the Btu supplied per hour $= \dfrac{\text{Motor horsepower}}{\text{Efficiency of motor}} \times 2546$, and in the second case Btu per hour = bhp $\times$ 2546, in which 2546 is the Btu equivalent of 1 hp-hr. In some mills this is the chief source of heating and it is frequently sufficient to overheat the building even in zero weather, thus requiring cooling by ventilation the year round.

The heat (in Btu per hour) from electric lamps is obtained by multiplying the watts per lamp by the number of lamps and by 3.413. One cubic foot of producer gas gives off about 150 Btu per hour; one cubic foot of illuminating gas about 535 Btu per hour; and one cubic foot of natural gas about 1000 Btu per hour. A Welsbach burner averages 3 cu ft of gas per hour and a fish-tail burner, 5 cu ft per hour. For information concerning the heat supplied by persons, see Chapter 2.

In intermittently heated buildings, besides the capacity necessary to care for the normal heat loss which may be calculated according to customary rules, additional capacity should be provided to supply the heat necessary to warm up the cold material of the interior walls, floors, and furnishings. Tests have shown that when a cold building has had its temperature raised to about 60 F from an initial condition of about 0 F, the heat absorbed from the air by the material in the structure may vary from 50 per cent to 150 per cent of the normal heat loss of the building. It is therefore necessary, in order to heat up a cold building within a reasonable length of time, to provide such additional capacity. If the

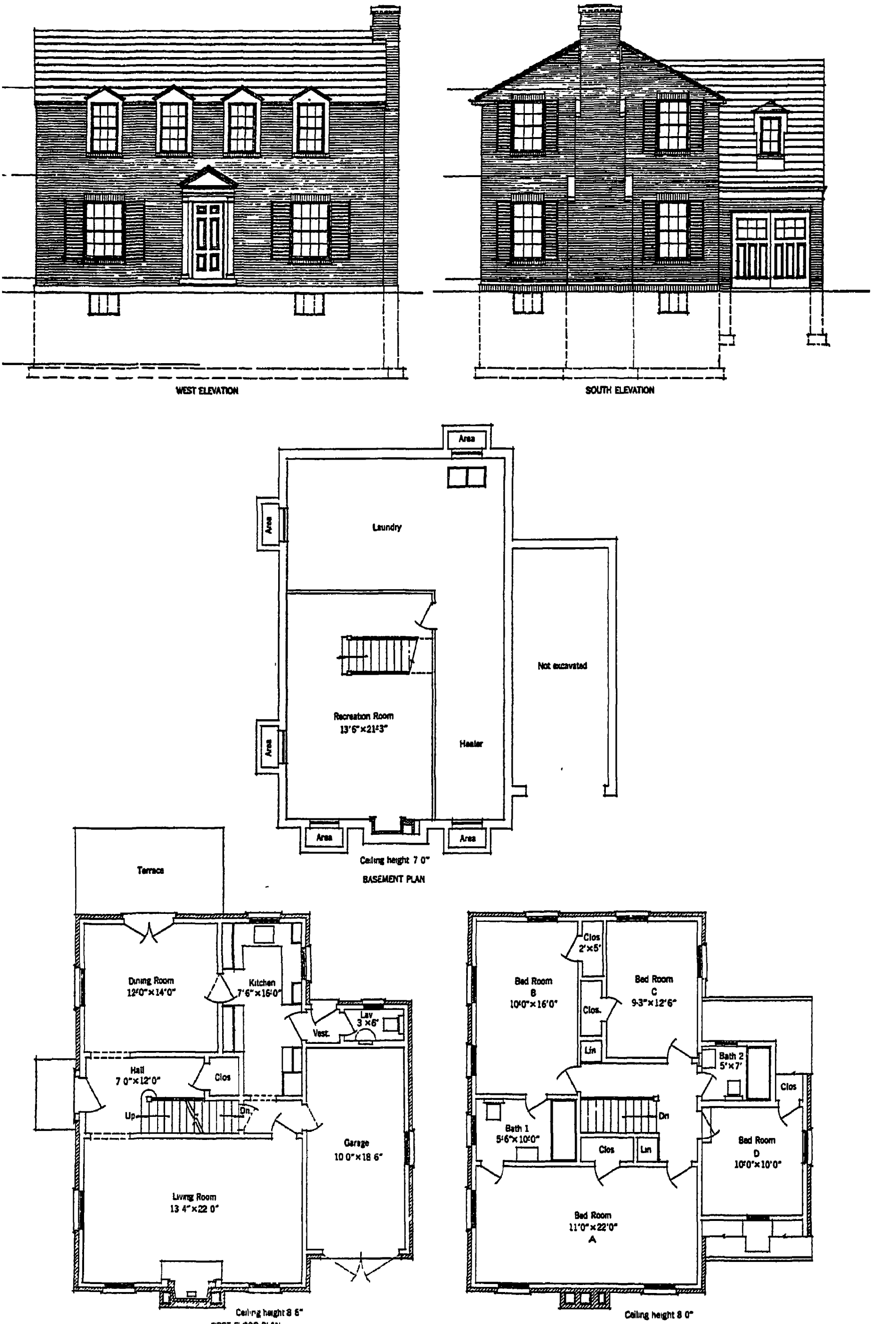


FIG. 1. ELEVATIONS AND FLOOR PLANS OF RESIDENCE

interior material is cold when people enter a building, the radiation of heat from the occupants to the cold material will be greater than is normal and discomfort will result. (See Chapter 2.)

RESIDENCE HEAT LOSS PROBLEMS

Example 2. Calculate the heat loss of residence shown in Fig. 1 located in the vicinity of Chicago. Assume inside and outside design temperatures to be 70 F and −10 F respectively. The attic is unheated. Assume ground temperature to be 45 F. Estimate infiltration by crack method, assuming average wind velocity to be 12.5 mph during December, January and February. No wall, ceiling or roof insulation is to be figured in this problem, but all first and second floor windows are to have storm sash. The building is constructed as follows (transmission coefficients (U) in parentheses):

Walls: Brick veneer, building paper, wood sheathing, studding, metal lath and plaster (0.28). Walls of dormer over garage, same except wood siding in place of brick veneer (0.26).

Attic Walls: Brick veneer, building paper, wood sheathing on studding (0.42).

Basement Walls: 10 in. concrete (0.70).

Roof: Asphalt shingles on wood sheathing on rafters (0.56).

Ceiling: (Second floor): Metal lath and plaster (0.69).

Windows: Double-hung wood windows with storm sash (0.45). Steel casement sash in basement (1.13).

Floor (Bedroom D): Maple finish flooring on yellow pine sub-flooring; metal lath and plaster ceiling below (0.25).

Floor (Basement and Garage): 4 in. stone concrete on 3 in. cinder concrete (0.64).

Solution: The calculations for this problem are given in Table 3, and a summary of the results in Table 4. The values in column F of Table 3 were obtained by multiplying together the figures in columns C, D and E. The heat losses are calculated to the nearest 10 Btu. See reference notes for Table 3 for further explanation of data.

Attention is called to the summary of heat losses (Table 4) of the uninsulated residence (Fig. 1). As storm windows are used in this instance the *glass* and *door* transmission heat losses of 19.5 per cent are relatively small. The infiltration losses (12.0 per cent) are also comparatively small in this case because the storm windows serve substantially the same purpose as weatherstripping. In this problem, the wall, ceiling and floor transmission losses comprise 68.5 per cent of the total. If the building is insulated, the relative heat loss percentages will materially change. (See Example 3 and Table 5.)

Example 3. Calculate the heat loss of residence shown in Fig. 1 based on the same conditions as in Example 2 but insulated throughout as follows (coefficients in parentheses):

Walls: Brick veneer, $^{25}/_{32}$ in. insulation board sheathing, studding, 1 in. insulation board lath and plaster (0.14). Walls of dormer over garage same except wood siding in place of brick veneer (0.13).

Attic Walls: Brick veneer, $^{25}/_{32}$ in. insulation board sheathing on studding (0.28).

Walls Adjoining Garage: Plaster on 1 in. insulation board, studding, metal lath and plaster (0.18).

Basement Walls (Recreation Room): 10 in. concrete, furring strips, ½ in. insulation board (0.26).

Roof: Asphalt shingles on wood sheathing on rafters (0.56).

Ceiling (Second floor): 1 in. insulation board and plaster; ½ in. insulation board on top of ceiling joists (0.15).

Windows: Same as Example 2.

Floor (Bedroom D): Maple finish flooring on yellow pine sub-flooring; ½ in. insulation board and plaster ceiling below (0.18).

Floor (Under Recreation Room): 4 in. stone concrete, 1 in. insulation board and 3 in cinder concrete (0.22).

Solution: The procedure for calculating the heat losses is similar to that for Example 2. A summary of the results is given in Table 5.

TABLE 3. HEAT LOSS CALCULATION SHEET FOR UNINSULATED RESIDENCE (FIG. 1)

A	B	C	D	E	F	G
ROOM OR SPACE	PART OF STRUCTURE	NET AREA OR CRACK LENGTH	COEFFICIENT	TEMP. DIFF [a]	HEAT LOSS (Btu per hour)	TOTALS (Btu per hour)
Bedroom A	Walls	238 sq ft	0.28	80	5330	
	Glass	40 sq ft	0.45	80	1440	
	Infiltration	36 lin ft[b]	0.35[c]	80	1010	
	Ceiling[d]	242 sq ft	0.69	39.8	6660	14,440
Bedroom B and Closet	Walls	156 sq ft	0.28	80	3490	
	Glass	40 sq ft	0.45	80	1440	
	Infiltration	36 lin ft[e]	0.35	80	1010	
	Ceiling[d]	160 sq ft	0.69	39.8	4400	10,340
Bedroom C	Walls	114 sq ft	0.28	80	2560	
	Glass	27 sq ft	0.45	80	970	
	Infiltration	18 lin ft[f]	0.35	80	500	
	Ceiling[d]	120 sq ft	0.69	39.8	3300	7,330
Bedroom D and Closet	Walls	118 sq ft	0.28	80	2650	
	Glass	20 sq ft	0.45	80	720	
	Infiltration	18 lin ft	0.35	80	500	
	Ceiling[d]	120 sq ft	0.69	39.8	3300	
	Floor over Garage	110 sq ft	0.25	35[g]	960[m]	8,130
Bathroom 1	Walls	30 sq ft	0.28	80	670	
	Glass	14 sq ft	0.45	80	500	
	Infiltration	18 lin ft	0.35	80	500	
	Ceiling[d]	55 sq ft	0.69	39.8	1510	3,180
Bathroom 2	Walls	79 sq ft	0.26	80	1770	
	Glass	9 sq ft	0.45	80	320	
	Infiltration	15 lin ft	0.35	80	420	
	Ceiling[d]	35 sq ft	0.69	39.8	960	
	Floor over Garage	35 sq ft	0.25	35	310[m]	3,780
Living Room	Walls	267 sq ft	0.28	80	5980	
	Walls (adjoining garage)	94 sq ft	0.39[h]	35	1280[m]	
	Glass	50 sq ft	0.45	80	1800	
	Infiltration	40 lin ft	0.35	80	1120	10,180
Dining Room	Walls	166 sq ft	0.28	80	3720	
	Glass (doors)	35 sq ft	1.13	80	3160	
	Glass (window)	20 sq ft	0.45	80	720	
	Infiltration[i]	31 lin ft	0.35	80	870	8,470
Kitchen and Entrance to Garage	Walls (outside)	96 sq ft	0.28	80	2150	
	Walls (adjoining garage)	51 sq ft	0.39[h]	35	700[m]	
	Infiltration	27 lin ft	0.35	80	760	
	Glass	18 sq ft	0.45	80	650	
	Door to garage	17 sq ft	0.51	35	300[m]	4,560
Lavette and Vestibule	Walls (outside)	82 sq ft	0.28	80	1840	
	Walls (adjoining garage)	85 sq ft	0.39[h]	35	1160[m]	
	Door	19 sq ft	0.51	80	780	
	Glass	9 sq ft	0.45	80	320	
	Infiltration	19 lin ft	0.35	80	530	4,630

TABLE 3. HEAT LOSS CALCULATION SHEET FOR UNINSULATED RESIDENCE (FIG. 1) (Concluded)

A	B	C	D	E	F	G
ROOM OR SPACE	PART OF STRUCTURE	NET AREA OR CRACK LENGTH	COEFFICIENT	TEMP. DIF.[a]	HEAT LOSS (Btu per hour)	TOTALS (Btu per hour)
Entrance Hall	Walls	39 sq ft	0.28	80	870	
	Door	21 sq ft	0.38	80	640	
	Infiltration	20 lin ft	0.35	80	560	
	Ceiling[d, p]	87 sq ft	0.69	39.8	2490	4,560
Garage	Walls	167 sq ft	0.28	45	2110	
	Glass	53 sq ft	1.13	45	2700	
	Doors	44 sq ft	0.51	45	1010	
	Infiltration	37 lin ft	1.62[j]	45	2700	
	Floor (heat gain)	185 sq ft	0.64[k]	−10[k]	−1180	
	Heat gain				−4710[m]	2,630
Recreation[n] Room	Floor	287 sq ft	0.64	25	4600	
	Walls	220 sq ft	0.70	25	3850	
	Glass	8 sq ft	1.13	80	720	
	Infiltration	8 lin ft	0.76	80	490	9,660
Total						91,890

[a]The inside-outside temperature difference is 70 − (−10) or 80 F, except where otherwise noted.

[b]Only the south windows are used for arriving at the window crack for this room, on the assumption that whatever air enters through the south window cracks will leave through the west window cracks or elsewhere.

[c]Double-hung wood windows with storm sash are assumed to have the same leakage per foot of crack as weatherstripped windows. The air leakage per foot of crack is about 19.5 cu ft per foot of crack for a wind velocity of 12.5 mph. (See Table 2, Chapter 5.) The heat equivalent of the air leakage per hour per degree temperature difference per foot of crack is obtained by multiplying this value by 0.018, or $19.5 \times 0.018 = 0.35$.

[d]In this problem the ceiling heat losses are calculated by estimating the attic temperature and then calculating the loss through the ceiling using the proper temperature difference. This unheated attic is not ventilated during the winter months. The attic temperature is estimated from Equation 1 to be 30.2 F when the outside temperature is −10 F and the room temperature is 70 F. The temperature difference is therefore 70 − 30.2 or 39.8 F.

[e]The window crack in the west wall having two windows is used.

[f]One-half the total crack is used in these rooms.

[g]Temperature in garage assumed to be 35 F.

[h]Coefficient for wall adjoining garage calculated on basis of metal lath and plaster on both sides of studs. ($U = 0.39$)

[i]The door crack is used for estimating the infiltration in this room and as the French doors are weatherstripped the infiltration coefficient is assumed to be the same as in Note b.

[j]The leakage for the garage doors is assumed to be twice that for poorly-fitted double-hung wood windows or about 90 cu ft per foot of crack for a wind velocity of 12.5 mph. The infiltration coefficient is therefore 0.018×90 or 1.62.

[k]The ground temperature is assumed to be 45 F and, as the garage temperature is 35 F, the heat transfer will be from the ground to the garage, and this heat gain should therefore be subtracted from the heat loss. The floor coefficient ($U = 0.64$) is based on 4 in. stone concrete and 3 in. cinder concrete. This coefficient should probably be lower as the ground itself has some heat resistance value. However, complete data are not as yet available.

[m]The heat losses from various rooms into the garage are heat gains for the garage.

[n]Heat is to be provided for the recreation room and this space is therefore figured on the basis of a 70 F temperature. Heat loss into the basement from recreation room is neglected, the calculations being based only on losses through the outside walls, glass and floor. Ground temperature assumed to be 45 F.

[p]The upstairs hall ceiling is included with the downstairs entrance hall because these are connected by means of the stairway. The heat should be provided downstairs.

TABLE 4. SUMMARY OF HEAT LOSSES OF UNINSULATED RESIDENCE

Heat losses given in Btu per hour

Room or Space	Walls	Ceiling and Roof	Floor	Glass and Door	Infiltration	Totals
Bedroom A	5330	6660	....	1440	1010	14,440
Bedroom B	3490	4400	....	1440	1010	10,340
Bedroom C	2560	3300	....	970	500	7,330
Bedroom D	2650	3300	960	720	500	8,130
Bathroom 1	670	1510	....	500	500	3,180
Bathroom 2	1770	960	310	320	420	3,780
Living Room	7260	....	....	1800	1120	10,180
Dining Room	3720	....	....	3880	870	8,470
Kitchen	2850	....	....	950	760	4,560
Lavette	3000	....	....	1100	530	4,630
Entrance Hall	870	2490	....	640	560	4,560
Garage	−1030*	....	−2450†	3410	2700	2,630
Recreation	3850	....	4600	720	490	9,660
Totals	36,990	22,620	3,420	17,890	10,970	91,890
Percentages	40.2	24.6	3.7	19.5	12.0	100.0

*Wall heat loss of 2110 Btu minus wall heat gain of 3140 Btu
†Heat gains; 960, 310 and 1180 Btu.

TABLE 5. SUMMARY OF HEAT LOSSES OF INSULATED RESIDENCE

Heat losses given in Btu per hour

Room or Space	Walls	Ceiling and Roof	Floor	Glass and Door	Infiltration	Totals
Bedroom A	2670	2370	....	1440	1010	7,490
Bedroom B	1750	1570	....	1440	1010	5,770
Bedroom C	1280	1170	....	970	500	3.920
Bedroom D	1320	1170	690	720	500	4,400
Bathroom 1	340	540	....	500	500	1,880
Bathroom 2	820	340	220	320	420	2,120
Living Room	3580	....	....	1800	1120	6,400
Dining Room	1860	....	....	3880	870	6,610
Kitchen	1400	....	....	950	760	3,110
Lavette	1460	....	....	1100	530	3,090
Entrance Hall	440	850	....	640	560	2,490
Garage	−400*	....	−2090†	3410	2700	3,620
Recreation	1430	....	1580	720	490	4,220
Totals	17,950	8,010	400	17,890	10,970	55,220
Percentages	32.5	14.5	0.7	32.4	19.9	100.0

*Wall heat loss of 1050 Btu minus wall heat gains of 590, 320 and 540 Btu.
†Heat gains; 690, 220 and 1180 Btu.

Chapter 7

COOLING LOAD

Design Outside Temperatures, Components of Heat Gain, Normal Heat Transmission, Solar Heat Transmission, Solar Radiation Through Glass, Heat Introduced by Outside Air, Heat Emission of Appliances

LOAD calculations for summer air conditioning are more complicated than heating load calculations because there are more factors to be considered. Due to the variable nature of some of the contributing load components and the fact that they do not necessarily impose their maximum effect simultaneously, considerable care must be used in determining their phase relationship so that equipment of proper capacity may be selected to maintain specified indoor conditions.

The conditions to be maintained in an enclosure are variable and depend upon several factors, especially the outside design conditions, duration of occupancy and relationship between air motion, dry-bulb and wet-bulb temperatures. Information concerning the proper indoor effective temperature to be maintained is given in Chapter 2, for different geographical locations and for various age groups of individuals.

Summer dry-bulb and wet-bulb temperatures of various cities are given in Table 1. The temperatures are not the maximums but the design temperatures which should be used in air conditioning calculations. The maximum outside wet-bulb temperatures as given in Weather Bureau reports usually occur only from 1 to 4 per cent of the time, and they are therefore of such short duration that it is not practical to design a cooling system for them. The temperatures shown in Table 1 are based on available design conditions known to be successfully applied.

COMPONENTS OF HEAT GAIN

A cooling load determination is composed of five components which are classified in the following manner:

1. Normal heat transfer through windows, walls, partitions, doors, floors, ceilings, etc.
2. Transfer of solar radiation through windows, walls, doors, skylights, or roof.
3. Heat emission of occupants within enclosures.
4. Heat introduced by infiltration of outside air or controlled ventilation.
5. Heat emission of mechanical, chemical, gas, steam, hot water and electrical appliances located within enclosures.

TABLE 1. DESIGN DRY- AND WET-BULB TEMPERATURES, WIND VELOCITIES, AND WIND DIRECTIONS FOR JUNE, JULY, AUGUST, AND SEPTEMBER

State	City	Design Dry-Bulb	Design Wet-Bulb	Summer Wind Velocity MPH	Prevailing Summer Wind Direction
Ala.	Birmingham	95	78	5.2	S
	Mobile	95	80	8.6	SW
Ariz.	Phoenix	105	76	6.0	W
Ark.	Little Rock	95	78	7.0	NE
Calif.	Los Angeles	90	70	6.0	SW
	San Francisco	90	65	11.0	SW
Colo.	Denver	95	64	6.8	S
Conn.	New Haven	95	75	7.3	S
Del.	Wilmington	95	78	9.7	SW
D. C.	Washington	95	78	6.2	S
Fla.	Jacksonville	95	78	8.7	SW
	Tampa	94	79	7.0	E
Ga.	Atlanta	95	76	7.3	NW
	Savannah	95	78	7.8	SW
Idaho	Boise	95	65	5.8	NW
Ill.	Chicago	95	75	10.2	NE
	Peoria	95	76	8.2	S
Ind.	Indianapolis	95	76	9.0	SW
Iowa	Des Moines	95	77	6.6	SW
Kansas	Wichita	100	75	11.0	S
Ky.	Louisville	95	76	8.0	SW
La.	New Orleans	95	79	7.0	SW
	Shreveport	100	78	6.2	S
Maine	Portland	90	73	7.3	S
Md.	Baltimore	95	78	6.9	SW
Mass.	Boston	92	75	9.2	SW
Mich.	Detroit	95	75	10.3	SW
Minn.	Minneapolis	95	75	8.4	SE
Miss.	Vicksburg	95	78	6.2	SW
Mo.	Kansas City	100	76	9.5	S
	St. Louis	95	78	9.4	SW
Mont.	Helena	95	67	7.3	SW
Nebr.	Lincoln	95	75	9.3	S
Nev.	Reno	95	65	7.4	W
N. H.	Manchester	90	73	5.6	NW
N. J.	Trenton	95	78	10.0	SW
N. Y.	Albany	92	75	7.1	S
	Buffalo	93	75	12.2	SW
	New York	95	75	12.9	SW
N. M.	Santa Fe	90	65	6.5	SE
N. C.	Asheville	90	75	5.6	SE
	Wilmington	95	79	7.8	SW
N. Dak.	Bismarck	95	73	8.8	NW
Ohio	Cincinnati	95	78	6.6	SW
	Cleveland	95	75	9.9	S
Okla.	Oklahoma City	101	76	10.1	S
Ore.	Portland	90	65	6.6	NW
Pa.	Philadelphia	95	78	9.7	SW
	Pittsburgh	95	75	9.0	NW
R. I.	Providence	93	75	10.0	NW
S. C.	Charleston	95	80	9.9	SW
	Greenville	95	76	6.8	NE
S. Dak.	Sioux Falls	95	75	7.6	S
Tenn.	Chattanooga	95	77	6.5	SW
	Memphis	95	78	7.5	SW

TABLE 1. DESIGN DRY- AND WET-BULB TEMPERATURES, WIND VELOCITIES, AND WIND DIRECTIONS FOR JUNE, JULY, AUGUST, AND SEPTEMBER (Concluded)

State	City	Design Dry-Bulb	Design Wet-Bulb	Summer Wind Velocity MPH	Prevailing Summer Wind Direction
Texas	Dallas	100	78	9.4	S
	El Paso	100	69	6.9	E
	Galveston	95	80	9.7	S
	Houston	95	78	7.7	S
	San Antonio	100	78	7.4	SE
Utah	Salt Lake City	92	63	8.2	SE
Vt.	Burlington	90	73	8.9	S
Va.	Norfolk	95	78	10.9	S
	Richmond	95	78	6.2	SW
Wash.	Seattle	85	65	7.9	S
	Spokane	90	65	6.5	SW
W. Va.	Parkersburg	95	75	5.3	SE
Wis.	Madison	95	75	8.1	SW
	Milwaukee	95	75	10.4	S
Wyo.	Cheyenne	95	65	9.2	S

The components of heat gain, classified by source, are further classified as sensible and latent heat gain.

The first two components fall into the classification of sensible heat gain, that is, they tend to raise the temperature of the air within the structure. The last three components not only produce sensible heat gain but they may also tend to increase the moisture content of the air within the structure.

Normal Heat Transmission

By normal heat transmission, as distinguished from solar heat transmission, is meant the transmission of heat through windows, walls, partitions, etc. from without to interior of enclosure by virtue of difference between outside and inside air temperatures. This load is calculated in a manner similar to that described in Chapter 6 (except that flow of heat is reversed) by means of the formula:

$$H_t = A\,U\,(t_o - t) \qquad (1)$$

where

H_t = heat transmitted through the material of wall, glass, floor, etc., Btu per hour.
A = net inside area of wall, glass, floor, etc., square feet.
t = inside temperature, degrees Fahrenheit.
t_o = outside temperature, degrees Fahrenheit.
U = coefficient of transmission of wall, glass, floor, etc., Btu per hour per square foot per degree Fahrenheit difference in temperature (Tables 3 to 13, Chapter 4).

Solar Heat Transmission

Calculations of the solar heat transmitted through walls and roofs are difficult to determine because of periodic character of heat flow and time lag due to heat capacity of construction.

TABLE 2. SOLAR RADIATION (DIRECT PLUS SKY) IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS AND A HORIZONTAL SURFACE

For 30 Deg North Latitude on August 1

Sun Time	Intensity of Solar Radiation, Btu per Sq Ft per Hour							
	Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal Surface
5:20	0	0	0	0	0	0	0	0
6:00	37	47	23	4.5	4.5	4.5	4.5	11
7:00	119	145	91	11	11	11	11	64
8:00	153	207	149	17	17	17	17	147
9:00	130	194	158	35	21	21	21	213
10:00	86	152	143	63	23.5	23.5	23.5	262
11:00	35	94	85	80	25.5	25.5	25.5	290
12:00	26	26	65	85	65	26	26	300
1:00	25.5	25.5	25.5	80	85	94	35	290
2:00	23.5	23.5	23.5	63	143	152	86	262
3:00	21	21	21	35	158	194	130	213
4:00	17	17	17	17	149	207	153	147
5:00	11	11	11	11	91	145	119	64
6:00	4.5	4.5	4.5	4.5	23	47	37	11
6:40	0	0	0	0	0	0	0	0

TABLE 3. SOLAR RADIATION (DIRECT PLUS SKY) IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS AND A HORIZONTAL SURFACE

For 35 Deg North Latitude on August 1

Sun Time	Intensity of Solar Radiation, Btu per Sq Ft per Hour							
	Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal Surface
5:07	0	0	0	0	0	0	0	0
6:00	43	49	27	4.5	4.5	4.5	4.5	13
7:00	121	151	97	11	11	11	11	72
8:00	147	207	155	25	17	17	17	151
9:00	120	194	169	49	21	21	21	213
10:00	71	152	156	83	23.5	23.5	23.5	245
11:00	28	94	129	103	25.5	25.5	25.5	288
12:00	26	26	84	109	84	26	26	298
1:00	25.5	25.5	25.5	103	129	94	28	288
2:00	23.5	23.5	23.5	83	156	152	71	245
3:00	21	21	21	49	169	194	120	213
4:00	17	17	17	25	155	207	147	151
5:00	11	11	11	11	97	151	121	72
6:00	4.5	4.5	4.5	4.5	27	49	43	13
6:53	0	0	0	0	0	0	0	0

TABLE 4. SOLAR RADIATION (DIRECT PLUS SKY) IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS AND A HORIZONTAL SURFACE
For 40 Deg North Latitude on August 1

SUN TIME	INTENSITY OF SOLAR RADIATION, BTU PER SQ FT PER HOUR							
	Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal Surface
4:50	0	0	0	0	0	0	0	0
5:00	5	6	4	2.5	2.5	2.5	2.5	5
6:00	49	56	32	4.5	4.5	4.5	4.5	20
7:00	123	162	109	11	11	11	11	85
8:00	137	211	166	29	17	17	17	160
9:00	102	195	181	74	21	21	21	212
10:00	54	152	171	103	23.5	23.5	23.5	244
11:00	28	94	144	124	41	25.5	25.5	281
12:00	26	26	98	128	98	26	26	290
1:00	25.5	25.5	41	124	144	94	28	281
2:00	23.5	23.5	23.5	103	171	152	54	244
3:00	21	21	21	74	181	195	102	212
4:00	17	17	17	29	166	211	137	160
5:00	11	11	11	11	109	162	123	85
6:00	4.5	4.5	4.5	4.5	32	56	49	20
7:00	2.5	2.5	2.5	2.5	4	6	5	5
7:10	0	0	0	0	0	0	0	0

TABLE 5. SOLAR RADIATION (DIRECT PLUS SKY) IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS AND A HORIZONTAL SURFACE
For 45 Deg North Latitude on August 1

SUN TIME	INTENSITY OF SOLAR RADIATION, BTU PER SQ FT PER HOUR							
	Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal Surface
4:25	0	0	0	0	0	0	0	0
5:00	22	20	17	3.5	3.5	3.5	3.5	9
6:00	87	99	56	5.5	5.5	5.5	5.5	27
7:00	151	192	134	12	12	12	12	89
8:00	144	237	188	48	17	17	17	156
9:00	100	199	197	93	21	21	21	205
10:00	46	153	184	121	23.5	23.5	23.5	243
11:00	28	94	158	146	63	25.5	25.5	259
12:00	26	26	116	156	116	26	26	281
1:00	25.5	25.5	63	146	158	94	28	259
2:00	23.5	23.5	23.5	121	184	153	46	243
3:00	21	21	21	93	197	199	100	205
4:00	17	17	17	48	188	237	144	156
5:00	12	12	12	12	134	192	151	89
6:00	5.5	5.5	5.5	5.5	56	99	87	27
7:00	3.5	3.5	3.5	3.5	17	20	22	9
7:35	0	0	0	0	0	0	0	0

The variation in radiation intensity on differently oriented surfaces is given in Fig. 1, and in Tables 2, 3, 4 and 5. The greater part of the radiation intensity is always direct radiation from the sun. However, during the time when the sun is shining, any surface receives radiation of a lower intensity coming from all parts of the sky due to reflection and refraction. This scattered radiation intensity was found to vary from a very low value to values as high as 20 per cent of the total radiation observed on certain days in Pittsburgh. The curves and tables are for combined direct solar and scattered sky radiation, and are given to represent expected design radiation intensity for August 1. They were prepared by the A.S.H.V.E. Laboratory from data[1] obtained by pyrheliometer observations.

A study of these curves discloses the periodic relationship and wide variation in solar intensity on various surfaces. It will be observed that

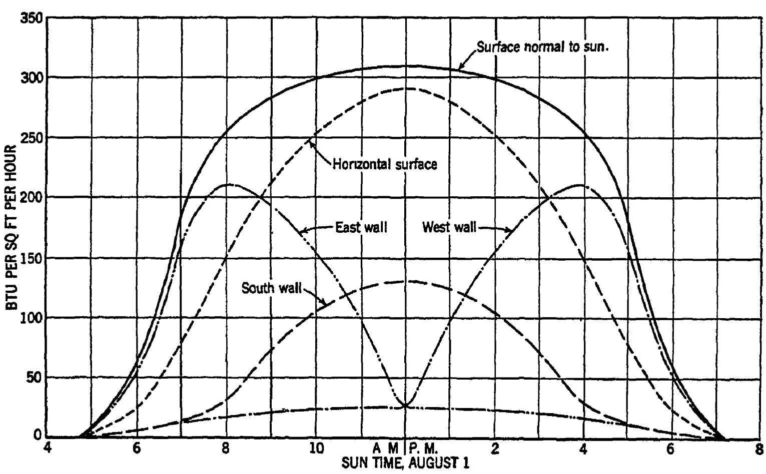


FIG. 1. SOLAR INTENSITY NORMAL TO SUN ON HORIZONTAL SURFACE AND ON WALLS FOR AUGUST 1 AT 40 DEG NORTH LATITUDE

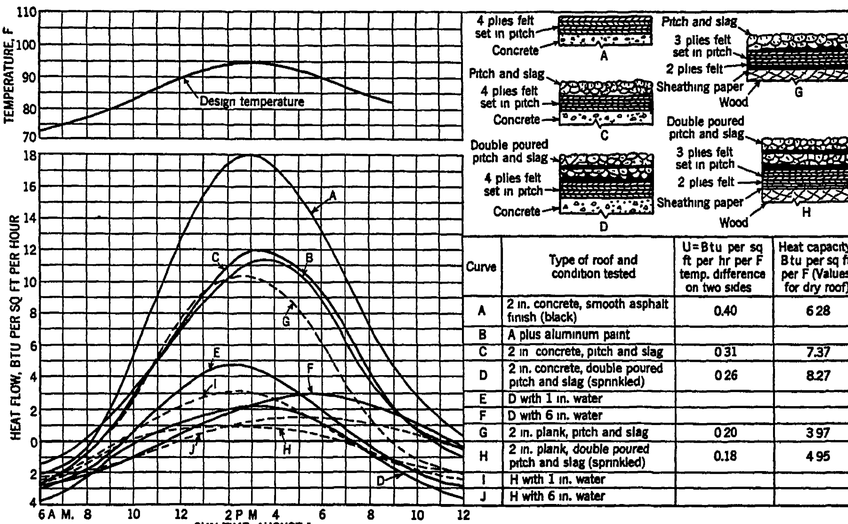
both the roof (horizontal surface) and south wall radiation curves are in exact phase relationship with each other, while those for the east and west walls overlap each other due to scattered sky radiation on the west wall during the forenoon and on the east wall during the afternoon. This phase relationship has an important bearing on the cooling load. Failure to consider the periodic character of heat flow resulting from diurnal movement of the sun and the lag due to heat capacity of the structure, which determine the timing and magnitude of the heat wave flowing through the wall, may result in a large error in load calculations.

The values of solar intensity appearing in Fig. 1 must not be confused with the actual heat transmission through the wall for much of the solar radiation impinging against the outer surface fails to pass through the wall. Instead it is delivered to the outside air by reflection, radiation,

[1]A.S.H.V.E. RESEARCH REPORT No. 1147—Heat Gain Through Glass Blocks by Solar Radiation and Transmittance, by F. C. Houghten, David Shore, H. T. Olson and Burt Gunst (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940).

convection and conduction. A mathematical solution for the determination of solar heat transmission has been developed but the equations involved are too complex for practical application[2].

The heat flow in summer through various types of roofs and walls has been measured by the A.S.H.V.E. Laboratory. The curves in Fig. 2, give the heat flow through the inside surface of roofs[3] with details of the construction of the roofs tested. The condition for which these results are given are: solar radiation for 40 deg north latitude on August 1 as given in Fig. 1 and Table 4; outdoor design temperature reaching a maximum of 95 F as shown by the temperature curve in Fig. 2 and an indoor temperature of 75 F.



Curve	Type of roof and condition tested	U=Btu per sq ft per hr per F temp. difference on two sides	Heat capacity, Btu per sq ft per F (Values for dry roof)
A	2 in. concrete, smooth asphalt finish (black)	0.40	6 28
B	A plus aluminum paint		
C	2 in concrete, pitch and slag	0 31	7.37
D	2 in. concrete, double poured pitch and slag (sprinkled)	0 26	8.27
E	D with 1 in. water		
F	D with 6 in. water		
G	2 in. plank, pitch and slag	0 20	3 97
H	2 in. plank, double poured pitch and slag (sprinkled)	0.18	4 95
I	H with 1 in. water		
J	H with 6 in. water		

FIG. 2. RELATION BETWEEN TIME AND HEAT FLOW THROUGH INSIDE SURFACE OF HORIZONTAL ROOFS CORRECTED TO DESIGN DAY OF AUGUST 1

Curves in Fig. 3 were prepared by the A.S.H.V.E. Laboratory from recent tests made there and show the heat flow through the inside surface of three types of walls for various orientations. The results are given for the following conditions: 90 per cent of the solar radiation given in Fig. 1 and Table 4 for 40 deg north latitude on August 1; outdoor design temperature reaching a maximum of 93 F as shown by the temperature curve in Fig. 3 and an indoor temperature of 78 F and 50 per cent relative humidity.

The heat flow shown in Figs. 2 and 3 is a combination of normal transmission and solar radiation transmission and is the total heat flow through the wall or roof. Due to the heat capacity of walls and roofs there is a

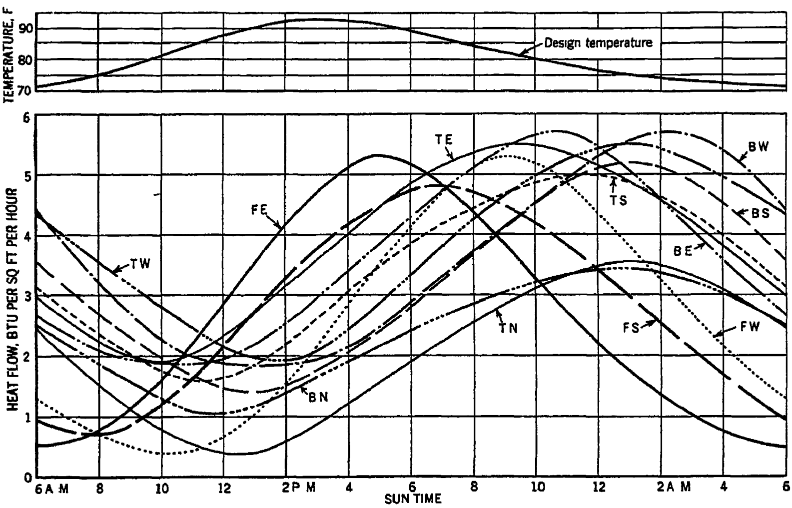
[2] A.S.H.V.E. RESEARCH REPORT No. 923—Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. W. Pugh and Paul McDermott (A.S.H V.E. TRANSACTIONS, Vol. 38, 1932, p. 231).

[3] A.S.H.V.E. RESEARCH REPORT No. 1158—Summer Cooling Load as Affected by Heat Gain Through Dry, Sprinkled and Water Covered Roofs, by F. C. Houghten, H. T. Olson and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940).

ne lag[4] in the transmission of heat through them as shown by the curves. ɪr the types of construction covered in Figs. 2 and 3 and for the con- :ions indicated, the heat flow through the inside surface at any given ne can be read directly. For other types of construction, the curves ɪy be used as a guide in estimating the heat flow. The time lag for ıer types of construction is included in Table 6 which was prepared by ɛ A.S.H.V.E. Laboratory from data collected by it and by other thorities.

ar Radiation Transmitted Through Glass

Windows present a problem somewhat different from that of opaque lls, because they permit a large percentage of the solar energy to pass



. 3. Relation Between Time and Heat Flow Through the Inside Surface of Walls of Different Construction and Orientation on a 93 F Design Day with 90 per cent of Design Solar Radiation

Valls BE, BS, BW and BN—12 in. solid brick and plaster facing east, south, west and north respectively.
Valls TE, TS, TW and TN—4 in. brick veneer, 8 in. tile and plaster facing east, south, west and north ·ctively.
Valls FE, FS and FW—4 in. brick veneer, building paper, ⅞ in., matched sheathing, 2 x 4 in. studs, l lath and plaster facing east, south and west respectively

ɔugh. A small amount is reflected and the balance is absorbed by the ss. The amount absorbed depends upon the character and thickness he glass and the angle between it and the sun's rays. The temperature he glass is raised by the absorbed heat and this heat is then delivered :he air on each side in proportion to the difference between the glass air temperatures[5].

'he A.S.H.V.E. tests[6] indicate that a single pane of double strength ıs 0.127 in. thick absorbs approximately 11 per cent of the solar

oc. Cit. Note 2.

leat Absorbing Glass Windows, by W. W. Shaver (A.S.H.V.E. Transactions, Vol. 41, 1935, p. 287).

.S.H.V.E. Research Report No. 974—Radiation of Energy Through Glass, by J. L. Blackshaw and Houghten (A.S.H.V.E. Transactions, Vol. 40, 1934, p 93). A.S.H.V.E Research Report No. 975 dies of Solar Radiation Through Bare and Shaded Windows by F. C. Houghten, Carl Gutberlet, . L. Blackshaw (A.S.H V E Transactions, Vol. 40, 1934, p. 101).

TABLE 6. TIME LAG IN TRANSMISSION OF SOLAR RADIATION THROUGH WALLS AND ROOFS

Type and Thickness of Wall or Roof	Time Lag, Hours
1-in. yellow pine horizontal roof, water proofing, smooth black finish	1
2-in. yellow pine horizontal roof, water proofing, smooth black finish	1¾
4-in. reinforced clay tile horizontal roof, water proofing, slag finish	2¼
2-in. gypsum horizontal roof, water proofing, slag finish	2¼
Slate and slaters felt on 2¼ in. tongue and grooved yellow pine, sloped roof	2½
4-in. gypsum horizontal roof, water proofing, slag finish	4½
6-in. concrete horizontal roof, water proofing, slag finish	5
1-in. concrete, 4-in. cinders, 1½-in. concrete, water proofing, smooth black finish	8
Wood siding, 1-in. sheathing, 2 x 4 studs, lath and plaster	2
Wood siding, 1-in. sheathing, 2 x 4 studs (studding space filled with insulation) lath and plaster	5
4-in. brick, 1-in. sheathing, 2 x 4 studs, lath and plaster	7
4-in. brick, 8-in. tile and plaster	10½
13-in. brick, plastered	12
9-in. brick, 3¾-in. tile, 5¼-in. air space, 3¾-in. tile and 1¼-in. plaster	16

radiation passing through it when the impingement is normal. For smaller angles of impingement, the glass retards percentages of the total radiant energy approximately in proportion to the sine of the angle.

The amount of solar radiation delivered to an unshaded glass surface may be obtained from Tables 2, 3, 4 or 5. These values must be used only for the net glass area on which the sun shines and not the entire glass area. Tests at the A.S.H.V.E. Research Laboratory[7] have determined the percentage of heat from solar radiation actually delivered to a room with various types of outdoor and indoor shading. The data in Table 7 are taken from these tests.

The percentage values in this table were obtained by dividing the total amount of heat actually entering through the shaded window by the total amount of heat calculated to enter through a bare window (solar radiation plus glass transmission, based on observed outside glass temperature). For bare windows on which the sun shines, the transmission of heat from outside air to glass may be small or negative as the glass temperature is raised by the solar radiation absorbed.

In calculating the total heat gain through windows on the sunny side of buildings, it is sufficiently accurate to proceed as outlined herewith:

TABLE 7. SOLAR RADIATION TRANSMITTED THROUGH SHADED WINDOWS

Type of Appurtenance	Finish Facing Sun	Per Cent Delivered to Room
Canvas awning	Plain	28
Canvas awning	Aluminum	22
Inside shade, fully drawn	Aluminum	45
Inside shade, one-half drawn	Buff	68
Inside Venetian blind, fully covering window, slats at 45 deg	Aluminum	58
Outside Venetian blind, fully covering window, slats at 45 deg	Aluminum	22

[7]Loc. Cit. Note 6.

Consider the total heat gain as that resulting from solar radiation and neglect the heat transmission through the glass caused by the difference between the temperatures of the inside and outside air. This method should be used except at times when the calculated heat gain per square foot due to normal transmission exceeds the solar intensity. At such times, solar radiation may be neglected and the total heat gain considered as resulting from normal transmission.

The solar heat transmission through windows or skylights may be expressed by the formula:

$$H_G = A_G f I \qquad (2)$$

where

H_G = solar radiation transmitted through a window, Btu per hour.

A_G = net area of glass exposed to sun's rays, square feet.

f = percentage of solar radiation (expressed as a decimal) transmitted to the inside (Table 7). For bare windows, $f = 1$.

I = intensity of solar radiation striking surface, Btu per hour per square foot (Tables 2, 3, 4 and 5).

In Equation 2, $f = 1$ for bare windows because the tests from which Table 7 was obtained showed that approximately all of the solar radiation impinging on a bare window became a part of the heat load in the room. This was because almost all of the heat absorbed by the glass flowed into the room by conduction. Other tests[8] have indicated that in the case of a building having floors of high heat capacity such as concrete floors on which the solar radiation falls, approximately one-half of the heat entering a bare window is absorbed by the floor and does not immediately become a part of the cooling load, but is delivered back to the air in the building at a slow rate over a period of 24 hours or longer.

The maximum solar intensity on any surface is of limited duration as shown in Fig. 1. In the case of windows the total energy impinging on the glass before and after the time of maximum intensity is further reduced by increased shading of the glass from the frame, or wall. The cooling load due to solar radiation therefore does not have to be calculated as a steady load. Another point which should be noted is that the maximum solar radiation load on the east wall occurs early in the morning when the outside temperature is low.

Tests have been made which indicated that solar radiation through window glass is the most important factor to contend with in the cooling of an office building. At times it was shown to account for as much as 75 per cent of the total cooling necessary. Because of the importance of the sun load, cooling systems should be zoned so that the side of the building on which the sun is shining can be controlled separately from the other sides of the building. If buildings are provided with awnings so that the window glass is shielded from sunshine, the amount of cooling required will be reduced and there will also be less difference in the cooling requirements of different sides of the building. The total cooling load for a building exposed to the sun on more than one side is of course less than the sum of the maximum cooling loads in the individual rooms since the maximum solar radiation load on the different sides occurs at different

[8]A.S.H.V.E. Research Report No. 1002—Cooling Requirements of Single Rooms in a Modern Office Building, by F. C. Houghten, Carl Gutberlet, and Albert J. Wahl (A.S.H.V.E. Transactions, Vol. 41, 1935, p. 53).

times. In determining the total cooling load for a building if the time when the maximum load occurs is not obvious, the load should be calculated for various times of day to determine the times at which the sum of the loads on the different sides of the building is a maximum.

The direct solar and scattered sky radiation penetration through glass block panels is given in Table 8 for various times of the day for south, east and west exposures for different latitudes on August 1. This table also gives the total heat gain into an air conditioned space when 78 F is maintained indoors, resulting from the effect of both radiation and air to air transmission. These values result from A.S.H.V.E. Laboratory data[9] and apply for expected design radiation intensity, and for a design day

TABLE 8. HEAT GAIN THROUGH GLASS BLOCKS[a]

		Solar Radiation Heat Gain (Direct plus Sky) Btu per Sq Ft per Hour						Total Heat Gain[b] (Solar Radiation plus Normal Transmission) Btu per Sq Ft per Hour					
Side		East[c]	West[c]	South				East[c]	West[c]	South			
N. Latitude Degrees		40	40	30	35	40	45	40	40	30	35	40	45
Sun Time	Outside Temp F												
7:00	74	65.0		1.0	2.8	3.0	5.0	61.0		−4.5	−2.0	−0.5	1.0
8:00	76	63.0	0.0	3.0	4.4	6.5	11.0	77.5		0.0	2.0	4.0	5.0
9:00	79	40.0	5.0	5.5	7.1	10.2	13.4	73.5	5.0	5.0	7.0	10.0	12.0
10:00	83	24.0	6.0	8.5	11.3	14.7	17.1	57.5	6.5	11.0	15.0	18.0	20.8
11:00	87	15.5	7.0	12.0	15.2	18.7	21.8	45.0	7.5	16.5	22.0	25.5	32.0
12:00	90	10.0	10.0	14.0	17.4	21.0	24.8	36.5	10.5	21.5	28.0	33.8	40.8
1:00	93	7.0	15.5	12.0	15.2	18.7	21.8	30.0	22.0	25.0	31.8	38.5	46.0
2:00	94	6.0	24.0	8.5	11.3	14.7	17.1	24.0	35.0	26.0	32.0	39.0	47.0
3:00	95	5.0	40.0	5.5	7.1	10.2	13.4	19.5	55.0	24.0	29.8	36.5	45.0
4:00	95	4.5	65.0	3.0	4.4	6.5	11.0	15.5	77.0	20.0	25.5	31.5	40.5
5:00	93	4.0	63.0	1.0	2.8	3.0	5.0	12.5	85.5	15.0	20.0	25.2	33.5
6:00	91	2.5	23.5	0.0	0.7	0.7	3.0	10.5	55.0	9.5	13.5	18.0	25.5
7:00	89	1.5	0.0			0.0	0.7	8.0	18.5	3.5	7.0	11.0	18.0

[a]For August 1.

[b]Inside temperature, 78 F.

[c]For east and west walls these values can be applied to all latitudes between 30 and 45 deg N without excessive errors.

having a maximum temperature of 95 F. The resulting heat gains are averages for four typical glass block designs, two having smooth exterior faces, and the other two having exterior ribbed faces, as made by two different manufacturers.

Heat Emission of Occupants

The heat and moisture given off by human beings under different states of activity are shown in various tables and figures of Chapter 2 which covers the physical and physiological principles of air conditioning. It will be observed from these data that the rate of sensible and latent heat emission by human beings varies greatly depending upon state of activity. In many applications this component becomes a large per-

[9]Loc. Cit. Note 1.

centage of total load. Metabolic rates are markedly variable for some extreme environmental conditions and this is another important factor which must be considered in cooling load computations.

Heat Introduced by Outside Air

An allowance must be made for the heat and moisture in the outside air introduced for ventilation purposes or entering the building through cracks, doors, and other places where infiltration might occur.

The volume of air entering due to infiltration may be estimated from data given in Chapter 5. Information on the amount of outside air required for ventilation will be found in Chapter 2.

In the event the volume of air entering an enclosure due to infiltration exceeds that required for ventilation, the former should be used as a basis for determining the portion of the load contributed by outside air. Where volume of air required for ventilation exceeds that due to infiltration it is assumed that a slight positive pressure will exist within the enclosure with a resulting exfiltration instead of infiltration. In this case the air required for ventilation is used in determining outside air load.

The heat gain resulting from outside air introduced may be determined by the following formula:

$$H = \frac{Q}{v}(h_o - h_i) \tag{3}$$

where

H = heat to be removed from outside air entering the building, Btu per hour.
Q = volume of outside air entering building, cubic feet per hour.
v = cubic feet of outside air per pound of dry air.
h_o = enthalpy of outside air, Btu per pound of dry air.
h_i = enthalpy of inside air, Btu per pound of dry air.

The latent heat gain resulting from outside air introduced may be determined by the following formula:

$$H_l = \frac{Q}{v} h_{fg} (W_o - W_i) \tag{4}$$

where

H_l = latent heat to be removed, Btu per hour.
h_{fg} = latent heat of evaporation at temperature at which water is condensed, Btu per pound.
W_o = humidity ratio of outside air, pounds water per pound dry air.
W_i = humidity ratio of inside air, pounds water per pound dry air.

Heat Emission of Appliances

Heat generating appliances which give off either sensible heat or both sensible and latent heat in an air conditioned enclosure may be divided into three general classes of equipment or devices:

1. Electrical appliances.
2. Gas appliances.
3. Steam heating appliances.

In the first group may be found such devices as lights, motors, toasters, waffle irons, etc. The capacities of most electrical devices may be determined from the watt capacity indicated on their name plates. The Btu equivalent of heat generated per hour is determined by multiplying the watt capacity by 3.4 (one watthour is equivalent to 3.413 Btu).

The capacities of electric motors are usually expressed in terms of horsepower instead of watts. If the motor efficiency is known, the watts input may be calculated from the formula:

$$P = \frac{746\ (hp)}{n} \tag{5}$$

where

P = motor input, watts.
hp = motor load, horsepower.
n = motor efficiency (expressed as a decimal).

When the motor efficiency is not known the heat equivalent of electrical input can be approximately determined by applying data given in Table 9.

TABLE 9. HEAT GENERATED BY MOTORS

Nameplate Rating Horsepower	Heat Gain in Btu per Hour per Horsepower	
	Connected Load in Same Room	Connected Load Outside of Room
⅛ to ½	4250	1700
½ to 3	3700	1150
3 to 20	2950	400

In the second group belong such appliances as coffee urns, gas ranges, steam tables, broilers, hot plates, etc. For heat generating capacities of such appliances refer to Table 10.

Considerable judgment must be exercised in the use of data given in Table 10. Consideration must be given to time of day when appliances are used and the heat they contribute at time of peak load. Only those appliances in use at the time of the peak load need be considered. Consideration must also be given to the way appliances are installed, whether products of combustion are vented to a flue, whether they escape into the space to be conditioned, or whether appliances are hooded allowing part of the heat to escape through a stack. There are no generally accepted data available on the effects of venting and shielding heating appliances but it is believed that, when they are properly hooded with a positive fan exhaust system through the hood, 50 per cent of the heat will be carried away and 50 per cent dissipated in the space to be conditioned. Where latent as well as sensible heat is given off, it is usually safe to assume that all latent heat will be removed by a properly designed and operated vent or hood.

ILLUSTRATION

From the foregoing discussion it is obvious that the determination of the maximum cooling load is rather complicated by reason of the variable nature of contributing load components.

TABLE 10. HEAT GAIN FROM VARIOUS SOURCES

Source	Btu Per Hour		
	Sensible	Latent	Total
Electric Heating Equipment			
Electrical Equipment—Dry Heat—No Evaporated Water	100%	0%	100%
Electric Oven—Baking	80%	20%	100%
Electric Equipment—Heating Water—Stewing, Boiling, etc.	50%	50%	100%
Electric Lights and Appliances per Watt (Dry Heat)	3.4	0	3.4
Electric Lights and Appliances per Kilowatt (Dry Heat)	3413	0	3413
Coffee Urn—Per Gallon Capacity	1025	1025	2050
Electric Range, Household—Small Burner (60% of connected load)	*	*	2050
Electric Range, Household—Large Burner (60% of connected load)	*	*	4505
Electric Range, Household—Oven	8000	2000	10000
Steam Table—Per Square Foot of Top Surface (35% of connected load)	105	300	405
Plate Warmer—Per Cubic Foot Inside Volume (50% of connected load)	615	0	615
Bakers Oven—Per Cubic Foot Inside Volume (60% of connected load)	1300	500	1800
Frying Griddles—Per Square Foot of Top Surface (60% of connected load)	2160	240	2400
Hot Plates—Per Square Foot of Top Surface (60% of connected load)	*	*	6000
Waffle Baker—Per Section (40% of connected load)	*	*	1365
Toaster—Per Slice (50% of connected load)	945	105	1050
Glass Coffee Maker—Per Section	*	*	1365
Sandwich Grille—Per Square Foot of Area (60% of connected load)	*	*	2750
Fry Kettle—Per Pound Fat Capacity (60% of connected load)	*	*	700
Hair Dryer in Beauty Parlor—600 w.	2050	0	2050
Permanent Wave Machine in Beauty Parlor—24-25 w Units	2050	0	2050
Gas Burning Equipment			
Gas Equipment—Dry Heat—No Water Evaporated	90%	10%	100%
Gas Heated Oven—Baking	67%	33%	100%
Gas Equipment—Heating Water—Stewing, Boiling, etc.	50%	50%	100%
Stove, Domestic Type—No Water Evaporated—Per Medium Size Burner	9000	1000	10000
Gas Heated Oven—Domestic Type	12000	6000	18000
Stove, Domestic Type—Heating Water—Per Medium Size Burner	5000	5000	10000
Residence Gas Range—Giant Burner (About 5½ in. Diameter)	*	*	12000
Residence Gas Range—Medium Burner (About 4 in. Diameter)	*	*	9000
Residence Gas Range—Double Oven (Total Size 18 in. x 18 in. x 22 in. High)	*	*	18000
Residence Gas Range—Pilot	*	*	250
Restaurant Range—4 Burners and Oven	*	*	100000
Cast-Iron Burner—Low Flame—Per Hole	*	*	100
Cast-Iron Burner—High Flame—Per Hole	*	*	250
Simmering Burner	*	*	1800
Coffee Urn—Large, 18 in. Diameter—Single Drum	5000	5000	10000
Coffee Urn—Small, 12 in. Diameter—Single Drum	3000	3000	6000
Coffee Urn—Per Gallon of Rated Capacity	500	500	1000
Egg Boiler—Per Egg Compartment	2500	2500	5000
Steam Table or Serving Table—Per Square Foot of Top Surface	400	900	1300
Dish Warmer—Per Square Foot of Shelf	540	60	600
Cigar Lighter—Continuous Flame Type	2250	250	2500
Curling Iron Heater	2250	250	2500
Bunsen Type Burner—Large—Natural Gas	*	*	5000
Bunsen Type Burner—Large—Artificial Gas	*	*	3000
Bunsen Type Burner—Small—Natural Gas	*	*	3000
Bunsen Type Burner—Small—Artificial Gas	*	*	1800
Welsbach Burner—Natural Gas	*	*	3000
Welsbach Burner—Artificial Gas	*	*	1800
Fish-tail Burner—Natural Gas	*	*	5000
Fish-tail Burner—Artificial Gas	*	*	3000
Lighting Fixture Outlet—Large, 3 Mantle 480 C.P.	4500	500	5000
Lighting Fixture Outlet—Small, 1 Mantle 160 C.P.	2250	250	2500
One Cubic Foot of Natural Gas Generates	900	100	1000
One Cubic Foot of Artificial Gas Generates	500	50	550
One Cubic Foot of Producer Gas Generates	135	15	150
Steam Heated Equipment			
Steam Heated Surface Not Polished—Per Square Foot of Surface	330	0	330
Steam Heated Surface Polished—Per Square Foot of Surface	130	0	130
Insulated Surface, Per Square Foot	80	0	80
Bare Pipes, Not Polished Per Square Foot of Surface	400	0	400
Bare Pipes, Polished Per Square Foot of Surface	220	0	220
Insulated Pipes, Per Square Foot	110	0	110
Coffee Urn—Large, 18 in. Diameter—Single Drum	2000	2000	4000
Coffee Urn—Small, 12 in. Diameter—Single Drum	1200	1200	2400
Egg Boiler—Per Egg Compartment	2500	2500	5000
Steam Table—Per Square Foot of Top Surface	300	800	1100
Miscellaneous			
Heat Liberated By Food per person, as in a Restaurant	30	30	60
Heat Liberated from Hot Water used direct and on towels per hour—Barber Shops	100	200	300

*Per cent sensible and latent heat depends upon use of equipment; dry heat, baking or boiling.

Application of the foregoing data in determining cooling load requirements is illustrated in Example 1.

Example 1. Determine cooling load requirements for a clothing store illustrated in Fig. 4 and located in Pittsburgh, Pa., Latitude 40 deg. This is a one-story building located on a corner and it faces south and west. Assume building on east and north sides conditioned.

Wall construction, 8 in. hollow tile, 4 in. brick veneer, plaster on walls, $U = 0.33$ (Table 4, Chapter 4, No. 38 B).

Roof construction, 2 in. concrete, ½ in. rigid insulation, metal lath and plaster ceiling, $U = 0.26$ (Table 11, Chapter 4, No. 2 J).

Floor, maple flooring on yellow pine, no ceiling below, $U = 0.34$ (Table 8, Chapter 4, No. 1 D).

Partition, wood lath and plaster on both sides of studding, $U = 0.34$ (Table 6, Chapter 4, No. 77 B).

Show windows, provided with awnings and thin panel partition at rear.

Front doors, 2 ft 6 in. x 7 ft (glass paneled).

Side door, 3 ft x 7 ft (glass paneled), $U = 1.13$ (Table 13 A, Chapter 4).

Occupancy, 10 clerks, 40 patrons.

Lights, 4200 w.

Outside design conditions, dry-bulb 95 F; wet-bulb 75 F.

Inside design conditions, dry-bulb 80 F; wet-bulb 67 F.

Basement temperature, 85 F.

Store room temperature, 88 F.

Solution. It is obvious from the shape and exposure of this store and the large glass area on the west side that the maximum cooling load will occur during the afternoon when the sun is shining on the west wall. From Fig. 1, the peak load may be expected at 4:00 p.m.

The combined normal transmission and solar radiation transmission through the roof at 4:00 p. m. is obtained from Fig. 2. While none of the roofs in Fig. 2 is exactly like this one, roof *C* is similar. A heat flow of 11 Btu per square foot per hour was assumed,

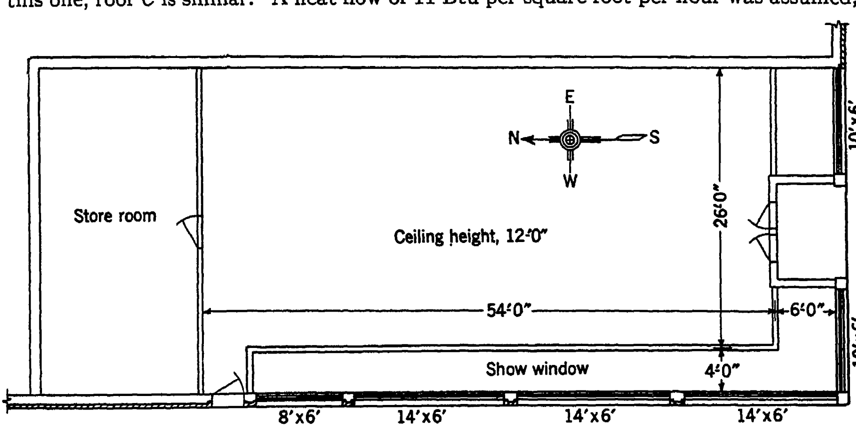


Fig. 4. Plan Diagram of Clothing Store

slightly less than for roof *C*. The combined normal transmission and solar radiation transmission through the south and west walls at 4:00 p.m. is obtained from curves *TS* and *TW* in Fig. 3.

The normal heat transmission through the south glass, floor and partition is determined by application of Formula 1. Solar radiation transmission through the south glass can be neglected. The solar intensity *I* for the south side at 4:00 p.m. is 29. Applying a shade factor of 0.28, the calculated solar radiation transmission is $29 \times 0.28 = 8$ Btu per square foot per hour which is less than the normal transmission, therefore the total heat gain can be taken as that due to normal transmission.

Solar radiation intensity on the west glass at 4:00 p.m. from Table 4 is 211 Btu per square foot per hour. As explained in the text, normal transmission can be neglected because it is small in comparison with solar radiation transmission.

To determine the heat gain from the outside air it is necessary first to determine the volume of the outside air to be introduced. Since the show windows are sealed so as not to permit infiltration and since there are only three doors in this store through which infiltration can take place, it is obvious that infiltration of air will be a negligible quantity. The volume of the store is 21,600 cu ft. Good practice indicates that in a store of this character there should be a minimum of from 1 to 1½ outside air changes per hour. On a basis of 1½ air changes the volume of outside air to be introduced would be 32,400 cfh. The minimum ventilation requirements as given in the CODE OF MINIMUM REQUIREMENTS FOR COMFORT AIR CONDITIONING[10] are 10 cfm per person. On this basis the ventilation requirements would be 30,000 cfh. Since this will produce approximately 1½ outside air changes per hour, 30,000 cfh will be considered in this application.

To determine load imposed by occupants it will be found from Table 3, Chapter 2 that the average person standing at rest will dissipate 431 Btu per hour and that the moisture dissipated is 0.198 lb per hour.

To determine the latent heat load, the sum of the moisture evaporated from occupants and that to be removed from outside air is multiplied by the latent heat of evaporation at the temperature at which the moisture is condensed in the conditioner. Since outside air is positively introduced, a mixture of outside and recirculated air passes through the conditioner. To remove the moisture, the air must be cooled to a temperature below the dew-point of the mixture. To obtain an approximate value of the latent heat of evaporation, assume that the air is cooled to 55 F. At this temperature, $h_{fg} = 1062.7$ Btu per hour (steam table).

COMBINED NORMAL AND SOLAR RADIATION TRANSMISSION:

SURFACE	DIMENSIONS	AREA SQ FT	BTU PER HOUR PER SQ FT	BTU PER HOUR
S Wall	(30 ft x 12 ft) − 155	205	3	615
W Wall	(60 ft x 12 ft) − 321	399	2.5	998
Roof	60 ft x 30 ft	1800	11	19,800
Total				21,413

NORMAL TRANSMISSION:

SURFACE	DIMENSIONS	AREA SQ FT	U	TEMP. DIFF. DEG F	BTU PER HOUR
S Glass	2 (2 ft 6 in. x 7 ft) + 2 (10 ft x 6 ft)	155	1.13	15	2,627
Floor	26 ft x 54 ft	1404	0.34	5	2,387
N Partition	30 ft x 12 ft	360	0.34	8	979
Total					5,993

[10]Code of Minimum Requirements for Comfort Air Conditioning (A.S H.V.E. TRANSACTIONS, Vol. 44, 1938 p. 27) Reprints of this code are available at $.10 a copy

SOLAR RADIATION THROUGH GLASS:

W Glass. A_G = 3 (14 ft × 6 ft) + (8 ft × 6 ft) + (3 ft × 7 ft) = 321 sq ft

H_G = 321 × 0.28 × 211 = 18,965 Btu per hour (Equation 2).

OUTSIDE AIR:

$H = \frac{Q}{v}(h_o - h_i)$ (Equation 3).

$v = v_a + \mu v_{as}$ (Equation 17, Chapter 1).

v_a = specific volume of dry air at 95 F = 13.97 cu ft per pound (Table 6, Chapter 1).

v_{as} = difference between volume of saturated mixture and specific volume of dry air at 95 F = 0.82 cu ft per pound (Table 6, Chapter 1).

μ = per cent saturation at 95 F dry-bulb and 75 F wet-bulb = 38.4 per cent (by calculation, Chapter 1).

v = 13.97 + (0.384 × 0.82) = 14.28 cu ft per pound dry air.

$h_o = h_a + \mu h_{as}$ (Equation 19, Chapter 1).

h_a = specific enthalpy of dry air at 95 F = 22.80 Btu per pound (Table 6, Chapter 1).

h_{as} = *difference between enthalpy of saturated mixture and specific enthalpy of* dry air at 95 F = 40.25 Btu per pound (Table 6, Chapter 1).

h_o = 22.80 + (0.384 × 40.25) = 38.26 Btu per pound dry air.

μ at 80 F dry-bulb and 67 F wet-bulb = 50.2 per cent (by calculation, Chapter 1).

$h_i = h_a + \mu h_{as}$ = 19.19 + (0.502 × 24.32) = 31.40 Btu per pound dry air (Table 6, Chapter 1).

$H = \frac{30,000}{14.28}(38.26 - 31.40)$ = 14,410 Btu per hour.

W_o = humidity ratio of outside air at 95 F and 75 F = 0.384 × 0.03652 = 0.01402 lb water per pound dry air. (Equation 14, Chapter 1).

W_i = humidity ratio of inside air at 80 F and 67 F = 0.502 × 0.02221 = 0.01115 lb water per pound dry air. (Equation 14, Chapter 1,)

Weight of water to be removed $= \frac{Q}{v}(W_o - W_i) = \frac{30,000}{14.28}(0.01402 - 0.01115) =$ 6.03 lb per hour.

OCCUPANTS:

50 × 431 = 21,550 Btu per hour.

50 × 0.198 = 9.95 lb water per hour evaporated.

LIGHTS:

4200 × 3.413 = 14,335 Btu per hour.

SUMMARY:

Component of Load	Btu per Hour
Combined Normal and Solar Radiation Transmission	21,413
Normal Transmission	5,993
Solar Radiation Through Glass	18,965
Outside Air	14,410
Occupants	21,550
Lights	14,335
Total	96,664

LATENT HEAT:

Outside air	6.03 lb water per hour.
Occupants	9.95 lb water per hour.
	15.98 lb water per hour.

$15.98 \times h_{fg} = 15.98 \times 1062.7 = 16{,}980$ Btu per hour.

Chapter 8

COMBUSTION AND FUELS

Principles of Combustion, Classification of Coals, Firing Methods for Coals, Firing Methods for Coke, Dustless Treatment of Coal, Classification of Oils, Combustion of Oil, Classification of Gas, Combustion of Gas

THE data given in the first part of this chapter are of general application to the various fuels used in domestic heating which are coal, coke, oil and gas. The choice of fuel is a question of dependability, cleanliness, fuel availability, economy, operating requirements and control.

FUNDAMENTAL PRINCIPLES OF COMBUSTION

Combustion may be defined as the chemical combination of a substance with oxygen with a resultant evolution of heat. The rate of combustion depends partly upon the specific rate of reaction of the combustible substance with oxygen and partly upon the rate at which oxygen is supplied and the surrounding conditions as they define the temperature.

Complete combustion is obtained when all of the combustible elements in the fuel are oxidized with all of the oxygen with which they can combine. All of the oxygen supplied may not be utilized.

Perfect combustion is defined as the result of supplying the required amount of oxygen for combination with all of the combustible elements of the fuel and utilizing all of the oxygen so supplied.

The oxygen required for the process of combustion is obtained from air which is a mechanical mixture of oxygen, nitrogen and small amounts of carbon dioxide, water vapor and inert gases. These inert gases are generally included with the nitrogen, and for engineering purposes the values given herewith may be used.

	By Volume Per Cent	By Weight Per Cent
Oxygen, O_2	20.9	23.15
Nitrogen, N_2	79.1	76.85

The combination of oxygen with the combustible elements and compounds of a fuel is in accordance with fixed laws. In the case of perfect combustion the reactions and resultant combinations are shown in Table 1.

The most important condition governing the process of combustion is temperature. It is necessary to bring a combustible substance to its

TABLE 1. GENERAL DATA OF COMBUSTIBLE ELEMENTS AND COMPOUNDS

Substance	Molecular Symbol	Chemical Reaction of Combustion	Ignition Temperature[a] Deg F	Calorific Value: Btu per Lb, Higher	Calorific Value: Btu per Lb, Lower	Calorific Value: Btu per Cubic Foot, Higher	Theoretical Oxygen and Air Requirements: Lb per Lb, O_2	Lb per Lb, Air	Cubic Ft per Cubic Ft, O_2	Cubic Ft per Cubic Ft, Air
Carbon (to *CO*)	—	$2C + O_2 = 2CO$	—	4380	—	—	1.333	5.76	—	—
Carbon (to CO_2)	—	$2C + 2O_2 = 2CO_2$	—	14540	—	—	2.667	11.52	—	—
Sulphur	S_2	—	—	—	—	—	1.000	4.32	—	—
Sulphur (to SO_2)	—	$S + O_2 = SO_2$	—	4050	—	—	—	—	—	—
Sulphur (to SO_3)	—	$2S + 3O_2 = 2SO_3$	—	5940	—	—	—	—	—	—
Carbon monoxide	CO	$2CO + O_2 = 2CO_2$	1166-1319	4380	—	342	0.572	2.46	0.5	2.391
Methane	CH_4	$CH_4 + 2O_2 = CO_2 + 2H_2O$	1202-1346	23850	21670	1073	4.000	17.28	2.0	9.564
Acetylene	C_2H_2	$2C_2H_2 + 5O_2 = 4CO_2 + 2H_2O$	763-824	21460	21020	1590	3.077	13.29	2.5	11.955
Ethylene	C_2H_4	$C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$	986-1123	21450	20420	1675	3.429	14.81	3.0	14.346
Ethane	C_2H_6	$2C_2H_6 + 7O_2 = 4CO_2 + 6H_2O$	986-1123	22230	20500	1883	3.733	16.13	3.5	16.737
Hydrogen	H_2	$2H_2 + O_2 = 2H_2O$	1063-1166	62000	52920	348	8.000	34.56	0.5	2.391
Hydrogen sulphide	H_2S	$2H_2S + 3O_2 = 2H_2O + 2SO_2$	599-608	—	—	—	1.412	6.10	1.5	7.173

[a]From International Critical Tables, 1927.

ignition temperature before it will unite in chemical combination with oxygen to produce combustion. The ignition temperatures for several of the combustible constituents of fuels are presented in Table 1.

HEAT OF COMBUSTION

As previously stated, the process of combustion results in the evolution of heat. The heat generated by the complete combustion of a unit of fuel is constant for a given combination of combustible elements and compounds, and is known as the *heat of combustion*, *calorific value*, or *heating value* of the fuel.

The heat of combustion of the several fuel elements and compounds in their *pure* state is given in Table 1.

The reaction of the carbon in the fuel with oxygen may result in the formation of carbon monoxide or carbon dioxide. In burning to carbon monoxide, the carbon is not completely oxidized and, as shown by the data, the heat produced is considerably less than if it were completely oxidized. This fact is of greatest importance in considering the efficiency of combustion.

The calorific value of a fuel is determined by direct measurement of the heat evolved during combustion in a calorimeter. Although the ash and moisture content of coal from a given mine or locality may vary widely, the heating value of the coal, on a *moisture and ash free* basis, remains relatively constant. It is therefore possible to approximate the heating value of a shipment of coal *as received* if its moisture and ash content are determined, and if the heating value of similar coal on a moisture and ash free basis is known. This may be calculated by Equation 1.

$$\text{Heating value, } \textit{as received} = \frac{\text{Heating value, } \textit{moisture and ash free} \times [100 - (\text{Moisture} + \text{Ash})]}{100} \tag{1}$$

where, moisture and ash are expressed in per cent.

The heating values for Illinois coals are published[1] and it is to be expected that values for other coals will be available in the future.

As practically all fuels contain hydrogen they produce a certain amount of water vapor as one of the products of combustion. The amount of water vapor produced increases as the hydrogen content of the fuel increases. When the heating value of a fuel is determined in a calorimeter the water vapor is condensed and the latent heat of vaporization that is given up during the condensation is reported as a portion of the heat value of the fuel. The heat value so determined is termed the *gross* or *higher* heat value and this is what is ordinarily meant when the heating value of a fuel is specified. In burning the fuel, however, the products of combustion are not cooled to the dew-point and the higher heating value cannot be obtained.

FLAME

The appearance of the flame or products of combustion may serve as an approximate measure of the temperatures developed in the combustion

[1]*State Geological Survey Bulletin*, No. 62, Classification and Selection of Illinois Coals.

TABLE 2. FLAME TEMPERATURE DATA

APPEARANCE OF FLAME	TEMPERATURE DEG F
Red, visible in daylight	975
Light red	1832
Orange-red	2012
Orange-yellow	2192
Yellow-white	2372
Bright white	2550

process. The luminosity of a flame is caused by the heating to incandescence of unconsumed particles of combustible matter in the gases, and the higher the temperature of these particles the whiter the flame. Table 2 gives some approximate flame temperature data.

AIR AND COMBUSTION

The weight of air required for the perfect combustion of a pound of fuel may be determined by use of the ultimate analysis of the fuel as applied to Equations 2 to 4. The various elements are expressed in percentages by weight.

Solid and Liquid Fuels:

$$\text{Pounds air required per pound fuel} = 34.56 \left[\frac{C}{3} + \left(H - \frac{O}{8}\right) + \frac{S}{8}\right] \quad (2)$$

Gaseous Fuels:

$$\text{Pounds air required per pound fuel} = 2.46\ CO + 34.56\ H_2 + 17.28\ CH_4 + 13.29\ C_2H_2 + 14.81\ C_2H_4 + 16.13\ C_2H_6 + 6.10\ H_2S - 4.32\ O_2 \quad (3)$$

When the analysis is given on a volumetric basis the equation is expressed as follows:

$$\text{Cubic feet air required per cubic foot gas} = 2.39\ (CO + H_2) + 9.56\ CH_4 + 11.98\ C_2H_2 + 14.35\ C_2H_4 + 16.74\ C_2H_6 - 4.78\ O_2 \quad (4)$$

Equations 5 and 6 may be used as approximate methods of determining the theoretical air requirement for any fuel.

$$\text{Pounds air required per pound fuel} = 0.755 \times \frac{\text{Heating value (Btu per pound)}}{1000} \quad (5)$$

$$\text{Cubic feet air required per unit fuel} = \frac{\text{Heating value (Btu per unit)}}{100} \quad (6)$$

Approximate values for the theoretical air required for different fuels are given in Table 3.

It is customary to make use of the analysis of the products of combustion to determine the amount of flue gas produced and the actual amount of air supplied for combustion. The analysis of flue gases has been well described in various publications of the *Bureau of Mines* and in the literature and the details of Orsat manipulation need not be considered in this discussion. (See Chapter 35.)

The weight of dry flue gas per pound of fuel burned is used in combustion loss calculations and may be determined by Equation 7.

$$\text{Pounds dry flue gas per pound fuel} = \frac{11\,CO_2 + 8\,O_2 + 7\,(CO + N_2)}{3\,(CO_2 + CO)} \times C \tag{7}$$

Values for CO_2, O_2, CO and N_2 are percentages by volume from the flue gas analysis and C is the weight of carbon burned per pound of fuel corrected for carbon in the ash.

EXCESS AIR

Because the real measure of the efficiency of combustion is the relation existing between the amount of air theoretically required for *perfect* combustion and the amount of air actually supplied, a method of determining the latter factor is of value. Equation 8 will give reasonably accurate results, for most solid and liquid fuels, for determining the amount of air supplied per pound of fuel.

$$\text{Pounds dry air supplied per pound of fuel} = \frac{3.036\,N_2}{(CO_2 + CO)} \times C \tag{8}$$

Values for CO_2, CO and N are percentages by volume from the flue gas analysis and C is the weight of carbon burned per pound of fuel corrected for carbon in the ash.

The difference between the air actually supplied for combustion and the theoretical air required is known as excess air.

$$\text{Per cent excess air} = \frac{\text{Air supplied} - \text{Theoretical air}}{\text{Theoretical air}} \tag{9}$$

Since the calculation is usually made from Orsat readings, Equation 10 will be found to be a convenient statement of this relationship.

TABLE 3. THEORETICAL AIR REQUIREMENTS

SOLID FUEL	POUNDS AIR PER POUND FUEL
Anthracite	9.6
Semi-bituminous coal	11.2
Bituminous coal	10.3
Lignite	6.2
Coke	11.2

FUEL OIL	POUNDS AIR PER GALLON FUEL
Commercial Standard No. 1	102.6
Commercial Standard No. 2	104.5
Commercial Standard No. 3	106.5
Commercial Standard No. 5	112.0
Commercial Standard No. 6	114.2

GASEOUS FUELS	CUBIC FEET AIR PER CUBIC FOOT GAS
Natural gas	10.0
Mixed, natural and water gas	4.4
Carbureted water gas	4.4
Water gas, coke	2.1
Coke oven gas	5.2

$$\text{Per cent excess air} = \frac{100\left(O_2 - \frac{CO}{2}\right)}{N_2 \times 0.264 - \left(O_2 - \frac{CO}{2}\right)} \tag{10}$$

In this formula the symbols represent volumetric percentages of the flue gas constituents as determined by analysis.

The amount of excess air in its relation to the percentage of CO_2 is shown by the curves in Fig. 1 for several fuels. These are approximate values. It should be noted that in hand-fired furnaces with long periods between firings the combustion goes through a cycle in each period and the quantity of excess air present varies.

Due to the different carbon-hydrogen ratios of the different fuels the maximum CO_2 attainable varies. Representative values for perfect combustion of several fuels are given in Table 4.

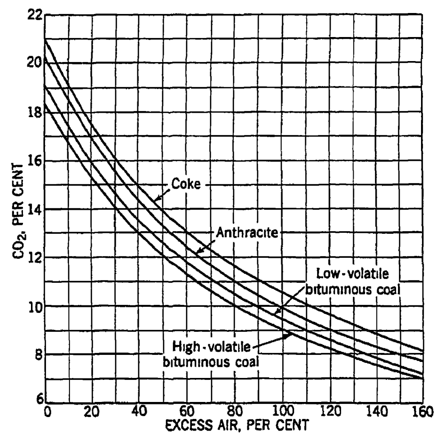


FIG. 1. RELATION BETWEEN CO_2 AND EXCESS AIR IN GASES OF COMBUSTION

In considering the factor of excess air it should be noted that a deficiency of air supply will result in combustible products passing to the stack unburned. An excess of air absorbs heat from the products of combustion and results in a greater loss of sensible heat to the stack. An excess of air is always required, however, to eliminate combustible losses occasioned by poor mixing of the fuel and air. It is considered good practice, under usual operating conditions, to supply from 25 to 50 per cent excess air, dependent upon the fuel utilized.

SECONDARY AIR

When bituminous coal is hand-fired in a furnace the volatile matter in the fuel distills off leaving coke on the grate. The product of combustion of the coke is CO_2 and under certain conditions some CO may arise from the bed. The combustion of the volatile matter and the CO may amount to the liberation of from 40 to 60 per cent of the heat in the fuel in the combustion space over the fuel bed.

TABLE 4. MAXIMUM CO_2 VALUES

FUEL	PER CENT CO_2
Coke	21.0
Anthracite	20.2
Bituminous coal	18.2
Oil	15.5
Natural gas	12.0
Coke oven gas	11.0

The air that passes through the fuel bed is called *primary air* and the air that is admitted over the fuel bed in order to burn the volatile matter and CO is called *secondary air*.

This process of combustion is illustrated in Fig. 2[2]. The free oxygen of the air passes through the grate and the ash above it and burns the carbon in the lower three or four inches of the fuel bed forming carbon dioxide. This layer noted as the oxidizing zone is indicated by the symbols CO_2 and O_2. Some of the carbon dioxide of the oxidizing zone is reduced to carbon monoxide in the upper layer of the fuel bed noted as the reducing zone and indicated by the symbols CO_2 and CO. The gases leaving the fuel

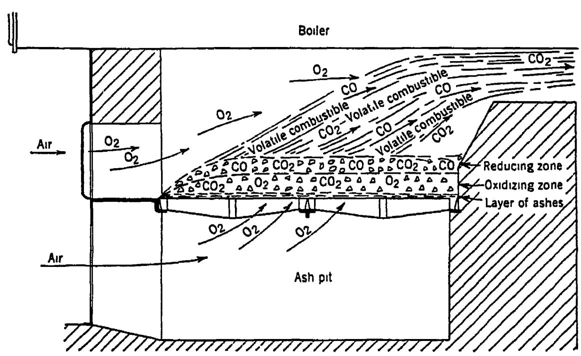


FIG 2. COMBUSTION OF FUEL IN A HAND-FIRED FURNACE

bed are mainly carbon monoxide, carbon dioxide, nitrogen and very little free oxygen. Free oxygen is admitted through the firing door to burn carbon monoxide and the volatile combustible distilled from the freshly fired fuel.

The division of the total into primary and secondary air necessary to produce the same rate of burning and the same excess air depends on a number of factors which include size and type of fuel, depth of fuel bed, and size of fire-pot. The ratio of the secondary to the primary air increases with decrease in the size of the fuel pieces, with increase in the depth of the fuel bed, and with increase in the area of the fire-pot; the ratio also increases with increase in rate of burning.

Size of the fuel is a very important factor in fixing the quantity of secondary air required for non-caking coals. With caking coals it is not so important because small pieces fuse together and form large lumps. Fortunately a smaller size fuel gives more resistance to air flow through

[2]From *Bureau of Mines Technical Paper* No. 80.

the fuel bed and thus automatically causes a larger draft above the fuel bed, which draws in more secondary air through the same slot openings. In spite of this, a small size fuel requires a larger opening of the door slots; for a certain size for each fuel no slot opening is required, and for larger sizes too much excess air gets through the fuel bed.

It is impossible to establish a single rule for the correct slot opening for all types and sizes of fuels and for all rates of burning. Furthermore, the effect of slot opening is dependent on whether the ashpit damper is open or closed. It is better to have too much than too little secondary air; the opening is too small if there is a puff of flame when the firing door is opened.

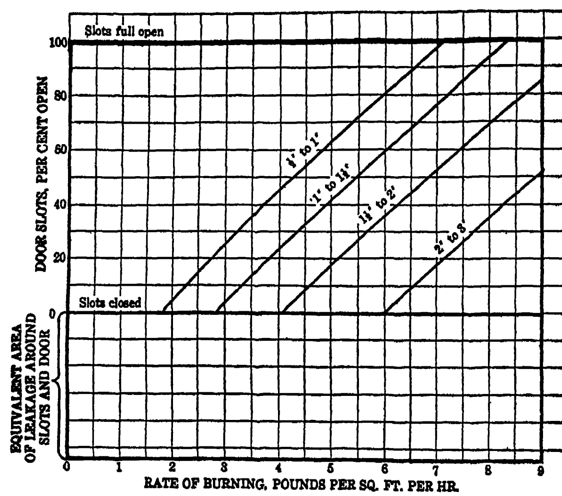


FIG. 3. RELATIVE AMOUNT OF FIRE DOOR SLOT OPENING REQUIRED IN A GIVEN FURNACE TO GIVE EQUALLY GOOD COMBUSTION FOR HIGH TEMPERATURE COKE OF VARIOUS SIZES WHEN BURNED AT VARIOUS RATES

The relationship of the slot opening, for a domestic furnace, to the size of coke and the rate of burning is shown in Fig. 3[3]. These openings are with the ashpit damper wide open, and would be less if the available draft permitted the damper to be partly closed. The same openings are satisfactory for anthracite.

Bituminous coals require a large amount of secondary air during the period subsequent to a firing in order to consume the gases and to reduce the smoke. The smoke produced is a good indicator, and that opening is best which reduces the smoke to a minimum. Too much secondary air will cool the gases below the ignition point, and prove harmful instead of beneficial. The following suggestions will be helpful:

1. In cold weather, with high combustion rates, the secondary air damper should be half open all the time.

2. In very mild weather, with a very low combustion rate, the secondary air damper should be closed all the time.

3. For temperatures between very mild and very cold, the secondary air damper should be in an intermediate position.

[3]From *Bureau of Mines Report of Investigations*, No. 2980.

4. For ordinary house operation, secondary air is needed after each firing for about one hour.

In the field of domestic heating the use of secondary air in the combustion of oil is generally restricted to the larger semi-commercial types of oil burners used in large heating boilers. This factor is discussed in Chapter 10, Automatic Fuel Burning Equipment.

The air that is supplied around the flame in a domestic heating gas burner is considered as secondary air. As it is drawn into the appliance by natural draft action, the need for proper draft control is evident.

Draft Requirements

The draft required to effect a given rate of burning the fuel as measured at the smokehood is dependent on the following factors:

1. Kind and size of fuel.
2. Combustion rate per square foot of grate area per hour.
3. Thickness of fuel bed.
4. Type and amount of ash and clinker accumulation.
5. Amount of excess air present in the gases.
6. Resistance offered by the boiler passes to the flow of the gases.
7. Accumulation of soot in the passes.

Insufficient draft will necessitate additional manipulation of the fuel bed and more frequent cleanings to keep its resistance down. Insufficient draft also restricts the control by adjustment of the dampers.

The quantity of excess air present has a marked affect on the draft required to produce a given rate of burning. If the excess is caused by holes in the fuel bed or an extremely thin fuel bed it is often possible to produce a higher rate of burning by increasing the thickness of the bed. The thickness of the fuel bed should not, however, be increased too much because the increased draft resistance will reduce the rate of primary air supply and the rate of burning.

DRAFT REGULATION

Because of the varying heating load demands present in most installations it is necessary to vary the rate of fuel burning. The maintenance of the proper air supply for the various rates of burning is accomplished by regulation of the drafts. Correct and incorrect methods of draft regulation are shown in Fig. 4. The air enters through the ashpit draft door, firing door and by leaks in the setting, whereas the gases leave only through the uptake. By throttling the gases with the damper in the uptake all the air entering by each of the three intakes is reduced in the same proportion. If the ashpit draft door is closed the air admitted through the ashpit is reduced and increased through the other two intake openings.

Methods of control of draft conditions when burning oil or gas are noted in Chapter 10, Automatic Fuel Burning Equipment.

CLASSIFICATION OF COALS

The complex composition of coal makes it difficult to classify it into clear-cut types. Its chemical composition is some indication but coals

having the same chemical analysis may have distinctly different burning characteristics. Users are mainly interested in the available heat per pound of coal, in the handling and storing properties, and in the burning characteristics. A description of the relationship between the qualities of coals and these characteristics requires considerable space; a treatment applicable to heating boilers is given in *Bureau of Mines Bulletin* No. 276.

Coal composition may be expressed by either an *ultimate* or *proximate* analysis. In the ultimate analysis the proportions of carbon, hydrogen, oxygen, nitrogen, sulphur, and ash are determined. This form of analysis is difficult to make and is used only for extremely close studies. The proximate analysis is more easily made and is satisfactory for most purposes. In this analysis, the proportions of moisture, volatile matter, fixed carbon, and ash are determined. Moisture is obtained by noting the loss of weight of a sample of coal when dried at about 220 F. To

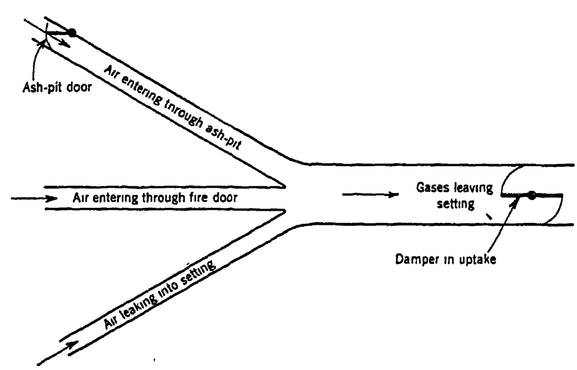


FIG. 4. CORRECT AND INCORRECT METHODS OF DRAFT REGULATION IN A HAND-FIRED FURNACE

determine volatile matter, the dried sample is heated to about 1750 F in a closed crucible, and the loss of weight is noted. The sample is then burned in an open crucible, and the accompanying loss of weight represents the fixed carbon. The unburned residue is ash. Although determined separately, the sulphur content is frequently reported with a proximate analysis.

Other important qualities of coals are the screen sizes, ash fusion temperature, friability, caking tendency, and the qualities of the volatile matter. In considering these factors the following points are of interest. The volatile products given off by coals when they are heated differ materially in the ratios by weight of the gases to the oils and tars. No heavy oils or tars are given off by anthracite, and very small quantities are given off by semi-anthracite. As the volatile matter in the coal increases to as much as 40 per cent of ash and moisture-free coal, increasing amounts of oils and tars are released. For coals of higher volatile content, the relative quantity of oils and tars decreases, so it is low in the sub-bituminous coals and in lignite. The percentage of ash and its fusion temperature do not indicate how the ash is distributed or how much of it is less fusible lumps of slate or shale.

A classification of coals is given in Table 5, and a brief description of the kinds of fuels is given in the following paragraphs, but it should be recognized that there are no distinct lines of demarcation between the kinds, and that they graduate into each other.

Anthracite is a clean, dense, hard coal which creates little dust in handling. It is comparatively hard to ignite but it burns freely when well started. It is non-caking, it burns uniformly and smokelessly with a short flame, and it requires no attention to the fuel bed between firings. It is capable of giving a high efficiency in the common types of hand-fired furnaces. A tabulation of the quality of the various anthracite sizes will be found in *Bureau of Mines Report of Investigations* No. 3283.

Semi-anthracite has a higher volatile content than anthracite. It is not so hard and ignites somewhat more easily; otherwise its properties are similar to those of anthracite.

Semi-bituminous coal is soft and friable, and fines and dust are created by handling it. It ignites somewhat slowly and burns with a medium length of flame. Its caking properties increase as the volatile matter increases, but the coke formed is relatively weak.

TABLE 5. CLASSIFICATION OF COALS BY RANK[a]

Legend: F.C. = Fixed Carbon. V.M. = Volatile Matter. Btu = British thermal units.

CLASS	GROUP	LIMITS OF FIXED CARBON OR BTU MINERAL-MATTER-FREE BASIS	REQUISITE PHYSICAL PROPERTIES
I. Anthracite	1. Meta-anthracite	Dry F.C., 98 per cent or more (Dry V.M., 2 per cent or less)	Non-agglomerating[b]
	2. Anthracite	Dry F.C., 92 per cent or more and less than 98 per cent (Dry V.M., 8 per cent or less and more than 2 per cent)	
	3. Semi-anthracite	Dry F.C., 86 per cent or more and less than 92 per cent (Dry V.M., 14 per cent or less and more than 8 per cent)	
II Bituminous[d]	1. Low volatile bituminous coal	Dry F.C., 78 per cent or more and less than 86 per cent (Dry V.M., 22 per cent or less and more than 14 per cent)	Either agglomerating[b] or non-weathering[f]
	2. Medium volatile bituminous coal	Dry F.C., 69 per cent or more and less than 78 per cent (Dry V.M., 31 per cent or less and more than 22 per cent)	
	3. High volatile *A* bituminous coal	Dry F.C., less than 69 per cent (Dry V.M., more than 31 per cent); and moist[c] Btu, 14,000[e] or more	
	4. High volatile *B* bituminous coal	Moist[c] Btu, 13,000 or more and less than 14,000[e]	
	5. High volatile *C* bituminous coal	Moist Btu, 11,000 or more and less than 13,000[e]	
III. Sub-bituminous	1. Sub-bituminous *A* coal	Moist Btu, 11,000 or more and less than 13,000[e]	Both weathering and non-agglomerating[b]
	2. Sub-bituminous *B* coal	Moist Btu, 9500 or more and less than 11,000[e]	
	3. Sub-bituminous *C* coal	Moist Btu, 8300 or more and less than 9500[e]	
IV. Lignitic	1. Lignite	Moist Btu less than 8300	Consolidated
	2. Brown coal	Moist Btu less than 8300	Unconsolidated

[a]This classification does not include a few coals which have unusual physical and chemical properties and which come within the limits of fixed carbon or Btu of the high-volatile bituminous and sub-bituminous ranks. All of these coals either contain less than 48 per cent dry, mineral-matter-free fixed carbon or have more than 15,500 moist, mineral-matter-free Btu.

[b]If agglomerating, classify in low-volatile group of the bituminous class.

[c]Moist Btu refers to coal containing its natural bed moisture but not including visible water on the surface of the coal.

[d]It is recognized that there may be non-caking varieties in each group of the bituminous class.

[e]Coals having 69 per cent or more fixed carbon on the dry, mineral-matter-free basis shall be classified according to fixed carbon, regardless of Btu.

[f]There are three varieties of coal in the High-volatile *C* bituminous coal group, namely, Variety 1, agglomerating and non-weathering; Variety 2, agglomerating and weathering; Variety 3, non-agglomerating and non-weathering.

Adapted from *A.S.T.M.* Standards, 1937, Supplement, p. 145, *American Society for Testing Materials* Philadelphia.

Having only half the volatile matter content of the more abundant bituminous coals it can be burned with less production of smoke, and it is sometimes called *smokeless coal.*

The term *bituminous coal* covers a large range of coals and includes many types having distinctly different composition, properties, and burning characteristics. The coals range from the high-grade bituminous coals of the East to the poorer coals of the West. Their caking properties range from coals which completely melt, to those from which the volatiles and tars are distilled without change of form, so that they are classed as non-caking or free-burning. Most bituminous coals are strong and non-friable enough to permit of the screened sizes being delivered free from fines. In general, they ignite easily and burn freely; the length of flame varies with different coals, but it is long. Much smoke and soot are possible especially at low rates of burning.

Sub-bituminous coals occur in the western states; they are high in moisture when mined and tend to break up as they dry or when exposed to the weather; they are liable to ignite spontaneously when piled or stored. They ignite easily and quickly and have a medium length flame, are non-caking and free-burning; the lumps tend to break into small pieces if poked; very little smoke and soot are formed.

Lignite is of woody structure, very high in moisture as mined, and of low heating value; it is clean to handle. It has a greater tendency than the sub-bituminous coals to disintegrate as it dries, and it also is more liable to spontaneous ignition. Freshly mined lignite, because of its high moisture, ignites slowly. It is non-caking. The char left after the moisture and volatile matter are driven off burns very easily, like charcoal. The lumps tend to break up in the fuel bed and pieces of char falling into the ashpit continue to burn. Very little smoke or soot is formed.

It is often desirable to learn about the properties of a coal, such as the various items noted in the discussion of proximate analyses. As a guide for the consumer as to the expected characteristics of coals several commercial publications are available and numerous reports of the *Bureau of Mines* discuss the coals produced in individual state areas.

CLASSIFICATION OF COKES

Coke is produced by the distillation of the volatile matter from coal. The type of coke depends on the coal or mixture of coals used, the temperatures and time of distillation and, to some extent, on the type of retort or oven; coke is also produced as a residue from the destructive distillation of oil.

High-temperature cokes. Coke as usually available is of the high-temperature type, and contains between 1 and 2 per cent volatile matter. High-temperature cokes are subdivided into *beehive coke* of which comparatively little is now sold for domestic use, *by-product coke,* which covers the greater part of the coke sold, and *gas-house coke.* The differences among these three cokes are relatively small; their denseness and hardness decrease and friability increases in the order named. In general, the lighter and more friable cokes ignite and burn the more easily.

Low-temperature cokes are produced at low coking temperatures, and only a portion of the volatile matter is distilled off. Cokes as made by various processes under development have contained from 10 to 15 per cent volatile matter. In general, these cokes ignite and burn more readily than high-temperature cokes. The properties of various low-temperature cokes may differ more than those of the various high-temperature cokes because of the differences in the quantities of volatile matter and because some may be light and others briquetted.

Petroleum cokes, which are obtained by coking the residue left from the distillation of petroleum, vary in the amount of volatile matter they contain, but all have the common property of a very low ash content, which necessitates the use of refractory pieces to protect the grates from being burned.

FIRING METHODS FOR ANTHRACITE[4]

An anthracite fire should never be poked or disturbed, as this serves to bring ash to the surface of the fuel bed where it may melt into clinker.

[4]See reports published by *Anthracite Industries Laboratory,* Primos, Delaware County, Pennsylvania.

Egg size is suitable for large firepots (grates 24 in. and over) if the fuel can be fired at least 16 in. deep. The air spaces between the pieces of coal are large, and for best results this coal should be fired deeply.

Stove size coal is the proper size of anthracite for many boilers and furnaces used for heating buildings. It burns well on grates at least 16 in. in diameter and 12 in. deep. The only instructions needed for burning this type of fuel are that the grate should be shaken daily, the fire should never be poked or disturbed, and the fuel should be fired deeply and uniformly.

Chestnut size coal is in demand for fire-pots up to 20 in. in diameter, with a depth of from 10 to 15 in.

Pea size coal is often an economical fuel to burn. It is relatively low in price. When fired carefully, pea coal can be burned on standard grates. Care should be taken to shake the grates only until the first bright coals begin to fall through the grates. The fuel bed, after a new fire has been built, should be increased in thickness by the addition of small charges until it is at least level with the sill of the fire-door. This keeps a bed of ignited coal in readiness against the time when a sudden demand for heat shall be made on the heater. A satisfactory method of firing pea coal consists of drawing the red coals toward the front end and piling fresh fuel toward the back of the fire-box.

Pea size coal requires a strong draft and therefore the best results generally will be obtained by keeping the choke damper open and regulating solely by means of the cold air check and the air inlet damper. As a precaution against clinker, it is well to adjust the air inlet damper so that it can never be completely closed under any operating conditions.

Buckwheat size coal for best results requires more attention than pea size coal, and in addition the smaller size of the fuel makes it more difficult to burn on ordinary grates. Greater care must be taken in shaking the grates than with the pea coal on account of the danger of the fuel falling through the grate. In house heating furnaces the coal should be fired lightly and more frequently than pea coal. When banking a buckwheat coal fire it is advisable after coaling to expose a small spot of hot fire by putting a straight poker down through the bed of fresh coal. This will serve to ignite the gas that will be distilled from the fresh coal and prevent delayed ignition within the fire-pot, which in some cases, depending upon the thickness of the bed of fresh coal, is severe enough to blow open the doors and dampers of the furnace. Where frequent attention can be given and care exercised in manipulation of the grates this fuel can be burned satisfactorily without the aid of any special equipment.

In general it will be found more satisfactory with buckwheat coal to maintain a uniform heat output and consequently to keep the system warm all the time, rather than to allow the system to cool off at times and then to attempt to burn the fuel at a high rate while warming up. A uniform low fire will minimize the clinker formation and keep the clinker in an easily broken up condition so that it readily can be shaken through the grate.

Forced draft and small mesh grates are frequently used for burning buckwheat anthracite. For greater convenience, domestic stokers are used.

Buckwheat anthracite No. 2, or rice size, is used principally in stokers of the domestic, commercial and industrial type. No. 3 buckwheat anthracite, or barley, has no application in domestic heating.

The *Anthracite Institute* Standards of sizing are shown in Table 6 taken from *Anthracite Industries Manual,* Report No. 2403.

FIRING METHODS FOR BITUMINOUS COAL

Bituminous coal should never be fired over the entire fuel bed at one time. A portion of the glowing fuel should always be left exposed to ignite the gases leaving the fresh charge.

Air should be admitted over the fire through a special secondary air device, or through a slide in the fire-door or by opening the fire-door slightly. If the quantity of air admitted is too great the gases will be cooled below the ignition temperature and will fail to burn. The fireman can judge the quantity of air to admit by noting when the air supplied is just sufficient to make the gases burn rapidly and smokelessly above the fuel bed.

TABLE 6. ANTHRACITE STANDARDS

CLASSIFICATION	COAL SIZE, INCHES	
Egg	Through 3-1/4	Over 2-7/16
Stove	Through 2-7/16	Over 1-5/8
Nut	Through 1-5/8	Over 13/16
Pea	Through 13/16	Over 9/16
Buckwheat	Through 9/16	Over 5/16

The red fuel in the fire-box, before firing, excepting only a shallow layer of coke on the grate, should be pushed to one side or forward or backward to form a hollow in which to throw the fresh fuel. Some manufacturers recommend that all red fuel be pushed to the rear of the fire-box and that the fresh fuel be fired directly on the grate and allowed to ignite from the top. The object of this is to reduce the early rapid distillation of gases and to reduce the quantity of secondary air required for smokeless combustion.

It is well to have the bright fuel in the fire-box so placed that the gases from the freshly fired fuel, mixed with the air over the fuel bed, pass over the bed of bright fuel on the way to the flues. The bed of bright fuel then supplies the heat to raise the mixture of air and gas to the ignition temperature, thereby causing the gaseous matter to burn and preventing the formation of smoke.

The importance of firing bituminous coal in small quantities at short intervals is discussed in the *U. S. Bureau of Mines Technical Paper,* No. 80. Better combustion is obtained by this method in that the fuel supply is maintained more nearly proportional to the air supply.

This is demonstrated in Fig. 5 where diagram A shows the air supply and the distillation of the volatile combustible when the firings are 5 min apart; and diagram B indicates the same relationships when the firings are 15 min apart. In both cases the amount of coal fired per hour and the

weight of volatile combustible distilled from the coal are the same. This weight of volatile combustible is represented by the shaded area under the saw-tooth curve. The horizontal dotted lines represent the constant air supply sufficient to burn the volatile matter represented by the shaded areas under each line. The shaded areas above each horizontal line represent for each air supply the loss from incomplete combustion of the volatile matter. The clear area under each horizontal line represents the loss from excessive air. As the air supply increases the loss from incomplete combustion decreases but the loss from excessive air becomes larger. The sum of the two losses is the least when the air supply is introduced as noted by the average line. It is evident that the sum of the losses for

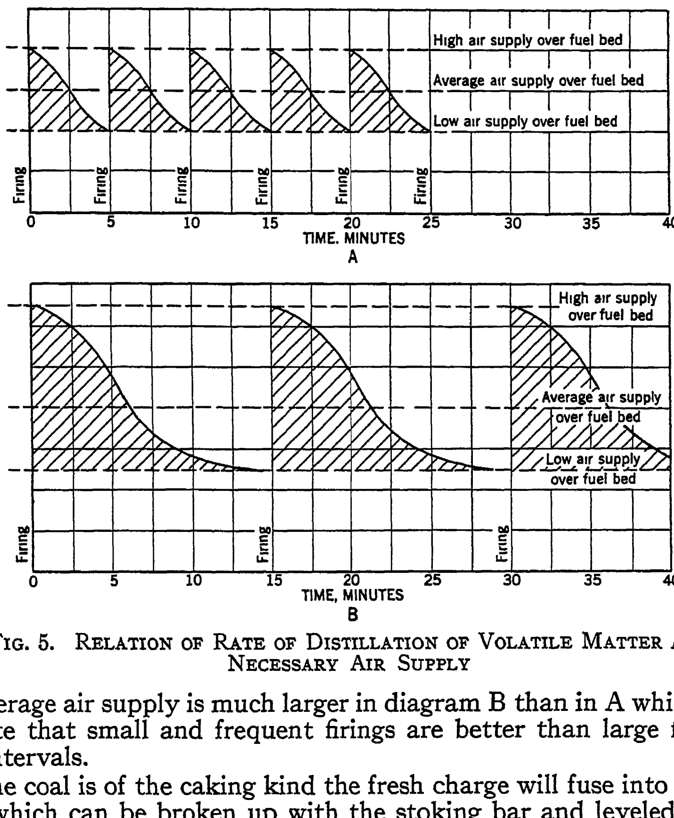


FIG. 5. RELATION OF RATE OF DISTILLATION OF VOLATILE MATTER AND NECESSARY AIR SUPPLY

the average air supply is much larger in diagram B than in A which would indicate that small and frequent firings are better than large firings at long intervals.

If the coal is of the caking kind the fresh charge will fuse into one solid mass which can be broken up with the stoking bar and leveled from 20 min to one hour after firing, depending on the temperature of the fire-box. Care should be exercised when stoking not to bring the bar up to the surface of the fuel as this will tend to bring ash into the high temperature zone at the top of the fire, where it will melt and form clinker. The stoking bar should be kept as near the grate as possible and should be raised only enough to break up the fuel. With fuels requiring stoking it may not be necessary to shake the grates, as the ash is usually dislodged during stoking.

It is acknowledged that it may be difficult to apply the outlined

methods to domestic heating boilers of small size, especially when frequent attendance is impractical. The adherence to these methods insofar as practical, however, will result in better combustion.

The output obtained from any heater with bituminous coal will usually exceed that obtainable with anthracite, since bituminous coal burns more rapidly than anthracite and with less draft. Bituminous coal, however, will require frequent attention to the fuel bed, because it burns unevenly, even though the fuel bed may be level, forming holes in the fire which admit too much air, chilling the gases over the fuel bed and reducing the available draft.

FIRING METHODS FOR SEMI-BITUMINOUS COAL

The *Pocahontas Operators Association* recommends the central cone method of firing, in which the coal is heaped on to the center of the bed forming a cone the top of which should be level with the middle of the firing door. This allows the larger lumps to fall to the sides, and the fines to remain in the center and be coked. The poking should be limited to breaking down the coke without stirring, and to gently rocking the grates. It is recommended that the slides in the firing door be kept closed, as the thinner fuel bed around the sides allows enough air to get through.

FIRING METHODS FOR COKE

Coke ignites less readily than bituminous coal and more readily than anthracite and burns rapidly with little draft. In order to control the air admitted to the fuel it is very important that all openings or leaks into the ashpit be closed tightly. A coke fire responds rapidly to the opening of the dampers. This is an advantage in warming up the system, but it also makes it necessary to watch the dampers more closely in order to prevent the fire from burning too rapidly. In order to obtain the same interval of attention as with other fuels a deep fuel bed always should be maintained when burning coke. The grates should be shaken only slightly in mild weather and should be shaken only until the first red particles drop from the grates in cold weather. The best size of coke for general use, for small firepots where the fuel depth is not over 20 in., is that which passes over a 1 in. screen and through a 1½ in. screen. For large firepots where the fuel can be fired over 20 in. deep, coke which passes over a 1 in. screen and through a 3 in. screen can be used, but a coke of uniform size is always more satisfactory. Large sizes of coke should be either mixed with fine sizes or broken up before using.

PULVERIZED COAL

Although several pulverized coal burning units for domestic heating plant firing have been developed, none has attained extended use. Two general methods of adaptation have been employed, one where the coal is pulverized by the unit at the furnace and one where the coal is delivered to the home in pulverized form.

FURNACE VOLUME

The principal requirements for a *hand-fired furnace* are that it shall have enough grate area and correctly proportioned combustion space. The

amount of grate area required is dependent upon the desired combustion rate.

The furnace volume is influenced by the kind of coal used. Bituminous coals, on account of their long-flaming characteristic, require more space in which to burn the gases of combustion completely than do the coals low in volatile matter. For burning high volatile coals provision should be made for mixing the combustible gases thoroughly, so that combustion is complete before the gases come in contact with the relatively cool heating surfaces. An abrupt change in the direction of flow tends to mix the gases of combustion more thoroughly. Anthracite requires practically no combustion space.

DUSTLESS TREATMENT OF COAL

The practice of treating the more friable coals to allay the dust they create is increasing. The coal is sprayed with petroleum products, particularly the lighter oils, a solution of calcium chloride or a mixture of calcium and magnesium chlorides. The latter salts are very hygroscopic and their moisture under normal atmospheric conditions keeps the surface of the coal damp, thus reducing the dust during delivery in the cellar, and obviating the necessity of sprinkling the coal in the bin.

The coal is usually treated at the mine, but sometimes by the local distributor just before delivery. The salt solutions are sprayed under high pressure, using from 2 to 4 gal or from 5 to 10 lb of the salt per ton of coal, depending on its friability and size. Oil for the dustless treatment of coal is also applied under high pressure, in concentrations of 1 to 8 qt per ton of coal, depending upon the characteristics of the coal and oil.

CLASSIFICATION OF OILS

The Commercial Standard Specifications for Fuel Oils (CS 12-40) of the *U. S. Department of Commerce* are given in Table 7. These specifications conform with *American Society for Testing Materials* Tentative Specifications for Fuel Oils D396-38T.

The specific gravity of oil is of interest in its relationship to the calorific value and these data are given in Table 8.

COMBUSTION OF OIL

With oil, as with any kind of fuel, efficient heat production requires that all combustible matter in the fuel shall be completely consumed and that it shall be done with a minimum of excess air. The combustion of oil is a rather rapid chemical reaction. Excess air provides an over supply of oxygen so that all of the oil, composed of carbon and hydrogen, will be completely oxidized and thus produce all the heat possible. The use of unreasonable quantities of air in excess of theoretical combustion requirements results in lowered efficiencies due to increased stack losses. Such losses, if not accompanied by unburned products of combustion (saturated and unsaturated hydrocarbons, hydrogen, etc.) may be offset somewhat by increasing the secondary heating surfaces of the heat absorbing medium boiler or furnace.

Oil is a highly concentrated fuel composed mainly of hydrogen and

TABLE 7. DETAILED REQUIREMENTS FOR FUEL OILS[a]

Grade[b]	Flash Point Deg F		Pour Point Deg F	Water and Sediment Per Cent	Carbon Residue Per Cent	Ash Per Cent	Distillation Temperatures Deg F: 10 Per Cent Point	90 Per Cent Point		End Point	Viscosity Seconds: Saybolt Universal at 100 F		Saybolt Furol at 122 F	
	Min.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Min.	Max.	Max.	Min.	Max.	Min.
No. 1 Fuel oil—a distillate oil for use in burners requiring a volatile fuel.	100 or Legal	165	0[c]	Trace	0.05 on 10% Residuum[d]		410			560[e]				
No. 2 Fuel oil—a distillate oil for use in burners requiring a moderately volatile fuel.	110 or Legal	190	10[c]	0.05	0.25 on 10% Residuum[f]		440	600						
No. 3 Fuel oil—a distillate oil for use in burners requiring a low viscosity fuel.	110 or Legal	230	20[c]	0.10	0.15 Straight			675	600[g]		45			
No. 5 Fuel oil—an oil for use in burners requiring a medium viscosity fuel.	130 or Legal			1.00		0.10						50	40	
No. 6 Fuel oil—an oil for use in burners equipped with preheaters permitting a high viscosity fuel.	150			2.00[h]									300	45

[a]Recognizing the necessity for low sulphur fuel oils used in connection with heat-treatment, non-ferrous metal, glass and ceramic furnaces and other special uses, a sulphur requirement may be specified in accordance with the following table:

Grade of Fuel Oil	Sulphur, Max. Per cent
No. 1	0.5
No. 2	0.5
No. 3	0.75
No. 5	No Limit
No. 6	No Limit

Other sulphur limits may be specified only by mutual agreement between the buyer and seller.

[b]It is the intent of these classifications that failure to meet any requirement of a given grade does not automatically place an oil in the next lower grade unless in fact it meets all requirements of the lower grade.

[c]Lower or higher pour points may be specified whenever required by conditions of storage or use. However, these specifications shall not require a pour point lower than 0 F under any conditions.

[d]For use in other than sleeve type blue flame burners carbon residue on 10 per cent residuum may be increased to a maximum of 0.12 per cent. This limit may be specified by mutual agreement between the buyer and seller.

[e]The maximum end point may be increased to 590 F when used in burners other than sleeve type blue flame burners.

[f]To meet certain burner requirements the carbon residue limit may be reduced to 0.15 per cent on 10 per cent residuum.

[g]The minimum distillation temperature of 600 F for 90 per cent may be waived if A.P.I. gravity is 26 or lower.

[h]Water by distillation, plus sediment by extraction. Sum, maximum 2.0 per cent. The maximum sediment by extraction shall not exceed 0.50 per cent. A deduction in quantity shall be made for all water and sediment in excess of 1.0 per cent.

carbon. In its liquid form oil cannot burn. It must be converted into a gas or vapor by some means. If the excess air is to be kept within efficient limits it means that air must be supplied in carefully regulated quantities. The air and oil vapor must be vigorously mixed to get a rapid and complete chemical reaction. The better the mixing, the less excess air will be needed. The combustion must take place in a space that maintains the temperatures high so the reaction will not be stopped before completion. When equipped with a means of igniting the oil, and safety devices to guard against mishaps, the oil burner becomes efficient and automatic.

CLASSIFICATION OF GAS

Gas is broadly classified as being either *natural* or *manufactured.* Natural gas is a mechanical mixture of several combustible and inert gases rather than a chemical compound. Manufactured gas as distributed is usually a combination of certain proportions of gases produced by two or more processes. Representative properties of gaseous fuels commonly used in domestic heating are presented in Table 9.

TABLE 8. APPROXIMATE GRAVITY AND CALORIFIC VALUE OF STANDARD GRADES OF FUEL OIL

COMMERCIAL STANDARD NO.	APPROXIMATE GRAVITY, RANGE BAUMÉ	CALORIFIC VALUE BTU PER GALLON
1	38-40	136,000
2	34-36	138,500
3	28-32	141,000
5	18-22	148,500
6	14-16	152,000

Natural gas is the richest of the gases and contains from 80 to 95 per cent methane, with small percentages of the other combustible hydrocarbons. In addition, it contains from 0.5 to 5.0 per cent of CO_2, and from 1 to 12 or 14 per cent of nitrogen. The heat value varies from 1000 to 1200 Btu per cubic foot, the majority of natural gases averaging about 1000 Btu per cubic foot. Table 9 shows typical values for the four main oil fields, although values from any one field vary materially.

Table 9 also gives the calorific values of the more common types of manufactured gas. Most states have legislation which controls the distribution of gas and fixes a minimum limit to its heat content. The gross or higher calorific value usually ranges between 520 and 545 Btu per cubic foot, with an average of 535. A given heat value may be maintained and yet leave considerable latitude in the composition of the gas so that as distributed the composition is not necessarily the same in different districts, nor at successive times in the same district. However, in any community the variations in gas composition are held within suitable limits so that the performance of approved gas appliances will not be adversely affected.

COMBUSTION OF GAS

The majority of gas burners utilized in central domestic heating plants are of the Bunsen type and operate with a non-luminous flame. In this

TABLE 9. REPRESENTATIVE PROPERTIES OF GASEOUS FUELS. BASED ON GAS AT 60 F AND 30 IN. HG.

Gas	Btu per Cu Ft		Specific Gravity, Air = 1.00	Air Required for Combustion, (Cu Ft)	Products of Combustion				Theoretical Flame Temperature, (deg F)
					Cubic Feet			Ultimate CO_2 Dry Basis	
	High (Gross)	Low (Net)			CO_2	H_2O	Total with N_2		
Natural gas—California	1200	1087	0.67	11.26	1.24	2.24	12.4	12.2	3610
Natural gas—Mid-Continental	967	873	0.57	9.17	0.97	1.92	10.2	11.7	3580
Natural gas—Ohio	1130	1025	0.65	10.70	1.17	2.16	11.8	12.1	3600
Natural gas—Pennsylvania	1232	1120	0.71	11.70	1.30	2.29	12.9	12.3	3620
Retort coal gas	575	510	0.42	5.00	0.50	1.21	5.7	11.2	3665
Coke oven gas	588	521	0.42	5.19	0.51	1.25	5.9	11.0	3660
Carbureted water gas	536	496	0.65	4.37	0.74	0.75	5.0	17.2	3815
Blue water gas	308	281	0.53	2.26	0.46	0.51	2.8	22.3	3800
Anthracite producer gas	134	124	0.85	1.05	0.33	0.19	1.9	19.0	3000
Bituminous producer gas	150	140	0.86	1.24	0.35	0.19	2.0	19.0	3160
Oil gas	575	510	0.35	4.91	0.47	1.21	5.6	10.7	3725

type of burner part of the air required for combustion is mixed with the gas as primary air, the air and gas mixture being fed to the burner ports. Additional secondary air is introduced around the flame by draft inspiration. In the luminous flame burner, which is sometimes used, all of the air for combustion is brought in contact with the flame as secondary air. The importance of bringing the secondary air into intimate contact with the gas is noted.

Some makes of burners use radiants or refractories to convert some of the energy in the gas to radiant heat. The radiants also serve as baffles in directing the flow of the products of combustion.

The quantity of air given in Table 9 is that required for theoretical combustion, but with a properly designed and installed burner the excess air can be kept low. The division of the air into primary and secondary is a matter of burner design and the pressure of gas available, and also of the type of flame desired.

The air gas ratio has a decided effect upon flame propagation. It is necessary that the gas will flow out of the burner ports fast enough so that the flame cannot travel back into the burner head, i.e. *flash back*, but the velocity must not be so high that it blows the flame away from the port.

The maximum and minimum flow speeds from burner ports which may be permitted are known to be very close together when air-gas mixtures in theoretical proportions are being supplied to the burner. As the air-gas ratio is lowered, and the mixture becomes more *gas rich*, the limiting speeds become further apart, until with 100 per cent gas, in an all-yellow flame, flash back cannot occur and a much higher velocity is needed to blow off the flames.

SOOT

The deposit of soot on the flue surfaces of a boiler or heater acts as an insulating layer over the surface and reduces the heat transmission to the water or air. The *Bureau of Mines Report of Investigations* No. 3272 shows that the loss of seasonal efficiency is not as great as has been believed and usually is not over 6 per cent because the greater part of the heat is transmitted through the combustion chamber surfaces. The *Bureau of Standards Report* BMS 54 points out that, although the decrease in efficiency of an oil fired boiler due to soot deposits is relatively small the attendant increase in stack temperature may become excessive.

The soot accumulation clogs the passages and reduces the draft; the loss of efficiency from this action may be considerably greater than from the reduction in heat transfer.

REFERENCES

Fuels and Their Combustion, Haslam and Russell (McGraw-Hill Co., 1926).

Principles of Combustion in the Steam Boiler Furnace, Arthur D. Pratt, (Babcock and Wilcox Co.).

Smoke-Producing Tendencies in Coals of Various Ranks, by H. J. Rose and F. P. Lasseter (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 329).

Bureau of Mines Publications:

Bulletin No. 97, Sampling and Analyzing Flue Gases, by Henry Kreisinger and F. K. Ovitz.

Report of Investigations (R.I. 2980), Coke as a Domestic Heating Fuel, by P. Nicholls and B. A. Landry.

Bulletin No. 276, Five Hundred Tests of Various Coals in House-heating Boilers, by P. Nicholls, S. B. Flagg and C. E. Augustine.

Report of Investigations (R.I. 3283), Quality of Anthracite as Prepared at Breakers, 1935.

Technical Paper No. 80, Hand Firing Soft Coal under Power-Plant Boilers, by Henry Kreisinger.

Technical Paper No. 303, Value of Coke, Anthracite, and Bituminous Coal for Generating Steam in a Low-pressure Cast-iron Boiler, by John Blizard, James Neil and F. C. Houghten.

National Bureau of Standards Publication:

Report BMS 54, Effect of Soot on the Rating of an Oil-Fired Heating Boiler, by R. S. Dill and P. R. Achenbach.

Anthracite Industries Laboratories Publications:

Report 2015, Comparison of Sizes, Egg, Stove and Chestnut Anthracite.

Report 2018, Domestic Survey.

Report 2062, Utilization of Anthracite for Domestic Heating.

Report 2204, The Crater Method of Firing.

Report 2403, Anthracite Industries Manual.

An Experimental Investigation of the Use of Oil for the Treatment of Coal, by Ralph A. Sherman and J. M. Pilcher (*A.S.M.E. Transactions*, February, 1938).

Handbook of Oil Burning, by Harry F. Tapp, *American Oil Burner Association.*

Industrial Gas Series, Combustion, *American Gas Association.*

Comfort Heating, *American Gas Association.*

Tests of Gas Home-Heating Equipment, by R. B. Leckie and C. H. B. Hotchkiss (Purdue University, *Engineering Bulletin Research Series* No. 36).

Chapter 9

CHIMNEYS AND DRAFT CALCULATIONS

Natural Draft, Mechanical Draft, Draft Control, Characteristics of Natural Draft Chimneys, Determining Chimney Sizes, General Equation, Domestic Chimneys, Construction Details, Chimneys for Gas Heating

DRAFT, in general, may be defined as the pressure difference between the atmospheric pressure and that at any part of an installation through which the gases flow. Since a pressure difference implies a head, draft is a static force. While no element of motion is inferred, yet motion in the form of circulation of gases throughout an entire boiler plant installation is the direct result of draft. This motion is due to the pressure difference, or unbalanced pressure, which compels the gases to flow. Draft is often classified into two kinds according to whether it is created thermally or artificially, such as, (1) natural or thermal draft, and (2) artificial or mechanical draft.

Natural Draft

Natural draft is the difference in pressure produced by the difference in weight between the relatively hot gases inside a natural draft chimney and an equivalent column of the cooler outside air, or atmosphere. Natural draft, in other words, is an unbalanced pressure produced thermally by a natural draft chimney as the pressure transformer and a temperature difference. The intensity of natural draft depends, for the most part, upon the height of the chimney above the grate bar level and also the temperature difference between the chimney gases and the atmosphere.

A typical natural draft system consists essentially of a relatively tall chimney built of steel, brick, or reinforced concrete, operating with the relatively hot gases which have passed through the boilers and accessories and from which all the heat has not been extracted. Hot gases are an essential element in the operation of a natural draft system, although inherently a heat balance loss.

A natural draft chimney performs the two-fold service of assisting in the creation of draft by aspiration and also of discharging the gases at an elevation sufficient to prevent them from becoming a nuisance.

Natural draft is quite advantageous in installations where the total loss of draft due to resistances is relatively low and also in plants which have practically a constant load and whose boilers are seldom operated above their normal rating. Natural draft systems have been, and are still being,

employed in the operation of large plants during the periods when the boilers are operated only up to their normal rating. When the rate of operation is increased above the normal rating, some form of mechanical draft is employed as an auxiliary to overcome the increased resistances or draft losses. Natural draft systems are used almost exclusively in the smaller size plants where the amount of gases generated is relatively small and it would be expensive to install and operate a mechanical draft system.

The principal advantages of natural draft systems may be summarized as follows: (1) simplicity, (2) reliability, (3) freedom from mechanical parts, (4) low cost of maintenance, (5) relatively long life, (6) relatively low depreciation, and (7) no power required to operate. The principal

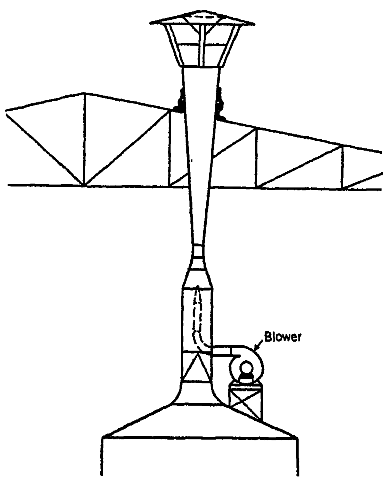


FIG. 1. DIAGRAM OF VENTURI EJECTOR

disadvantages are: (1) lack of flexibility, (2) irregularity, (3) affected by surroundings, and (4) affected by temperature changes.

Mechanical Draft

Artificial draft, or mechanical draft, as it is more commonly called, is a difference in pressure produced either directly or indirectly by a forced draft fan, an induced draft fan, or a Venturi chimney as the pressure transformer. The intensity of mechanical draft is dependent for the most part upon the size of the fan and the speed at which it is operated. The element of temperature does not enter into the creation of mechanical draft and therefore its intensity, unlike natural draft, is independent of the temperature of the gases and the atmosphere. The purpose of any mechanical draft system is to produce a difference in pressure between the point at which the air for combustion enters the boiler and the point at which the products of combustion leave the boiler. Such systems include the blower or fan type which produces a plenum or pressure above that

of the atmosphere under the fire and the exhaust fan and Venturi types which produce a partial vacuum that is minimum under the fire, and maximum at the point of exit of the products of combustion from the boiler. The latter types are known as induced draft systems. A mechanical draft system called a Venturi ejector[1] is illustrated in Fig. 1, in which the blower forces air, taken from the outside, through a Venturi tube which draws the gases from the furnace, boiler or hood. With this system, the hot or corrosive gases do not come in contact with the blower. A mechanical draft system may be used either in conjunction with, or as an adjunct to, a natural draft system.

Draft Control

To obtain the maximum efficiency of combustion, a definite minimum supply of air to the combustion chamber must be maintained. To pro-

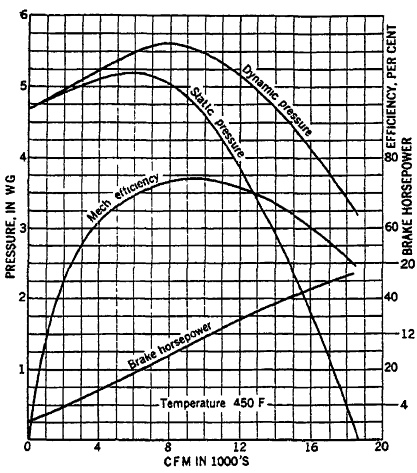


FIG. 2. GENERAL OPERATING CHARACTERISTICS OF TYPICAL INDUCED DRAFT FAN

vide this condition, it is necessary to have some automatic mechanical means of draft control or adjustment, because of variable wind velocities, fluctuations in atmospheric temperatures and barometric pressures, each of which has an effect upon draft.

For this purpose there are various mechanical devices which automatically control the volume of air admitted to the combustion chamber. Mechanical draft regulators designed to control or adjust draft should not be confused with mechanical draft systems that *create* draft mechanically, but which must also be automatically controlled.

The use of such a device, to provide a more uniform and dependable control of draft than could be maintained by manually operated dampers, will produce better combustion of fuel. This higher efficiency of combus-

[1]The Venturi Ejector for Handling Air, by F. F. Kravath (*Heating and Ventilating*, June, p. 17, August, p. 46, 1940).

tion, together with the reduced heat losses up the chimney by reason of decreased gas velocity, results in fuel economy, with consequent lower costs of plant operation.

CHARACTERISTICS OF CHIMNEYS

In order to analyze the performance of a natural draft chimney, it may be advantageous to compare its general operating characteristics with those of a centrifugal pump and also of a centrifugally-induced draft fan, there being a similarity among the three. Figs. 2, 3 and 4 show the general operating characteristics of a typical centrifugally-induced draft fan, a typical centrifugal pump, and a typical natural draft chimney, respectively. The draft-capacity curve of the chimney corresponds to

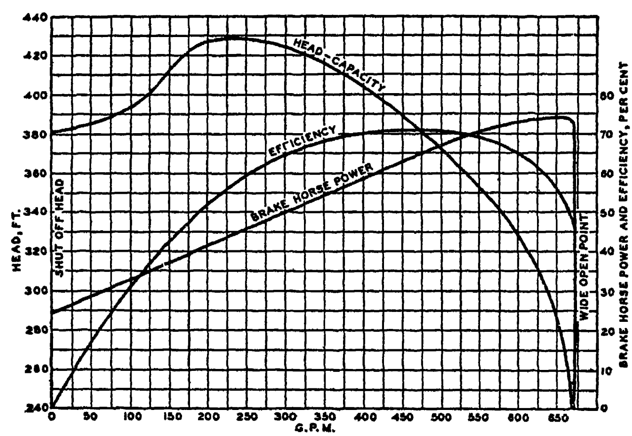


FIG. 3. OPERATING CHARACTERISTICS OF TYPICAL CENTRIFUGAL PUMP

the head-capacity curve of the pump and also to the dynamic-head-capacity curve of the fan.

When the gases in the chimney are stationary, the draft created is termed the *theoretical draft*. When the gases are flowing, the theoretical intensity is diminished by the draft loss due to friction, the difference between the two being termed the total *available draft*.

If pressures at the bases of a column of air and a column of chimney gas, each of height H feet; and d_o and d_c represent the respective densities of the air and the gas in pounds per cubic foot, then the theoretical draft D_t in pounds per square foot is:

$$D_t = d_oH - d_cH$$

Expressing the densities under standard conditions of pressure and temperature, and assuming that the absolute pressure of the gas is the same as that of the air, the theoretical draft becomes:

$$D_t = 15.36\ HB_o \left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right)$$

Expressed in inches of water this is:

$$D_t = 2.96\ HB_o \left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right)$$

The friction loss in the chimney may be determined from the Fanning equation:

$$\text{Head lost in feet of fluid} = \frac{fRL}{A} \times \frac{V^2}{2g}$$

in which R is the inside perimeter of the cross-section in feet, A the cross-section area in square feet, and V the velocity of fluid in feet per second.

Substituting for V its value in terms of W and the cross-section area, and expressing the loss of head in inches of water this becomes:

For cylindrical stacks

$$h_L = 0.01936 \frac{fLW^2}{D^5 d_c}$$

and for a rectangular stack of sides x and y in feet,

$$h_L = 0.00597 \frac{fLW^2 (x + y)}{\overline{xy}^3 d_c}$$

Substituting for d_c its value:

$$\frac{460 B_o W_c}{29.92 T_c}$$

gives for a cylindrical stack,

$$h_L = 0.00126 \frac{W^2 T_c fL}{D^5 B_o W_c}$$

and for a rectangular stack,

$$h_L = 0.000388 \frac{W^2 T_c fL (x + y)}{\overline{xy}^3 B_o W_c}$$

The available draft then is, for a cylindrical stack:

$$D_a = 2.96 H B_o \left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.00126 W^2 T_c fL}{D^5 B_o W_c} \tag{1}$$

and for a rectangular stack:

$$D_a = 2.96\, H B_o \left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.000388\, W^2 T_c fL (x + y)}{\overline{xy}^3 B_o W_c} \tag{2}$$

where

D_a = available draft, inches of water.
H = height of chimney above grate bars, feet.
B_o = barometric pressure corresponding to altitude, inches of mercury.
W_o = unit weight of a cubic foot of air at 0 F and sea level atmospheric pressure, pounds per cubic foot.
W_c = unit weight of a cubic foot of chimney gases at 0 F and sea level atmospheric pressure, pounds per cubic foot.
T_o = absolute temperature of atmosphere, degrees Fahrenheit.
T_c = absolute temperature of chimney gases, degrees Fahrenheit.
W = weight of gases generated in the combustion chamber of the boiler and passing through the chimney, pounds per second.
f = coefficient of friction.
L = length of friction duct of the chimney, feet.
D = minimum diameter of chimney, feet.

The first term of the right hand expression of Equation 1 represents the theoretical draft intensity, and the second term, the loss due to friction.

Example 1. Determine the available draft of a natural draft chimney 200 ft in height and 10 ft in diameter operating under the following conditions: atmospheric temperature, 62 F; chimney gas temperature, 500 F; sea level atmospheric pressure, B_0 = 29.92 in. of mercury; atmospheric and chimney gas density, 0.0863 and 0.09, respectively; coefficient of friction, 0.016; length of friction duct, 200 ft. The chimney discharges 100 lb of gases per second.

Substituting these values in Equation 1 and reducing:

$$D_a = 2.96 \times 200 \times 29.92 \times \left(\frac{0.0863}{522} - \frac{0.09}{960}\right) - \frac{0.00126 \times 100^2 \times 960 \times 0.016 \times 200}{10^5 \times 29.92 \times 0.09}$$

$$= 1.27 - 0.14 = 1.13 \text{ in.}$$

Fig. 4 shows the variation in the available draft of a typical 200 ft by 10 ft chimney operating under the general conditions noted in Example 1. When the chimney is under static conditions and no gases are flowing, the

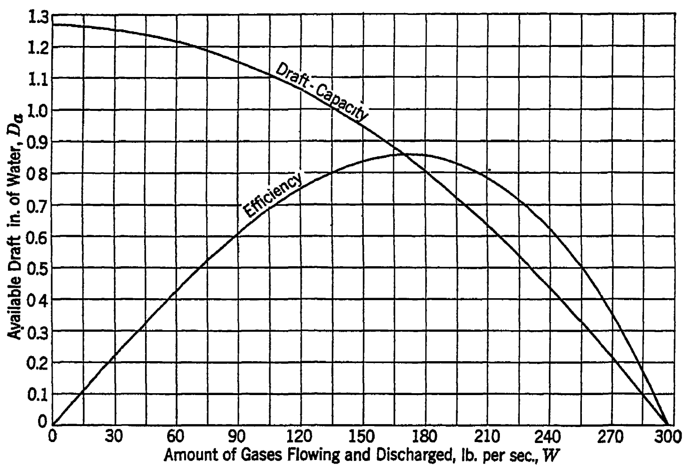


FIG. 4. TYPICAL SET OF OPERATING CHARACTERISTICS OF A NATURAL DRAFT CHIMNEY

available draft is equal to 1.27 in. of water, the theoretical intensity. As the amount of gases flowing increases, the available intensity decreases until it becomes zero at a gas flow of 297 lb per second, at which point the draft loss due to friction is equal to the theoretical intensity. The draft-capacity curve corresponds to the head-capacity curve of centrifugal pump characteristics and the dynamic-head-capacity curve of a fan. The point of maximum draft and zero capacity is called shut-off draft, or point of impending delivery, and corresponds to the point of shut-off head of a centrifugal pump. The point of zero draft and maximum capacity is called the wide open point and corresponds to the wide open point of a centrifugal pump. A set of operating characteristics may be developed for any size chimney operating under any set of conditions by substituting the proper values in Equation 1 and then plotting the results in the manner shown in Fig. 4.

In substituting the values for the various factors in Equation 1, care should be exercised that the selections be as near the actual conditions as

is practically possible. The following notes will serve as a guide for these selections:

1. The *barometric pressure*, represented by B_0, is the actual pressure at the site of the chimney and not the pressure reduced to sea level datum.

In general, the barometric pressure decreases approximately 0.1 in. of mercury per 100 ft increase in elevation.

2. The *unit weight of a cubic foot* of chimney gases at 0 F and sea level barometric pressure is given by the equation:

$$W_c = 0.131 CO_2 + 0.095\, O_2 + 0.083\, N_2 \tag{3}$$

In this equation CO_2, O_2 and N_2 represent the percentages of the parts by volume of the carbon dioxide, oxygen and nitrogen content, respectively, of the gas analysis. For ordinary operating conditions, the value of W_c may be assumed at 0.09.

The density effect on the chimney gases due to superheated water vapor resulting from moisture and hydrogen in the fuel, or due to any air infiltrations in the chimney proper are disregarded. Though water vapor content is not disclosed by Orsat analysis, its presence tends to reduce the actual weight per cubic foot of chimney gases.

3. The *atmospheric temperature* is the actual observed temperature of the outside air at the time the analysis of the operating chimney is made. The mean atmospheric temperature in the temperate zone is approximately 62 F.

4. The *chimney gas temperature* decreases from the breeching connection to the top of the stack. This drop in temperature depends upon the material and construction of the stack, its tightness or freedom from leaks, its area, its height, and the velocity of the gases through it. The same chimney will suffer different temperature losses depending upon the capacity under which it is working and the variable atmospheric conditions. No general equation covering all these variables has been suggested, but from observations on chimneys varying in diameter from 3 to 16 ft and in height from 100 to 250 ft the following equation was deduced[2];

$$T_c = \frac{3.13\, T_1 \left[\left(\frac{H_b}{3}\right)^{0.96} - 1\right]}{H_b - 3} \tag{4}$$

where

T_1 = absolute temperature at the center of the connection from the breeching, degrees Fahrenheit.

H_b = the height of the stack above center line connection to breeching, feet.

5. The *coefficient of friction* between the chimney gases and a sooted surface has been taken by many workers in this field as a constant value of 0.016 for the conditions involved. This value, of course, would be less for a new unlined steel stack than for a brick or brick-lined chimney, but in time the inside surface of all chimneys regardless of the materials of construction becomes covered with a layer of soot, and thus the coefficient of friction has been taken the same for all types of chimneys and in general constant for all conditions of operation. For reasons of simplicity and convenience to the reader, this constant value of 0.016 has been employed in the development of the various special equations and charts shown in this chapter.

In important chimney design, especially when the construction or the materials are unusual, it is recommended that use be made of Reynolds number[3] in determining the friction factor, f.

6. The *length of the friction duct* is the vertical distance between the bottom of the breeching opening and the top of the chimney. Ordinarily this distance is approximately equal to the height of the chimney above the grate level.

7. Assuming no air infiltration the *amount of gases flowing and being discharged* is, of course, equal to the amount of gases generated in the combustion chamber of the

[2]Notes on Power Plant Design, by E. F. Miller and James Holt (*Massachusetts Institute of Technology*, 1930).

[3]For more complete discussion see Flow of Fluids in Closed Conduits, by R. J. S. Pigott (*Mechanical Engineering*, August, 1933).

boiler. The total products of combustion in pounds per second for a grate-fired boiler may be computed from the equation:

$$W = \frac{C_g G W_{tp}}{3600} \tag{5}$$

where

C_g = pounds of fuel burned per square foot of grate surface per hour.
G = total grate surface of boilers, square feet.
$C_g \times G$ = total weight of fuel burned per hour.
W_{tp} = total weight of products of combustion per pound of fuel.

A similar computation may be made in the case of gas, oil, or stoker-fired fuel.

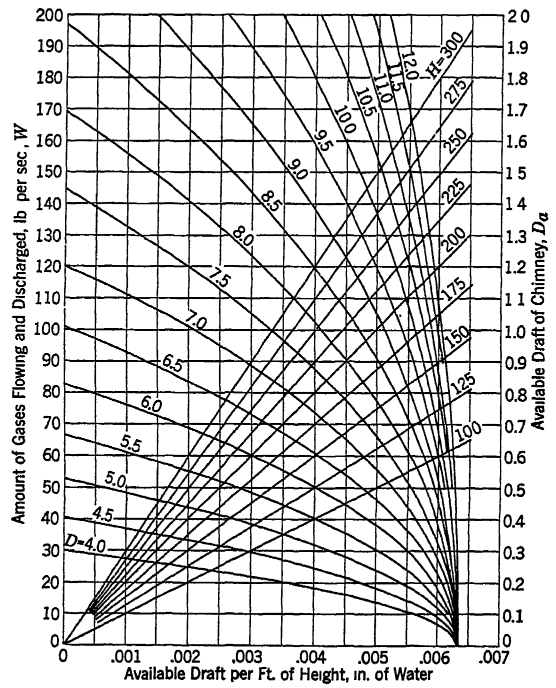


FIG. 5. CHIMNEY PERFORMANCE CHART[a]

[a]*To solve a typical example:* Proceed horizontally from a Weight Flow Rate point to intersection with diameter line; from this intersection follow vertically to chimney height line; from this intersection follow horizontally to the right to Available Draft scale. Starting from a point of Available Draft, take steps in reverse order.

Fig. 5 is a typical chimney performance chart giving the available draft intensities for various amounts of gases flowing and sizes of chimney. This chart is based on an atmospheric temperature of 62 F, a chimney gas temperature of 500 F, a unit chimney gas weight of 0.09 lb per cubic foot, sea level atmospheric pressure, a coefficient of friction of 0.016, and a friction duct length equal to the height of the chimney above the grate level. These curves may be used for general operating conditions. For specific conditions, a new chart should be prepared from Equation 1.

It has been the usual custom, and still is to a lamentably great extent, to select the required size of a natural draft chimney from a table of

chimney sizes based only on boiler horsepowers. After the ultimate horsepower of the projected plant had been determined, the chimney size in the table corresponding to this figure was then selected as the proper size required. Generally, no further attempt was made to determine if the height thus selected was sufficient to help create the required draft demanded by the entire installation, or the diameter sufficiently large to enable the chimney quickly, efficiently, and economically to dispose of the gases. Since the operating characteristics of a natural draft chimney are similar in all respects to those of a centrifugal pump, or a centrifugal fan, it is no more possible to select a proper size chimney from such a table, even with correction factors appended, than it is to select the proper size pump from tables based only on the amount of water to be delivered.

DETERMINING CHIMNEY SIZES

The required diameter and height of a natural draft cylindrical chimney are given by the following equations:

$$H = \frac{D_T}{2.96B_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.184fW_cB_oV^2}{T_cD}} \tag{6}$$

The weight of gas per second,

$$W = 12.075 \frac{D^2 VB_o W_c}{T_c}$$

from which

$$D = 0.288 \sqrt{\frac{WT_c}{B_oW_cV}} \tag{7}$$

where

H = required height of chimney above grate bar level, feet.
D = required minimum diameter of chimney, feet (constant for entire height).
V = chimney gas velocity, feet per second.
D_T = total required draft demanded by the entire installation outside of the chimney, inches of water.

Equations 6 and 7 give the required size of a natural draft chimney with all of the operating factors taken into consideration. Values for all of the factors with the exception of the chimney gas velocity may be either observed or computed. It is, of course, necessary to assume an arbitrary value for the velocity in order to arrive at some definite size. For any one set of operating conditions there will be as many sizes of chimney as there are values of reasonable velocities to assume. Of the number of sizes corresponding to the various assumed velocities, there is one size which will be least expensive. Since the cost of a chimney structure, regardless of the kind of material used in the construction, varies as the volume of material in the structure, the cost criterion then may be represented by the approximate equation:

$$Q = \pi tHD \tag{8}$$

where

Q = volume of material, cubic feet.
t = average wall thickness, feet.

For all practical purposes, the value of πt may be taken as a constant regardless of the size of the structure. Hence, in general, the volume, and consequently the cost, of a chimney structure may be based on the factor HD as a criterion. Therefore, the value of the chimney gas velocity which will result in the least value of HD for any one set of operating conditions will produce a structure which will be the most economical to use, because its cost will be least.

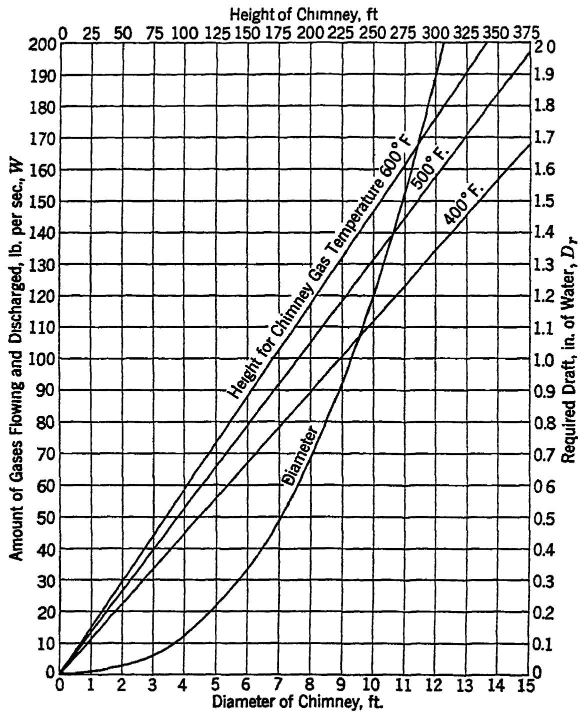


FIG. 6. ECONOMICAL CHIMNEY SIZES[a]

[a]Diameter values also for gas temperatures of 400, 500 and 600 F

The problem is to deduce an equation for the chimney gas velocity which will result in a combination of a height and a diameter whose product HD will be least. The solution is obtained by equating the product of Equations 6 and 7 to HD, differentiating this product with respect to V and equating the resulting expression to zero. This procedure results in the following expression:

$$V_e = \left(\frac{0.772 T_c \left(\frac{W_o}{T_o} - \frac{W_c}{T_c} \right) \sqrt{\frac{W T_c}{B_o W_c}}}{f W_c} \right)^{2/5} \tag{9}$$

where V_e = economical chimney gas velocity, feet per second.

Equation 9 gives the economical velocity of the chimney gases for any set of operating conditions, and represents the velocity which will

result in a chimney the size of which will cost less than that of any other size as determined by any other velocity for the same operating conditions. After the value of the economical velocity has been determined, the corresponding height and diameter can then be determined from Equations 6 and 7, respectively, and the economical size will then be attained. Equations 6, 7 and 9 may be simplified considerably for average operating conditions in an average size steam plant by assuming typical conditions.

Average chimney gas temperature, 500 F .... $T_c = 960$
Mean atmospheric temperature, 62 F .... $T_o = 522$
Average coefficient of friction, 0.016 .... $f = 0.016$
Average chimney gas density, 0.09 .... $W_c = 0.09$
Sea level elevation, with barometer of 29.92 .... $B_o = 29.92$

Substituting these values in Equations 9, 7 and 6, respectively, and reducing, the results are substantially:

$$V_e = 13.7W^{1/5} \tag{10}$$

$$D = 1.5W^{2/5} \tag{11}$$

$$H = 190D_r \tag{12}$$

Fig. 6 gives the economical chimney sizes for various amounts of gases flowing and for required draft intensities as computed from Equations 10, 11 and 12. They are based on the operating factors used in reducing Equations 6, 7 and 9 to their simpler form. The sizes shown by the curves in the chart should be used for general operating conditions only, or for installations where the required data necessary for an exact determination are difficult or impossible to secure. Whenever it is possible to secure accurate data, or the anticipated operating conditions are fairly well known, the required size should be determined from Equations 6, 7 and 9. The recommended minimum inside dimensions and heights of chimneys for small and medium size installations are given in Table 1.

GENERAL EQUATION

The general draft equation for a steam producing plant may be stated as follows:

$$D_t - h_f = h_F + h_B + h_{Bd} + h_C + h_{Br} + h_V + h_O + h_E + h_R \tag{13}$$

where

D_t = theoretical draft intensity created by pressure transformer, inches of water.
h_f = draft loss due to friction in pressure transformer, inches of water.
h_F = draft loss through the fuel bed, inches of water.
h_B = draft loss through the boiler and setting, inches of water.
h_{Br} = draft loss through the breeching, inches of water.
h_V = draft loss due to velocity, inches of water.
h_{Bd} = draft loss due to bends, inches of water.
h_C = draft loss due to contraction of opening, inches of water.
h_O = draft loss due to enlargement of opening, inches of water.
h_E = draft loss through the economizer, inches of water.
h_R = draft loss through recuperators, regenerators, or air heaters, inches of water.

The left hand member of Equation 13 represents the total amount of available draft created by the pressure transformer, that is, the natural draft chimney, Venturi chimney, or fan, and is equal to the theoretical intensity less the internal losses incidental to operation. The right hand member represents the sum of all of the various losses of draft throughout the entire boiler plant installation outside of the pressure transformer itself. The left hand member expresses the available intensity and is analogous to the head developed by a centrifugal pump in a water works system, while the right hand member expresses the required draft in-

Table 1. Recommended Minimum Chimney Sizes for Heating Boilers and Furnaces[a]

Warm Air Furnace Capacity in Sq In. of Leader Pipe	Steam Boiler Capacity Sq Ft of Radiation	Hot Water Heater Capacity Sq Ft of Radiation	Nominal Dimensions of Fire Clay Lining in Inches	Rectangular Flue: Actual Inside Dimensions of Fire Clay Lining in Inches	Rectangular Flue: Actual Area Sq In.	Round Flue: Inside Diameter of Lining in Inches	Round Flue: Actual Area Sq In	Height in Ft Above Grate
790	590	973	8½ x 13	7 x 11½	81			35
1000	690	1,140				10	79	
	900	1,490	13 x 13	11¼ x 11¼	127			
	900	1,490	8½ x 18	6¾ x 16¼	110			
	1,100	1,820				12	113	40
	1,700	2,800	13 x 18	11¼ x 16¼	183			
	1,940	3,200				15	177	
	2,130	3,520	18 x 18	15¾ x 15¾	248			
	2,480	4,090	20 x 20	17¼ x 17¼	298			45
	3,150	5,200				18	254	50
	4,300	7,100				20	314	
	4,600	7,590	20 x 24	17 x 21	357			
	5,000	8,250	24 x 24	21 x 21	441			55
	5,570	9,190		24 x 24[b]	576			60
	5,580	9,200				22	380	
	6,980	11,500				24	452	65
	7,270	12,000		24 x 28[b]	672			
	8,700	14,400		28 x 28[b]	784			
	9,380	15,500				27	573	
	10,150	16,750		30 x 30[b]	900			
	10,470	17,250		28 x 32[b]	896			

[a]This table is taken from the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929).

[b]Dimensions are for unlined rectangular flues.

tensity and is analogous to the total dynamic head in a water works system. For a general circulation of gases

$$D_a = D_r \tag{14}$$

where

D_a = available draft intensity, inches of water.
D_r = required draft, inches of water.

The *draft loss through the fuel bed* (h_F), or the amount of draft required to effect a given or required rate of combustion, varies between wide limits and represents the greater portion of the required draft. In coal-fired

installations, the draft loss through the fuel bed is dependent upon the following factors: (1) character and condition of the fuel, clean or dirty; (2) percentage of ash in the fuel; (3) volume of interstices in the fuel bed, coarseness of fuel; (4) thickness of the fuel bed, rate of combustion; (5) type of grate or stoker used; (6) efficiency of combustion.

There is a certain intensity of draft with which the best results will be obtained for every kind of coal and rate of combustion. Fig. 7 gives the intensity of draft, or the vacuum in the combustion chamber required to burn various kinds of coal at various rates of combustion. Expressed in other words, these curves represent the amount of draft required to force the necessary amount of air through the fuel bed in order to effect various rates of combustion. It will be noted that the amount of draft increases as the percentage of volatile matter diminishes, being comparatively low

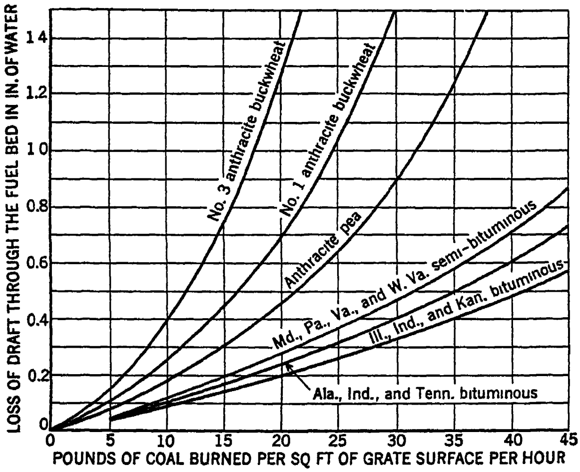


FIG. 7. DRAFT REQUIRED AT DIFFERENT RATES OF COMBUSTION FOR VARIOUS KINDS OF COAL

for the lower grades of bituminous coals and highest for the high grades and small sizes of anthracites. Also, when the interstices of the coal are large and the particles are not well broken up, as with bituminous coals, much less draft is required than when the particles are small and are well broken up, as with bituminous slack and the small sizes of anthracites. In general, the draft loss through the fuel bed increases as: (1) the percentage of volatile matter diminishes; (2) the percentage of fixed carbon increases; (3) the thickness of the bed increases; (4) the percentage of ash increases; (5) the volume of the interstices diminishes.

In making the preliminary assumptions for the draft loss through the fuel bed, due allowances should be made for a possible future change in the grade of fuel to be burned and also in the rate of combustion. A value should be selected for this loss which will represent not only the highest rate of combustion which will be encountered, but also the grade of coal which has the greatest resistance through the fuel bed and which may be burned at a later date.

In powdered-fuel and oil-fired installations, there will be no draft loss through the fuel bed since there is none and, consequently, this factor becomes zero in the general draft equation. All other factors being constant, the height of the chimney in installations of this character will be less than the height in coal-fired installations, and in the case of mechanical draft installations the driving units need not be as large since the head against which the fan is to operate is not as great in the former as in the latter.

The *draft loss through the boiler and setting* (h_b) also varies between wide limits and, in general, depends upon the following factors: (1) type of boiler, (2) size of boiler, (3) rate of operation, (4) arrangement of tubes, (5) arrangement of baffles, (6) type of grate, (7) design of brickwork setting, (8) excess air admitted, and (9) location of entrance into breeching.

Curves showing the draft loss through the boiler are usually based on the load or quantity of gases passing through the boiler, expressed in terms of percentage of normal rate of operation. Owing to the great variety of boilers of different designs and the various schemes of baffling, it is impossible to group together a set of curves for the draft loss through the boiler which may even be used generally. It is therefore necessary to secure this information from the manufacturer of the particular type of boiler and baffle arrangement under consideration.

When a boiler is installed and in operation, the draft loss depends upon the amount of gases flowing through it. This, in turn, depends upon the proportion of excess air admitted for combustion. Primarily, the amount of excess air is measured by the CO_2 content; the less the amount of CO_2, the greater the amount of excess air and hence the greater the draft loss.

The loss of draft through the boiler will vary directly as the size of the boiler and the length of the gas passages within. The loss also varies as the number of tubes high, but not in a direct ratio inasmuch as the loss due to the reversal of flow at the ends of the baffles remains constant regardless of the height of the boiler. The arrangement of the tubes, whether the gases flow parallel to or at right angles to the tubes, has an appreciable effect on the loss. The arrangement of the baffles influences the draft loss greatly, the loss through a boiler with five passes being greater than the loss through one of three or four passes. A poor design and a rough condition of the brickwork will increase the loss greatly, whereas a proper design and a smooth condition will keep the loss at a minimum. The loss through the boiler will be less when the breeching entrance is located at or near the top of the boiler than when it is located at or near the bottom since the gases have a shorter distance to travel in the former instance.

The *draft loss through the breeching* (h_{Br}) may be found by applying the last term on the right, with the sign changed, of Equation 1 or 2 depending upon whether the breeching is cylindrical or rectangular and observing the following changes in the symbols:

T_c = absolute temperature of breeching gases, degrees Fahrenheit.
f = coefficient of friction for the breeching.
L = length of breeching, feet.
D = diameter of cylindrical breeching, feet.
x and y = sides of breeching, if rectangular, feet.

It has been the general custom to *lump off* the intensity of the breeching loss at 0.10 in. of water per 100 ft of breeching length regardless of its size or shape or the amount and temperature of the gases flowing through it. This practice is hazardous and has no more foundation in fact than that of determining the friction head in a water works system without taking into consideration the size of the pipe or the amount of water flowing through it. When the length of the breeching is relatively short, any variation in any one of the factors in the equation will have no appreciable effect on the draft loss. However, when the breeching is relatively long, the draft loss is affected greatly by the various factors, particularly by the size and shape as well as by the weight of gases flowing.

The *draft loss due to velocity* (h_V) is given by the equation

$$h_V = \frac{0.000194 W^2 T_c}{A^2 B_o W_c} \qquad (15)$$

where

A = cross-section area at the top of the stack, square feet.

The *draft loss due to bends in the breeching* (h_{Bd}) is dependent upon the center line radius of curvature of the bends and the form of the cross-section. This loss is expressed in terms of the velocity head. (See Fig. 4, Chapter 32.)

The *draft loss due to sudden contraction of an area* (h_C) is given by the equation:

$$h_C = \frac{0.000194 K_c W^2 T_c}{A_s^2 B_o W_c} \qquad (16)$$

where

K_c = coefficient of sudden contraction based on $\frac{A_s}{A_1}$, the ratio of the areas of the smaller to the larger section $= 0.5\left(1 - \frac{A_s}{A_1}\right)$

A_s = area of the smaller section.

When the flue or passage through which the gases flow is suddenly contracted, a considerable portion of the static head in the larger section is converted into velocity head and a draft loss of some consequence, particularly in a short breeching, takes place. A sudden contraction should always be avoided where possible. At times, however, due to obstructions or limited head-room, it is necessary to alter the size of the breeching, but a sudden contraction may be avoided by gradually decreasing the area over a length of several feet.

The *draft loss due to a sudden enlargement of an area* (h_O) is given by the equation:

$$h_O = \frac{0.000194 K_o W^2 T_c}{A_s^2 B_o W_c} \qquad (17)$$

where

K_o = coefficient of sudden enlargement based on $\frac{A_s}{A_1}$, the ratio of the areas of the smaller to the larger section $= \left(1 - \frac{A_s}{A_1}\right)^2$

When the flue or passage through which the gases flow is suddenly enlarged, a portion of the velocity head is converted into static head in the

larger section and, like the loss due to sudden contraction, a loss of some consequence, particularly in short breechings, takes place. A sudden enlargement in a breeching may be avoided by gradually increasing the area over a length of several feet. In large masonry chimneys, the area of the flue at the region of the breeching entrance is considerably larger than the area of the breeching at the chimney, and a sudden enlargement exists.

The *draft loss through the economizer* (h_E) should be obtained from the manufacturer but for general purposes it may be computed from the following general equation:

$$h_E = \frac{6.6 W_n^2 N T_c}{10^{12}} \qquad (18)$$

where

W_n = pounds of gases flowing per hour per linear foot of pipe in each economizer section.

N = number of economizer sections.

An economizer in a steam plant affects the draft in two ways, (1) it offers a resistance to the flow of gases, and (2) it lowers the average chimney gas temperature, thereby decreasing the available intensity. In the case of a natural draft installation, both of these factors result in a relative increase in the height of the chimney and, in the case of a large plant, they may add as much as 20 or 30 ft to the height. The decrease in the temperature of the gases after they have passed through the economizer has an extremely important effect on the performance of a natural draft chimney and also upon the performance of a fan.

GENERAL CONSIDERATIONS FOR DOMESTIC CHIMNEYS

The draft of domestic chimneys may be subject to a variety of influences not usually encountered in power chimneys, such as buildings in the immediate vicinity which may be higher than the chimney, trees, and even hills as well as the shape of the roof of the building which the chimney serves.

Horizontal winds have an aspirating effect as they pass across the chimney and are an aid to draft providing they remain horizontal at the chimney top. However, surrounding objects, such as trees or other buildings, may greatly affect the direction of the wind at the chimney top and may even direct it down the chimney, tending to reduce the draft or even to cause it to be negative. Although the chimney should, in general, extend well above the highest part of the roof, it is impracticable to carry it much beyond this point.

It is also important to consider the source of the air supply for proper combustion. Usually the boiler or furnace is located in the basement or cellar and perhaps, as a general thing, when the furnace room has windows or doors opening to the outside on two or more sides of the house, the leakage of air around the windows and doors will be sufficient for combustion, even though the windows and doors may be shut. However, if the leakage is not sufficient to prevent an appreciable drop of pressure in the furnace room below that of the air outside, the chimney draft will be reduced by the difference between the atmospheric pressure outside and

that inside the boiler room. In case the boiler room is fairly tight against leaks, as is desirable from the standpoint of dust, and is open to the outside on only one side of the house, then the draft will be affected in windy weather even with windows or doors open. If the wind is blowing toward the boiler room the draft will be increased, but if blowing in the opposite direction the draft may be seriously decreased.

It is not to be assumed that increasing the cross-section area of a chimney will always effect a cure for poor draft. The opposite result may be experienced because of the cooling effect of the larger area. This reduces the theoretical draft and the velocity of the gases, and affords a greater opportunity for counter currents in the chimney. The only practical remedy for a chimney with bad draft, when the chimney is of the proper size and is affected by conditions beyond the control of the owner, is to resort to mechanical draft. This can usually be done at small expense if operated only when necessary.

CHIMNEYS FOR GAS HEATING

The burning of gas differs from the burning of coal in that the force which supplies the air for combustion of the gas comes largely from the pressure of the gas in the supply pipe, whereas air is supplied to a bed of burning coal by the force of the chimney draft. If, with a coal-burning boiler, the draft is poor, or if the chimney is stopped, the fire is smothered and the combustion rate reduced. In a gas boiler or furnace such a condition would interfere with the combustion of the gas, but the gas would continue to pass to the burners and the resulting incomplete combustion would produce a dangerous condition. In order to prevent incomplete combustion from insufficient draft, all gas-fired boilers and furnaces should have a back-draft diverter in the flue connection to the chimney.

A study of a typical *back-draft diverter* shows that partial or complete chimney stoppage will merely cause some of the products of combustion to be vented out into the boiler room, but will not interfere with combustion. In fact, gas-designed appliances must perform safely under such a condition to be approved by the *American Gas Association* Laboratory. Other functions of the back-draft diverter are to protect the burner and pilot from the effects of down-drafts, and to neutralize the effects of variable chimney drafts, thus maintaining the appliance efficiency at a substantially constant value.

As is the case with the complete combustion of almost all fuels, the

Table 2. Minimum Round Chimney Diameters for Gas Appliances (Inches)

Height of Chimney Feet	Gas Consumption in Thousands of Btu per Hour								
	100	200	300	400	500	750	1000	1500	2000
20	4.50	5.70	6.60	7.30	8.00	9.40	10.50	12.35	13.85
40	4.25	5.50	6.40	7.10	7.80	9.15	10.25	12.10	13.55
60	4.10	5.35	6.20	6.90	7.60	8.90	10.00	11.85	13.25
80	4.00	5.20	6.00	6.70	7.35	8.65	9.75	11.50	12.85
100	3.90	5.00	5.90	6.50	7.20	8.40	9.40	11.00	12.40

products of combustion for gas are carbon dioxide (CO_2) and water vapor with just a trace of sulphur trioxide (SO_3). Sulphur usually burns to the trioxide in the presence of an iron oxide catalyst. The volume of water vapor in the flue products is about twice the volume of the carbon dioxide when coke oven or natural gas is burned. Because of the large quantity of water vapor which is formed by the burning of gas, it is quite important that all gas-fired central heating plants be connected to a chimney having a good draft. Lack of chimney draft causes stagnation of the products of combustion in the chimney and results in the condensation of a large amount of the water vapor. A good chimney draft draws air through the openings in the back-draft diverter, lowers the dew-point of the mixture, and reduces the tendency of the water vapor to condense.

The flue connections from a gas-fired boiler or furnace should be of a non-corrosive material. The material used for the flue connection should not only be resistant to the corrosion of water but should resist the corrosion of dilute solutions of sulphur trioxide. Local practice should be followed in the selection of the most appropriate flue materials.

When condensation in a chimney proves troublesome, it may be necessary to provide a drain to a dry well or sewer. The cause of the excessive condensation should be investigated and remedied if possible. The protection of unlined chimneys has been investigated and the results indicate that after the loose material has been removed, the spraying with a water emulsion of asphalt-chromate provides an excellent protection.

A chimney for a gas-fired boiler or furnace should be constructed similarly to the principles applicable to other boilers. Table 2 gives the minimum cross-sectional diameters of round chimneys (in inches) for various amounts of heat supplied to the appliance, and for various chimney heights. This is in accordance with *American Gas Association* recommendations.

CONSTRUCTION DETAILS

For general data on the construction of chimneys reference should be made to the Standard Ordinance for Chimney Construction of the *National Board of Fire Underwriters*. Briefly summarized, these provisions are as follows for heating boilers and furnaces:

The walls of brick chimneys shall be not less than 3¾ in. thick (width of a standard size brick) and shall be lined with fire-clay flue lining meeting the standard specification of the *Eastern Clay Products Association*. The flue sections shall be set in special mortar, and shall have the joints struck smooth on the inside. The masonry shall be built around each section of lining as it is placed, and all spaces between masonry and linings shall be completely filled with mortar. No broken flue lining shall be used. Flue lining shall start at least 4 in. below the bottom of smoke-pipe intakes of flues, and shall be continued the entire heights of the flues and project at least 4 in. above the chimney top to allow for a 2 in. projection of lining.

Flue lining may be omitted in brick chimneys, provided the walls of the chimneys are not less than 8 in. thick, and that the inner course shall be a refractory clay brick. All brickwork shall be laid in spread mortar, with all joints push-filled. Exposed joints both inside and outside shall be struck smooth. No plaster lining shall be permitted.

Chimneys shall extend at least 3 ft above flat roofs and 2 ft above the ridges of peak roofs when such flat roofs or peaks are within 30 ft of the chimney. The chimney shall be high enough so that the wind from any direction shall not strike the top of the chimney from an angle above the horizontal. The chimney shall be properly capped, but no such cap or coping shall decrease the flue area.

Chapter 10

AUTOMATIC FUEL BURNING EQUIPMENT

Classification of Stokers, Combustion Process and Adjustments, Furnace Design, Classification of Oil Burners, Combustion Chamber Design, Classification of Gas-Fired Appliances

AUTOMATIC mechanical equipment for the combustion of solid, liquid and gaseous fuels is considered in this chapter.

MECHANICAL STOKERS

A mechanical stoker is a device that feeds a solid fuel into a combustion chamber, provides a supply of air for burning the fuel under automatic control and, in some cases, incorporates a means of removing the ash and refuse of combustion automatically. Coal can be burned more efficiently by a mechanical stoker than by hand firing because the stoker provides a uniform rate of fuel feed, better distribution in the fuel bed and positive control of the air supplied for combustion.

Stokers may be divided into four types according to their construction, namely, (1) overfeed flat grate, (2) overfeed inclined grate, (3) underfeed side cleaning type, and (4) underfeed rear cleaning type.

Overfeed Flat Grate Stokers

This type is represented by the various chain- or traveling-grate stokers. These stokers receive fuel at the front of the grate in a layer of uniform thickness and move it back horizontally to the rear of the furnace. Air is supplied under the moving grate to carry on combustion at a sufficient rate to complete the burning of the coal near the rear of the furnace. The ash is carried over the back end of the stoker into an ashpit beneath. This type of stoker is suitable for small sizes of anthracite or coke breeze and also for bituminous coals, the characteristics of which make it desirable to burn the fuel without disturbing it. This type of stoker requires an arch over the front of the stoker to maintain ignition of the incoming fuel. Frequently, a rear combustion arch is required to maintain ignition until the fuel is fully consumed. A typical traveling-grate stoker is illustrated in Fig. 1.

Another and distinct type of overfeed flat-grate stoker is the spreader (Figs. 2 and 3) or sprinkler type in which coal is distributed either by

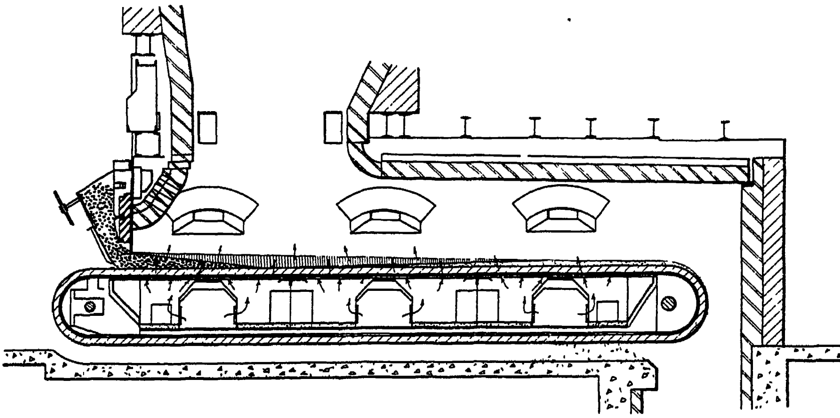


FIG. 1. OVERFEED TRAVELING-GRATE STOKER

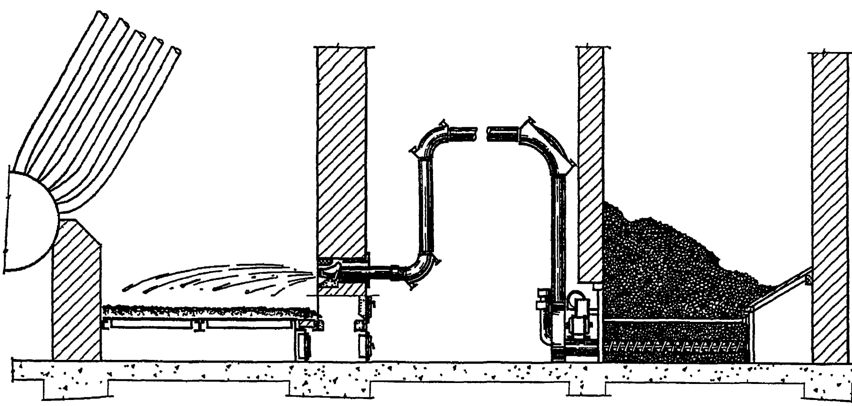


FIG. 2. OVERFEED SPREADER STOKER, PNEUMATIC TYPE

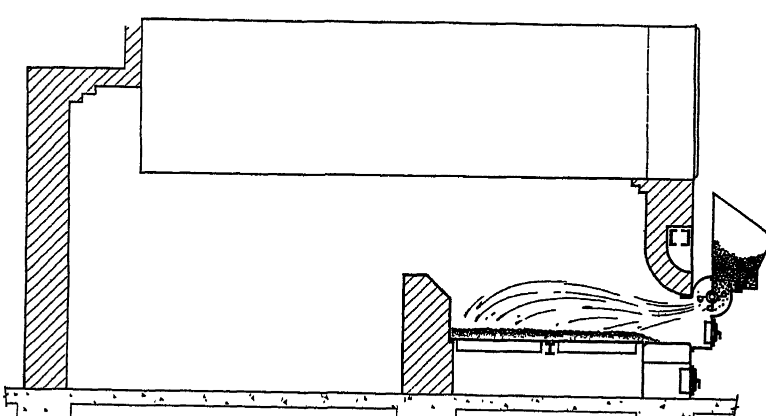


FIG. 3. OVERFEED SPREADER STOKER, ROTOR TYPE

rotating paddles or by air over the entire grate surface. This type of stoker has a wide application on small sized fuels and on fuels such as lignites, high-ash coals, and coke breeze.

Overfeed Inclined Grate Stokers

In general the combustion principle is similar to the flat-grate stoker, but this stoker is provided with rocking grates set on an incline to advance the fuel during combustion. Also this type is provided with an ash plate where ash is accumulated and from which it is dumped periodically. This type of stoker is suitable for all types of coking fuels but preferably for those of low volatile content. Its grate action has the tendency to keep the fuel bed well broken up thereby allowing for free passage of air. Because of its agitating effect on the fuel it is not so desirable for badly clinkering coals. Furthermore, it should usually be provided with a front arch to care for the volatile gases.

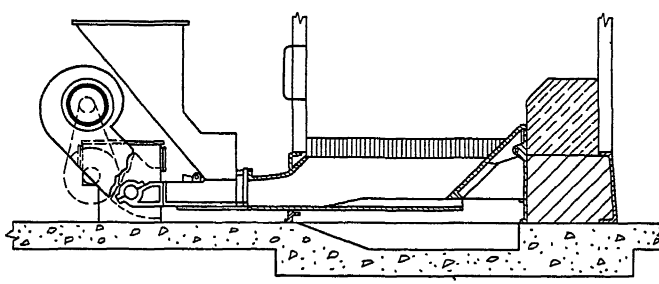


FIG. 4. UNDERFEED SIDE CLEANING STOKER

Underfeed Side Cleaning Stokers

In this type (Fig. 4), the fuel is introduced at the front of the furnace to one or more retorts, and is advanced away from the retort as combustion progresses, while finally the ash is disposed of at the sides. This type of stoker is suitable for all bituminous coals while in the smaller sizes it is suitable for small sizes of anthracite. In this type of stoker the fuel is delivered to a retort beneath the fire and is raised into the fire. During this process the volatile gases are released, are mixed with air, and pass through the fire where they are burned. The ash may be continuously discharged as in the small stoker or may be accumulated and periodically discharged. This stoker requires no arch as it automatically provides for the combustion of the volatile gases.

Underfeed Rear Cleaning Stokers

This type of stoker accomplishes combustion in much the same manner as the side cleaning type, but consists of several retorts placed side by side and filling up the furnace width, while the ash disposal is at the rear. In principle, its operation is the same as the side cleaning underfeed type of stoker.

CLASSIFICATION OF STOKERS ACCORDING TO CAPACITY

Stokers may be classified according to their capacity or coal feeding rates. The following classification has been made by the *U. S. Department of Commerce*, in cooperation with the *Stoker Manufacturers Association.*

Class 1. Capacity under 61 lb of coal per hour.
Class 2. Capacity 61 to 100 lb of coal per hour.
Class 3. Capacity 101 to 300 lb of coal per hour.
Class 4. Capacity 300 to 1200 lb of coal per hour.
Class 5. Capacity 1200 lb of coal per hour and over.

Class 1 Stokers

These stokers are used primarily for home heating and are, therefore, designed for quiet, automatic operation. Simple, trouble-free construc-

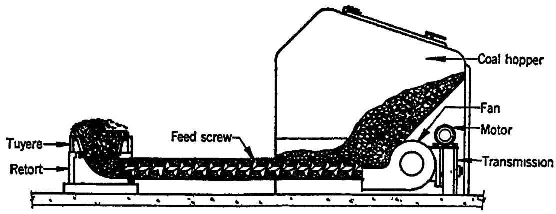


FIG. 5. UNDERFEED STOKER, HOPPER TYPE, CLASS 1

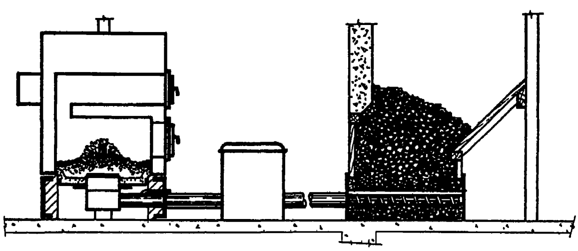


FIG. 6. UNDERFEED STOKER, BIN FEED TYPE, CLASS 1

tion and attractive appearance are very desirable characteristics of these small units. Equipment of this capacity is also used extensively for commercial and industrial applications requiring the burning of small quantities of fuel under automatic control.

A common type of stoker in this class (Fig. 5) consists essentially of a coal reservoir or hopper, a screw for conveying the coal from the reservoir to the burner head or retort, a fan which supplies the air for combustion, a transmission for driving the coal feed worm, and an electric motor or motors for supplying the motive power for both coal feed and air supply.

Air for combustion is admitted to the fuel through tuyeres at the top of the retort and in this class, the tuyeres and retort are usually round, although they may be either round or rectangular. Stokers in this class are made for burning anthracite, bituminous, semi-bituminous, and lignite coals and coke. The *U. S. Department of Commerce* has issued commercial standards for household anthracite stokers[1].

[1]Household Anthracite Stoker Standards (*U. S. Department of Commerce, National Bureau of Standards,* Commercial Standard No. CS48-40).

Units are available in either the hopper type, as shown in Fig. 5, or in the type as shown in Figs. 6 and 7, which feeds the coal directly to the furnace from the coal bin. Some stokers, particularly those designed for use with anthracite coal, have equipment for automatically removing the ash from the ash pit and depositing it in an ash receptacle outside of the furnace, as shown in Fig. 7. Most of the bituminous models, however, operate on the principle of removing the ash from the fuel bed after it is fused into a clinker at the outer periphery of the tuyere.

Most of the stokers in this class feed coal to the furnace intermittently in accordance with temperature or pressure demands. A small amount of heat is also released from the fuel bed during the inoperative period of the stoker. Through the use of automatic controls (see Chapter 34), it is possible to maintain temperatures or pressures within very close limits when using stoker firing equipment.

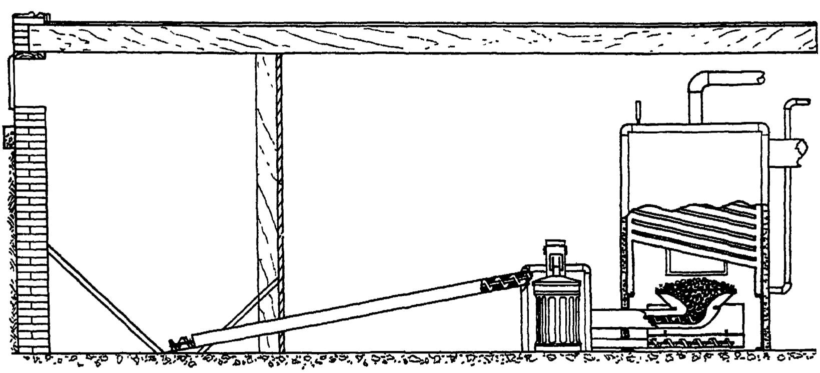


FIG. 7. UNDERFEED ANTHRACITE STOKER WITH AUTOMATIC ASH REMOVAL, BIN TYPE

Stoker-Fired Boiler and Furnace Units

Boilers, air conditioners, and space heaters especially designed for stokers are now available having certain design features closely coordinating the heat absorber and the stoker. Although very efficient and satisfactory performance can be obtained from the application of automatic stokers to existing boilers and furnaces, some of the combination stoker-fired units (Fig. 8) are more compact and attractive in appearance in addition to other design features.

Class 2 and 3 Stokers

Stokers in this class are usually of the screw feed type without auxiliary plungers or other means of distributing the coal. They are used extensively for heating plants in apartments and hotels, also, for industrial plants, such as laundries, bakeries, and creameries.

They are primarily of the underfeed type and are available in both the hopper type, as illustrated in Fig. 9, and also, the bin feed type, which delivers the coal directly to the furnace from the coal bin, as illustrated

in Fig. 10. These units are also built in a plunger feed type and the drive for the coal feed may be an electric motor or a steam or hydraulic cylinder.

Stokers in this class are available for burning all types of anthracite, bituminous and lignite coals. The tuyere and retort design varies widely according to the fuel and load conditions. On the bituminous models, the grates are normally of the stationary type and the ash accumulates on the grates surrounding the retort. With the average bituminous coal, the ash then fuses into a clinker which is removed periodically.

The anthracite stokers in this class are normally equipped with moving grates which discharge the ash into a pit below the grate. This ash pit may be located on one or both sides of the grate and on some installations is made of sufficient capacity to hold the ash from several days or weeks operation.

Class 4 Stokers

Stokers in this group vary widely in details of mechanical design and the several methods of feeding coal previously described are employed.

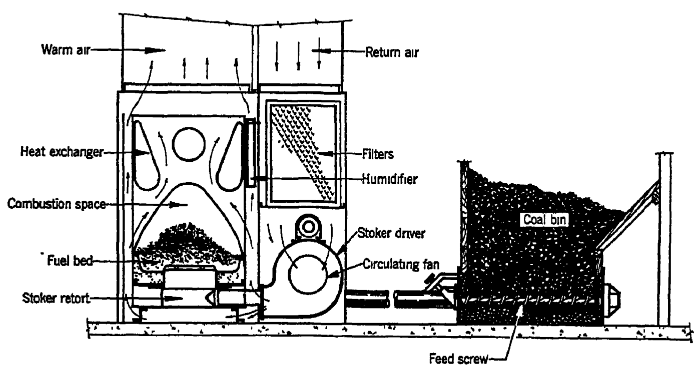


FIG. 8. STOKER-FIRED WINTER AIR CONDITIONING UNIT

The underfeed stoker is the most widely used, although a number of the overfeed types are also used in the larger sizes of this class. Bin feed, as well as hopper models, are available in both the underfeed and overfeed types.

Class 5 Stokers

The prevalent stokers in this field are: (*a*) overfeed flat grate, (*b*) overfeed inclined grate, (*c*) underfeed side cleaning, and (*d*) underfeed rear cleaning.

The rear cleaning underfeed stoker is usually of the multiple retort design and is used in some of the largest industrial plants and central power stations. In some instances, zoned air control has been applied on these stokers, both longitudinally and transversely of the grate surface.

Underfeed side cleaning stokers are made in sizes up to approximately 500 boiler horsepower. They are not so varied in design as those in the smaller classes, although the principle of operation is much the same. The overfeed spreader type stoker (Figs. 2 and 3) is adaptable to a wide

variety of coals and is being extensively used in capacities up to 1000 boiler horsepower.

Combustion Process

Due to the marked differences in design and operating characteristics of stokers and the widely different characteristics of stoker coals, it is difficult to generalize on the subject of combustion in automatic stokers.

In anthracite stokers of the small Class 1 overfeed type, burning takes place entirely within the stoker retort and tuyere. The ash and refuse of combustion spills over the edge of the tuyere into an ashpit or receptacle from which it may be removed either manually or automatically (Fig. 8).

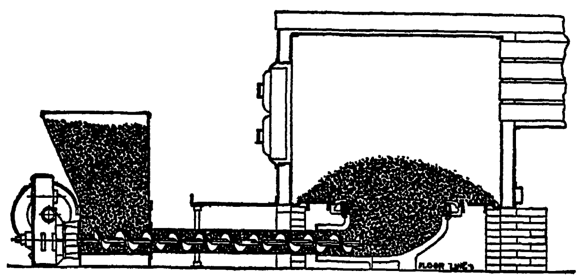


FIG. 9. UNDERFEED SCREW STOKER, HOPPER TYPE, CLASS 2, 3 OR 4

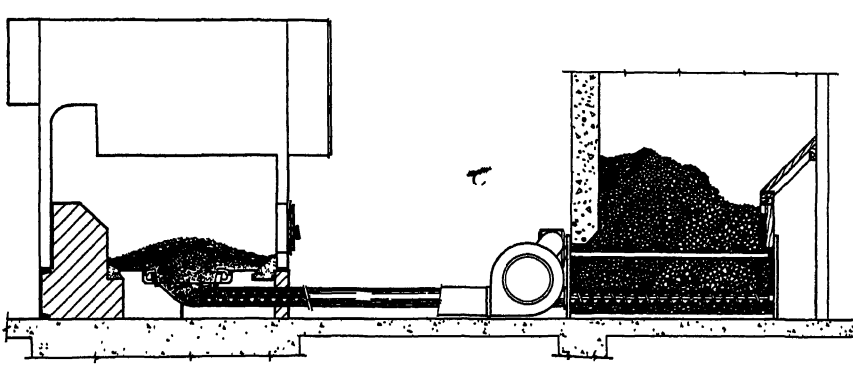


FIG. 10. UNDERFEED SCREW STOKER, BIN TYPE, CLASS 2, 3 OR 4

The larger underfeed anthracite stokers operate on the same principle except that the retort is rectangular and the refuse spills over only one or two sides of the grate. Anthracite for stoker firing is usually supplied in the No. 1 buckwheat or No. 2 buckwheat size.

Since the majority of the smaller bituminous coal stokers operate on the underfeed principle, a general description of their operation will be given. When the coal is fed from the hopper or bin into the retort, it moves upward toward the zone of combustion and is heated by conduction and radiation from the burning fuel in the combustion zone. As the temperature of the coal rises, it gives off moisture and occluded gases, which are largely non-combustible. When the temperature increases to

around 700 or 800 F the coal particles become plastic, the degree of plasticity varying with the type of coal.

A rapid evolution of the combustible volatile matter occurs during and directly after the plastic stage of the coal. The distillation of volatile matter continues above the plastic zone where the coal is coked. The strength and porosity of the coke formed will vary according to the size and characteristics of the coal used. While some of the ash fuses into particles in the surface of the coke as it is released, most of it remains on the hearth or grates and as this ash layer becomes thicker with time, that portion exposed to the higher temperatures surrounding the retort normally fuses into a clinker. The temperature attained in the fuel bed, the chemical composition and homogeneity of the ash, and the time of heating are factors which govern the degree of fusion.

Most bituminous coal stokers of Classes 1, 2 and 3 operate on the principle of the removal of the ash in this clinker form. Clinker tongs are provided to facilitate removal of the clinker on the smaller models.

There are a number of factors which materially affect the burning rate and the combustion results obtained with automatic stokers, the most important of these being the type and design of stoker, the characteristics of the fuel, and the manner in which the stoker is installed and operated.

Furnace Design

Due to the wide differences in stoker, boiler and furnace design, and fuel burning characteristics, it has not been found practical to establish fixed rules for the proportioning of furnaces for automatic coal stokers. It is, therefore, essential that an experienced stoker installer give careful consideration to these factors when applying the equipment.

The *Stoker Manufacturers Association* has published standard recommendations on setting heights for stokers having capacities up to 1200 lb of coal per hour.[2]

The empirical formula for determining these setting heights are:

For burning rates up to 100 lb coal per hour.

$$H = 0.1125B + 15.75 \quad (1)$$

For burning rates from 100 to 1200 lb coal per hour.

$$H = 0.03\,B + 24 \quad (2)$$

where

H = minimum setting height, inches.
B = burning rate coal per hour, pounds.

In considering these recommendations, it should be clearly understood that they show merely the average recommended minimum. There are many factors affecting the proper application of stokers to various types of boilers and furnaces and in many instances greater or less setting heights may give far better performance than the average values that are shown. They cannot, therefore, be used as arbitrary values in specifying stoker settings, and may be varied considerably by the installer based on

[2]Minimum Setting Heights. Copies of this standard may be obtained from the *Stoker Manufacturers Association*, 307 North Michigan Ave., Chicago, Ill.

experience with a particular stoker equipment, the type of coal that is to be used, and the construction of the boiler or furnace.

Combustion Adjustments

The coal feeding rate and air supply to the stoker should be regulated so as to maintain as nearly as possible an ideal balance between the load demand and the heat liberated by the fuel. Under such conditions no manual attention to the fuel bed should be required, other than the removal of clinker in stokers which operate on this principle of ash removal.

As in all combustion processes, the problem of maintaining the correct proportions of air and fuel is of utmost importance. It is desirable to supply a minimum amount of air required to properly burn the fuel at the rate it is being fed to the furnace.

While there may be only slight variations in the specified rate at which the coal is being fed to the furnace, due to variations in the size or density of the coal being used, there may be wide variations in the rate of air supplied as the result of changes in fuel bed resistance. These changes in resistance may be caused by changes in the porosity of the fuel bed due to variations in size or friability of the coal, ash and clinker accumulation, and variations in depth of the fuel bed. Because of this variable fuel bed resistance, many of the bituminous stokers, even in the smaller domestic sizes, incorporate air controls which automatically compensate for these changes in resistance and maintain a constant air fuel ratio.

It is also desirable on most stoker installations to provide automatic draft regulation in order to reduce air infiltration and provide better control during the banking or off periods of the stoker. The efficiency of combustion may be determined by analyzing with an Orsat apparatus the gases formed by the combustion process. With this equipment the percentage by volume of carbon dioxide (CO_2), oxygen (O_2), and carbon monoxide (CO) in the flue gases may be obtained. The percentage of CO_2 indicates the amount of excess air supplied. The presence of CO indicates a loss due to incomplete combustion, as the result of a deficiency of air or the improper mixing of air in the gases of combustion.

Sizing Stokers and Stoker Ratings

The capacity or rating of small underfeed stokers is usually stated as the burning rate in pounds of coal per hour. The *Stoker Manufacturers Association* has adopted a uniform method of rating stokers which is published in convenient tables and charts for selecting the size of stoker required.[3] The required capacity of the stoker may be calculated as follows:

$$\frac{\text{Load (Btu per hour)}}{\text{Heating value of coal (Btu per pound)} \times \text{overall efficiency of stoker and boiler or furnace}} = \text{Stoker burning rate required (pounds of coal per hour)}$$

In determining the total load placed on a stoker-fired boiler by a steam or hot water heating system, a piping and pick-up factor of 1.33 is com-

[3]Uniform Stoker Rating Code. Copies of this standard may be obtained from the *Stoker Manufacturers Association*, 307 North Michigan Ave., Chicago, Ill.

monly used in sizing the stoker, but this factor may be increased at times due to unusual conditions.

Controls

The heat delivery from the stoker of the smallest household type to the largest industrial unit can be accurately regulated with fully automatic controls. The smaller heating applications are normally controlled by a thermostat placed in the building to be heated. Limit controls are supplied to prevent excessive temperature or pressure being developed in the furnace or boiler and refueling controls are used to maintain ignition during periods of low heat demand. Automatic low water cut-outs are recommended for use with all automatically fired steam boilers. (See Chapter 34.)

DOMESTIC OIL BURNERS

An oil burner is a mechanical device for producing heat automatically and safely from liquid fuels. This heat is produced in the furnace or fire-pot of hot water or steam boilers or warm air furnaces and is absorbed by the boiler, and thus made available for distribution to the house through the heating system.

The number of combinations of the characteristic elements of domestic oil burners is rather large and accounts for the variety of burners found in actual practice. Domestic oil burners may be classified as follows:

1. AIR SUPPLY FOR COMBUSTION
   - *a. Atmospheric*—by natural chimney draft.
   - *b. Mechanical*—electric-motor-driven fan or blower.
   - *c. Combination of* (*a*) *and* (*b*)—primary air supply by fan or blower and secondary air supply by natural chimney draft.
2. METHOD OF OIL PREPARATION
   - *a. Vaporizing*—oil distills on hot surface or in hot cracking chamber.
   - *b. Atomizing*—oil broken up into minute globules.
     - (1) Centrifugal—by means of rotating cup or disc.
     - (2) Pressure—by means of forcing oil under pressure through a small nozzle or orifice.
     - (3) Air or steam—by high velocity air or steam jet in a special type of nozzle.
     - (4) Combination air and pressure—by air entrained with oil under pressure and forced through a nozzle.
   - *c. Combination of* (*a*) *and* (*b*).
3. TYPE OF FLAME
   - *a. Luminous*—a relatively bright flame. An orange-colored flame is usually best if no smoke is present.
   - *b. Non-luminous*—Bunsen-type flame (*i.e.*, blue flame).
4. METHODS OF IGNITION
   - *a. Electric.*
     - (1) Spark—by transformer producing high-voltage sparks. Usually shielded to avoid radio interference. May take place continuously while the burner is operating (continuous ignition) or just at the beginning of operation (intermittent ignition).
     - (2) Resistance—by means of hot wires or plates.

b. Gas.
  (1) Continuous—pilot light of constant size.
  (2) Expanding—size of pilot light expanded temporarily at the beginning of burner operation.

c. Combination—electric sparks light the gas and the gas flame ignites the oil.

d. Manual—by manually-operated gas torch for continuously operating burners.

5. MANNER OF OPERATION

a. On and off—burner operates at fixed firing rate for period determined by load demand.

b. High and low—burner operates continuously but varies from a high to a low flame.

c. Graduated—burner operates continuously but flame is graduated according to needs by regulating both air and oil supply.

A trade classification of domestic oil burners consists of the following general types: (*a*) gun or pressure atomizing, (*b*) rotary and (*c*) pot or

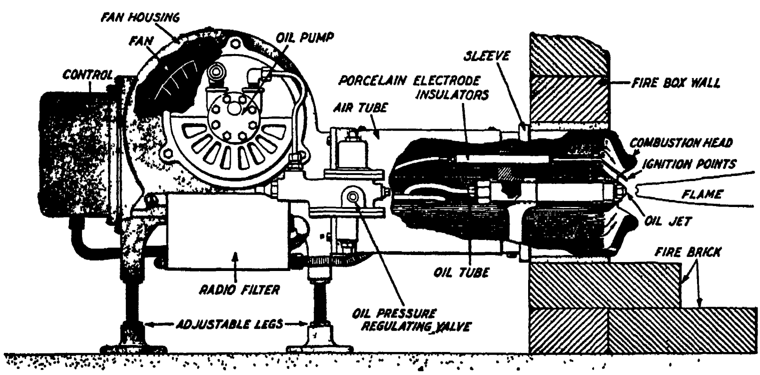


FIG. 11. GUN TYPE PRESSURE ATOMIZING OIL BURNER

vaporizing. These are further classified as mechanical draft and natural draft based on method used to supply the air for combustion.

The gun type, illustrated in Fig. 11, is characterized by an air tube, usually horizontal, with oil supply pipe centrally located in the tube and arranged so that a spray of atomized oil is introduced and mixed in the combustion chamber with the air stream emerging from the air tube. A variety of patented shapes are employed at the end of the air tube to influence the direction and speed of the air and thus the effectiveness of the mixing process.

The most distinguishing feature of vertical rotary burners is the principle of flame application. These burners are of two general types: the center flame and wall flame. In the former type (Fig. 12), the oil is atomized by being thrown from the rim of a revolving disc or cup and the flame burns in suspension with a characteristic yellow color. Combustion is supported by means of a bowl-shaped chamber or hearth. The wall flame burner (Fig. 13) differs in that combustion takes place in a ring of refractory material, which is placed around the hearth. These types of burners are further characterized by their installation within the ash pit of the boiler or furnace.

The pot type burner (Fig. 14) can be identified by the presence of a metal structure, called a pot or retort, in which combustion takes place.

When gun type (pressure atomizing) or horizontal rotary burners are used the combustion chamber is usually constructed of firebrick or other suitable refractory material, such as stainless steel, and is part of the installation procedure.

Most oil burners are operated by a small electric motor which pumps the oil and some or all of the air required. The smallest sizes can generally

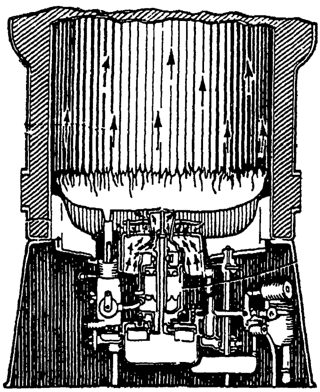


FIG. 12. CENTER FLAME VERTICAL ROTARY BURNER

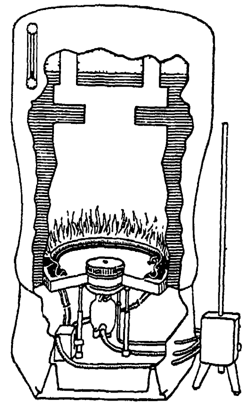


FIG. 13. WALL FLAME VERTICAL ROTARY BURNER

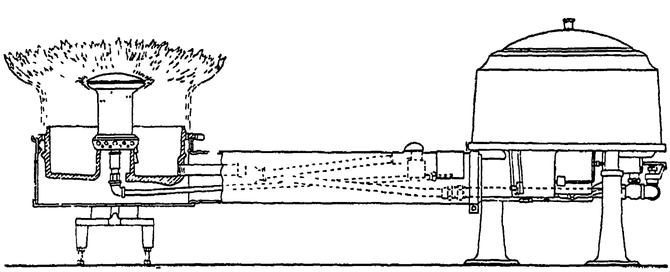


FIG. 14. POT TYPE VAPORIZING BURNER

burn not much less than from 0.50 to 0.75 gal of oil per hour. The grade of oil burned ranges from No. 1 to No. 3. No. 3 oil is the heaviest and most viscous of the various grades mentioned. An oil burner satisfactory for No. 3 oil can burn any of the lighter grades easily but an oil burner recommended for No. 2 oil should never be supplied with the heavier grades. While the heavier grades of oil have a smaller heat value per pound, they have, due to greater density, a larger heat value per gallon. The relative economy of the various grades must be based upon price and the amount of excess air required for clean and efficient combustion.

Operating Requirements for Mechanical Draft Oil Burners

The *U. S. Department of Commerce* in conjunction with the oil burner industry has established commercial standards for automatic mechanical

draft oil burners for domestic installations which cover installation requirements and performance tests[4].

Oil-Fired Boiler and Furnace Units

Boilers and furnaces especially designed for oil burners are available. This type of equipment usually has more heating surface than the older coal-burning designs. Flue proportions and gas travel have been changed with beneficial results.

COMMERCIAL OIL BURNERS

Liquid fuels are used for heating apartment buildings, hotels, public and office buildings, schools, churches, hospitals, department stores, as well as industrial plants of all kinds. Contrary to domestic heating, con-

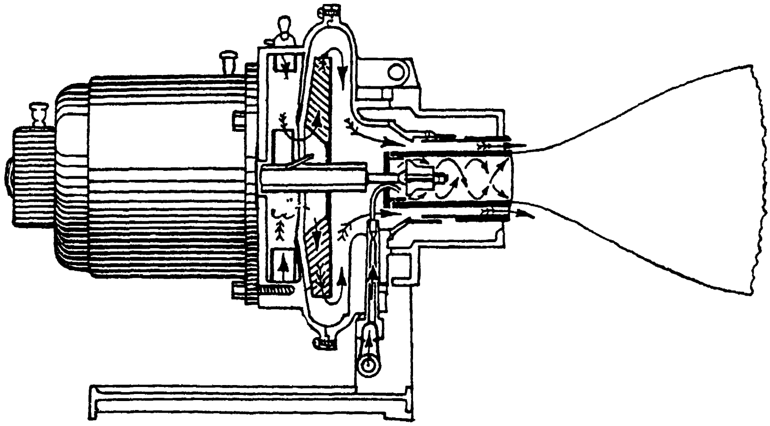


FIG. 15. HORIZONTAL ROTATING-CUP OIL BURNER

venience seldom is a dominating factor, the actual net cost of heat production usually controlling the selection of fuel. Some of the largest office buildings have been using oil for many years. Many department stores have found that floor space in basements and sub-basements can be used to better advantage for merchandising wares, and credit the heat producing department with this saving.

Wherever possible, the boiler plant should be so arranged that either oil or solid fuel can be used at will, permitting the management to take advantage of changes in fuel costs if any occur. Each case should be considered solely in the light of local conditions and prices.

Burners for commercial heating may be either large models of types used in domestic heating, or special types developed to meet the conditions imposed by the boilers involved. Generally speaking, such burners are of the mechanical or pressure atomizing types, the former using rotating cups which throw the oil from the edge of the cup at high velocity into the surrounding stream of air delivered by the blower (Fig. 15).

[4]Automatic Mechanical Draft Oil Burners Designed for Domestic Installations (*U. S. Department of Commerce, National Bureau of Standards*, Commercial Standard No. CS75-39).

As much as 350 gal of oil per hour can be burned in these units, and frequently they are arranged in multiple on the boiler face, from two to five burners to each boiler.

The larger installations are nearly always started with a hand torch, and are manually controlled, but the use of automatic control is increasing, and completely automatic burners are now available to burn the two heaviest grades of oil. Nearly all of the smaller installations, in schools, churches, apartment houses and the like, are fully automatic.

Because of the viscosity of the heavier oils, it is customary to heat them before transferring by truck tank. It also has been common practice to preheat the oil between the storage tank and the burner, as an aid to movement of the oil as well as to atomization. This heating is accomplished by heat-transfer coils, using water or steam from the heating boiler.

Unlike the domestic burner, units for large commercial applications frequently consist of atomizing nozzles or cups mounted on the boiler front with the necessary air regulators, the pumps for handling the oil and the blowers for air supply being mounted in sets adjacent to the boilers. In such cases, one pump set can serve several burner units, and common prudence dictates the installation of spare or reserve pump sets. Pre-heaters and other essential auxiliary equipment also should be installed in duplicate.

Boiler Settings

As the volume of space available for combustion is the determining factor in oil consumption, it is general practice to remove grates and extend the combustion chamber downward to include or even exceed the ashpit volume; in new installations the boiler should be raised to make added volume available. Approximately 1 cu ft of combustion volume should be provided for every developed boiler horsepower, and in this volume from 1.5 to 2.5 lb of oil per hour can properly be burned. This corresponds to an average liberation of about 38,000 Btu per cubic foot per hour. There are indications that at times much higher fuel rates may be satisfactory. For best results, care should be taken to keep the gas velocity below 40 ft per second. Where checkerwork of brick is used to provide secondary air, good practice calls for about 1 sq in. of opening for each pound of oil fired per hour. Such checkerwork is best adapted to flat flames, or to conical flames that can be spread over the floor of the combustion chamber. The proper bricking of a large or even medium sized boiler for oil firing is important and frequently it is advisable to consult an authority on this subject. The essential in combustion chamber design is to provide against flame impingement upon either metallic or fire brick surfaces. Manufacturers of oil burners usually have available detailed plans for adapting their burners to various types of boilers, and such information should be utilized.

Combustion Process

Efficient combustion must produce a clean flame and must use relatively small excess of air, *i.e.*, between 25 and 50 per cent. This can be done only by vaporizing the oil quickly and completely, and mixing it vigorously with air in a combustion chamber hot enough to support the combustion. A vaporizing burner prepares the oil, for combustion, by transforming the

liquid fuel to the gaseous state through the application of heat. This is accomplished before the oil vapor mixes with air to any extent and if the air and oil vapor temperatures are high and the fire pot hot, a clear blue flame is produced. There may be a deficiency of air or an excessive supply of air, depending upon burner adjustment, without altering the clean, blue appearance of the flame.

An atomizing burner *i.e.*, gun and rotary types is so named because the oil is mechanically separated into very fine particles so that the surface exposure of the liquid to the radiant heat of the combustion chamber is vastly increased and vaporization proceeds quickly. The result of such practice is the ability to burn more and heavier oil within a given combustion space or furnace volume. Since the air enters the fire pot with the liquid fuel particles, it follows that mixing, vaporization and burning are all occurring at once in the same space. This produces a luminous instead of a blue or non-luminous flame. In this case a deficient amount of air is indicated by a dull red or dark orange flame with smoky flame tips.

An excessive supply of air may produce a brilliant white flame in some cases or, in others, a short ragged flame with incandescent sparks flashing through the combustion space. While extreme cases may be easily detected, it is generally not possible to distinguish, by the eye alone, the finer adjustment which competent installation requires.

Tests indicate that there is no difference in economy between a blue flame and a luminous flame if the position, shape and the per cent of excess air of both flames are proper for each type.

Furnace or Combustion Chamber Design

With burners requiring a refractory combustion chamber the size and shape should be in accordance with the manufacturer's instructions. It is important that the chamber shall be as nearly air tight as is possible, except when the particular burner requires a secondary supply of air for combustion.

It is evident that the atomizing burner is dependent upon the surrounding heated refractory or fire brick surfaces to vaporize the oil and support combustion. Unsatisfactory combustion may be due to inadequate atomization and mixing. A combustion chamber can only compensate for these things to a limited extent. If liquid fuel continually reaches some part of the fire brick surface, a carbon deposit will result. The combustion chamber should enclose a space having a shape similar to the flame but large enough to avoid flame contact. The nearest approach in practice is to have the bottom of the combustion chamber flat but far enough below the nozzle to avoid flame contact, the sides tapering from the air tube at the same angle as the nozzle spray and the back wall rounded. A plan view of the combustion chamber thus resembles in shape the outline of the flame. In this way as much fire brick as possible is close to the flame so it may be kept quite hot. This insures quick vaporization, rapid combustion and better mixing by eliminating dead or inactive spaces in the combustion chamber. An overhanging arch at the back of the fire pot is sometimes used to increase the flame travel and give more time for mixing and burning and sometimes to prevent the gases from going too directly into the boiler flues. When good atomization and vigorous mixing are achieved by the burner, combustion chamber

design becomes a less critical matter. Where secondary air is used, combustion chamber design is quite important. With some of the vertical rotary burners considerable care must be exercised in definitely following the manufacturer's instructions when installing the hearth as in this class successful performance depends upon this factor.

Combustion Adjustments

Where adjustments of oil and air have been made which give efficient combustion, the problem of maintaining the adjustments constant becomes an important one. Particularly is this true when the change causes the per cent of excess air to decrease below allowable limits of the burner. A decrease in air supply while the oil delivery remains constant or an increase in oil delivery while the air supply remains constant will make the mixture of oil and air too rich for clean combustion. The more efficient the adjustment (*i.e.*, 25 per cent excess air) the more critical it will be of variations. The oil and air supply rates must remain constant.

The following factors may influence the oil delivery rate: (*a*) changes in oil viscosity due to temperature change or variations in grade of oil delivered, (*b*) erosion of atomizing nozzle, (*c*) fluctuations in by-pass relief pressures and (*d*) possible variations in methods 2*b* (3) and 2*b* (4) listed in the previous classification table. Note that any change due to partial stoppage of oil delivery will increase the proportion of excess air. This will result in less heat, reduced economy and possibly a complete interruption of service but usually no soot will form.

The following factors may influence the air supply: (*a*) changes in combustion draft due to a variety of causes (*i.e.*, changes in chimney draft because of weather changes, seasonal changes, back drafts, failure or inadequacy of automatic draft regulator, use of chimney for other purposes, possible stoppage of the chimney and changes in draft resistance of boiler due to partial stoppage of the flues), and (*b*) changes in air inlet adjustments to the fan.

It is recognized that a secondary source of air due to leakage in the boiler setting is present in many installations and it is highly desirable that this leakage be reduced to a minimum. Obviously the amount of air leakage will be determined by the draft in the combustion chamber. It is important that this draft should be reduced as low as is consistent with the proper disposal of the gases of combustion. When using mechanical draft burners with average conditions, the combustion chamber draft should not be allowed to exceed 0.02-0.05 in. water. An automatic draft regulator is very helpful in maintaining such values.

Even though a fan is generally used to supply the air for combustion, in most oil burners the importance of a proper chimney should not be overlooked. The chimney should have sufficient height and size to insure that the draft will be uniform within the limits given above if maximum efficiency throughout the whole heating season is to be maintained.

Measurement of the Efficiency of Combustion

Efficient combustion being based upon a clean flame and certain proportions of oil and air employed, it is possible to determine the results by analyzing the gases formed by the combustion process. Except in the case of a non-luminous flame it is usually sufficient to analyze only for

carbon dioxide (CO_2). A showing of 10 to 12 per cent indicates the best adjustment if the flame is clean. Most of the good installations at the present time show from 8 to 10 per cent CO_2. Taking into account the potential hazard of oil or air fluctuations with low excess air (high CO_2) a setting to give 10 per cent CO_2 constitutes a reasonable standard for most oil burners.

Controls

Controls for oil burner operation, including devices for the safety and protection of a boiler or furnace, are fully described in Chapter 34.

GAS-FIRED APPLIANCES

The increased use of gas for house heating purposes has resulted in the production of such a large number of different types of gas heating systems and appliances that today there is probably a greater variety of them than there is for any other kind of fuel.

Gas-fired heating systems may be classified as follows:

- I. Gas-Designed Heating Systems.
  - A. Central Heating Plants.
    1. Steam, hot water, and vapor boilers.
    2. Warm air furnaces.
  - B. Unit Heating Systems.
    1. Warm air floor furnaces.
    2. Industrial unit heaters.
    3. Space heaters.
    4. Garage heaters.
- II. Conversion Heating Systems.
  - A. Central Heating Plants.
    1. Steam, hot water and vapor boilers.
    2. Warm air basement furnaces.

These systems are supplied with either automatic or manual control. Central heating plants, for example, whether gas designed or conversion systems, may be equipped with room temperature control, push-button control, or manual control.

Gas-Fired Boilers and Furnaces

Specially designed boilers are available for gas-firing such as shown in Fig. 16. Additional information on gas-fired boilers will be found in Chapter 12. Either snap action or throttling control is available for gas boiler operation. Throttling control is especially advantageous in steam systems because steam pressures can be maintained at desired points, while at the same time complete cut-off of gas is possible when the thermostat calls for it.

Warm air furnaces are variously constructed of cast-iron, sheet metal and combinations of the two materials. If sheet metal is used, it must be of such a character that it will have the maximum resistance to the corrosive effect of the products of combustion. With some varieties of manufactured gases, this effect is quite pronounced. Warm air furnaces

are obtainable in sizes from those sufficient to heat the largest residence down to sizes applicable to a single room. The practice of installing a number of separate furnaces to heat individual rooms is peculiar to mild climates. Small furnaces, frequently controlled by electrical valves actuated by push-buttons in the room above, are often installed to heat rooms where heat may be desired for an hour or so each day. These furnaces are used also for heating groups of rooms in larger residences. In a system of this type each furnace should supply a group of rooms in which the heating requirements for each room in the group are similar.

The same fundamental principle of design that is followed in the construction of boilers, that is, breaking the hot gas into fine streams so that

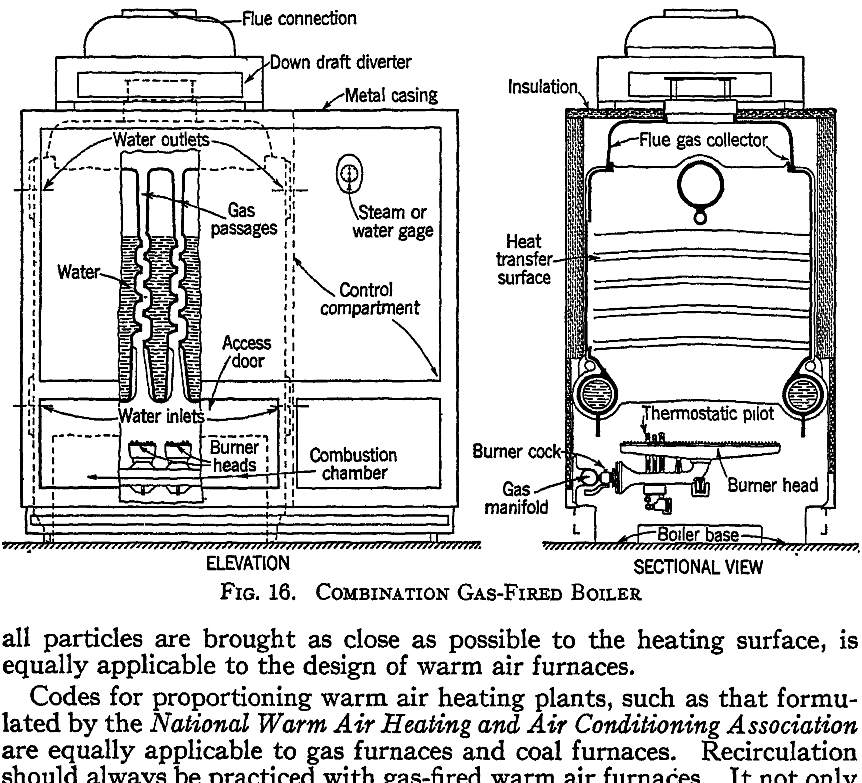


FIG. 16. COMBINATION GAS-FIRED BOILER

all particles are brought as close as possible to the heating surface, is equally applicable to the design of warm air furnaces.

Codes for proportioning warm air heating plants, such as that formulated by the *National Warm Air Heating and Air Conditioning Association* are equally applicable to gas furnaces and coal furnaces. Recirculation should always be practiced with gas-fired warm air furnaces. It not only aids in heating, but is essential to economy. Where fans are used in connection with warm air furnaces for residence heating, it is well to have the control of the fan and of the gas so coordinated that there will be sufficient delay between the turning on of the gas and the starting of the fan to prevent blasts of cold air being blown into the heated rooms. An additional thermostat in the air duct easily may be arranged to accomplish this.

Warm air floor furnaces are well adapted for heating first floors, or where heat is required in only one or two rooms. A number may be used to provide heat for the entire building where all rooms are on the ground floor, thus giving the heating system flexibility as any number of rooms

may be heated without heating the others. With the usual type the register is installed in the floor, the heating element and gas piping being suspended below.

Space Heaters

Space heaters are generally used for auxiliary heating, but may be, and are in many cases, installed for furnishing heat to entire buildings. With the exception of wall heaters, they are portable, and can be easily removed and stored during the summer season. Although it is generally desirable to connect space heaters with solid piping, use of flexible gas tubing, semi-rigid tubing, or flexible metal hose is frequently resorted to, particularly in the connection of portable types. Where flexible gas tubing is used, a gas shut-off on the heater is not permitted and only *American Gas Association* certified tubing should be used.

Parlor furnaces or *circulators* are usually of the cabinet type. They heat the room entirely by convection, *i.e.*, the cold air of the room is drawn in near the base and passes up inside the jacket around a drum or heating section, and out of the heater at or near the top. These heaters cause a continuous circulation of the air in the room during the time they are in operation. The burner or burners are located in the base at the bottom of an enclosed combustion chamber. The products of combustion pass around baffles within the heating element or drum, and out the flue at the back near the top. They are well adapted not only for residence room heating but also for stores and offices.

Radiant heaters give off considerable portion of their heat in the form of radiant energy emitted by an incandescent refractory that is heated by a Bunsen flame. They are made in numerous shapes and designs and in sizes ranging from two to fourteen or more radiants. Some have sheet-iron bodies finished in enamel or brass while others have cast-iron or brass frames with heavy fire-clay bodies. An atmospheric burner is supported near the center of the base, usually by set screws at each end. Others have a group of small atmospheric burners supported on a manifold attached to the base. Most radiant heaters are supported on legs and are portable; however, there are also types which are encased in a jacket which fits into the wall with a grilled front, similar to a wall register.

Gas-fired steam and hot water radiators are popular types of room heating appliances. They provide a form of heating apparatus for intermittently heated spaces such as stores, small churches and some types of offices and apartments. They are made in a large variety of shapes and sizes and are similar in appearance to the ordinary steam or hot water radiator. A separate combustion chamber is provided in the base of each radiator and is usually fitted with a one-piece burner. They may be secured in either the vented or unvented types, and with steam pressure, thermostatic or room temperature controls.

Warm air radiators are similar in appearance to steam or hot water radiators. They are usually constructed of pressed steel or sheet metal hollow sections. The hot products of combustion circulate through the sections and are discharged from a flue or into the room, depending upon whether the radiator is of the vented or unvented type.

Garage heaters are usually similar in construction to the cabinet

circulator space heaters, except that safety screens are provided over all openings into the combustion chamber to prevent any possibility of explosion from gasoline fumes or other gases which might be ignited by an open flame. They are usually provided with automatic room temperature controls and are well suited for heating either residence or commercial garages.

Conversion Burners

Residence heating with gas through the use of conversion burners installed in coal-designed boilers and furnaces represents a common type of gas-fired house heating system. In many conversion burners radiants or refractories are employed to convert some of the energy in the gas to radiant heat. Others are of the blast type, operating without refractories.

Many conversion units are equipped with a sheet metal secondary air duct which is inserted through the ashpit door. The duct is equipped with automatic air controls which open when the burners are operating and close when the gas supply is turned off. This prevents a large part of the circulation of cold air through the combustion space of the appliance when not in operation. With this duct the air necessary for proper combustion is supplied directly to the burner, thereby making it possible to reduce the excess air passing through the combustion chamber.

Conversion units are made in many sizes both round and rectangular to fit different types and makes of boilers and furnaces. They may be secured with manual, push-button, or room temperature control.

Combustion Process and Adjustments

Because of the varying composition of gases used for domestic heating it is difficult to generalize on the subject of gas burner combustion.

Little difficulty should be experienced in maintaining efficient combustion conditions when burning gas. The fuel supply is normally held to close limits of variation in pressure and calorific value and, therefore, the rate of heat supply is nominally constant. Since the force necessary to introduce the fuel into the combustion chamber is an inherent factor of the fuel, no draft by the chimney is required for this purpose. The use of a draft diverter insures the maintenance of constant low draft condition in the combustion chamber with a resultant stability of air supply. A draft diverter is also helpful in controlling the amount of excess air and preventing back drafts which might extinguish the flame. (See Chapter 8.)

Measurement of the Efficiency of Combustion

It is possible to determine the results of combustion by analyzing the gases of combustion with an Orsat apparatus. It is desirable to determine the percentage of carbon dioxide (CO_2), oxygen (O_2) and carbon monoxide (CO) in the flue gases. While ultimate CO_2 values of 10 to 12 per cent may be obtained from the combustion of gases commonly used for domestic heating, a combustion adjustment which will show from 8 to 10 per cent CO_2 represents a practical value. Under normal conditions no CO will be produced by a gas-fired boiler or furnace. Limitations as to output rating by the *A.G.A.* are based upon operation with not more than 0.04 per cent CO in the products of combustion. This is too small an amount to be determined by the ordinary flue gas analyzer.

Controls

Temperature controls for gas burners are described in Chapter 34. Some central heating plants are equipped with push-button or other manual control. The main gas valve may be of either the snap action or throttling type.

Sizing Gas-Fired Heating Plants

While gas-burning equipment usually is completely automatic, maintaining the temperature of rooms at a predetermined and set figure, there are in use installations which are manually controlled. Experience has shown that, in order to effectively overcome the starting load and losses in piping, a manually-controlled gas boiler should have an output as much as 100 per cent greater than the equivalent standard cast-iron column radiation which it is expected to serve.

Boilers under thermostatic control, however, are not subject to such severe pick-up or starting loads. Consequently, it is possible to use a much lower selection, or safety factor. A gas-fired boiler under thermostatic control is sensitive to variations in room temperatures so that in most cases a factor of 20 per cent is sufficient for pick-up load.

The factor to be allowed for loss of heat from piping, however, must vary somewhat, the proportionate amount of piping installed being considerably greater for small installations than for large ones. Liberal selection factors to be added to the installed steam radiation under thermostatic control are given in Fig. 1 of Chapter 12.

Appliances used for heating with gas should bear the approval seal of the *American Gas Association* Testing Laboratory. Installations should be made in accordance with the recommendations shown in the publications of that association.

Ratings for Gas Appliances

Since a gas appliance has a heat-generating capacity that can be predicted accurately to within 1 or 2 per cent, and since this capacity is not affected by such things as condition of fuel bed and soot accumulation, makers of these appliances have an opportunity to rate their product in exact terms. Consequently all makers give their product an hourly Btu output rating. This is the amount of heat that is available at the outlet of a boiler in the form of steam or hot water, or at the bonnet of the furnace in the form of warm air. The output rating is in turn based upon the Btu input rating which has been approved by the *American Gas Association* Testing Laboratory and upon an average efficiency which has been assigned by that association.

In the case of boilers, the rating can be put in terms of square feet of equivalent direct radiation by dividing it by 240 for steam, and 150 for water. This gives what is called the *American Gas Association* rating, and is the manner in which all appliances approved by the *American Gas Association* Laboratory are rated. To use these ratings it is only necessary to increase the calculated heat loss or the equivalent direct radiation load by an appropriate amount for starting and piping, and to select the boiler or furnace with the proper rating.

The rating given by the *American Gas Association* Laboratory is not only a conservative rating when considered from the standpoint of capacity and efficiency, but is also a safe rating when considered from the standpoint of physical safety to the owner or caretaker. The rating that is placed upon an appliance is limited by the amount of gas that can be burned without the production of harmful amounts of carbon monoxide. Gas boilers are available with ratings up to 14,000 sq ft of steam radiation, while furnaces with ratings up to about 500,000 Btu per hour are available.

REFERENCES

Domestic Burners for Pennsylvania Anthracite, Commercial Standard CS48-40, U. S. Department of Commerce.

Uniform Stoker Rating Code, *Stoker Manufacturers Association*, adopted November 15, 1937.

Recommended Standards Governing Minimum Setting Heights, *Stoker Manufacturers Association*, adopted March 12, 1940.

Performance Expectancy of Domestic Underfeed Stokers for Anthracite, by Allen J. Johnson (*Transactions, A.I.M.E.*, Coal Division, Vol. 119, 1936).

The Relation of the Size of Bituminous Coals to Their Performance on Small Underfeed Stokers—The Relation of the Size in the Hopper to That Burned in the Retort, by R. A. Sherman and E. R. Kaiser, Technical Report No. 1, Bituminous Coal Research, Inc., (December, 1935) Pt. I.

The Relation of the Size of Bituminous Coals to Their Performance on Small Underfeed Stokers—Burning Tests on Four Typical Coals, by R. A. Sherman, E. R. Kaiser and H. R. Limbacher, Technical Report No. 1, Bituminous Coal Research, Inc. (July, 1937) Pt. II.

Stoker Coals, Anonymous, Bulletin of Chesapeake and Ohio Railway Co., 1935.

A.S.H.V.E. RESEARCH REPORT No. 907—Study of Performance Characteristics of Oil Burners and Low Pressure Heating Boilers, by L. E. Seeley and E. J. Tavanlar (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 517).

A.S.H.V.E. RESEARCH REPORT No. 925—A Study of Intermittent Operation of Oil Burners, by L. E. Seeley and J. H. Powers (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 317).

Air Supply and Its Effect on Performance of Oil Burners and Heating Boilers, by L. E. Seeley, J. H. Powers and E. J. Tavanlar (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 75).

Study of Fuel Burning Rates and Power Requirements of Oil Burners in Relation to Excess Air, by L. E. Seeley and E. J. Tavanlar (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 319).

Oil Burning in Residences, by D. W. Nelson (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 355).

Domestic Oil Burners, by A. H. Senner (*Mechanical Engineering*, November, 1936).

A Study of Oil-Fired Heating Boilers, by R. C. Cross and W. R. Lyman (*Heating and Ventilating*, October, 1931).

Value of Oil Burner Installation Surveys, by R. C. Cross and W. R. Lyman (*Heating and Ventilating*, January, 1931).

Comfort Heating, *American Gas Association*.

Approval Requirements of Central House Heating Gas Appliances, *American Gas Association*.

A Method for Determining Fuel Burning Rates in Heating Boilers Fired by Automatic Devices, by R. C. Cross (*Heating and Ventilating*, January, 1932).

Heat Losses and Efficiencies of Fuels in Residential Heating, by R. A. Sherman and R. C. Cross (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 185).

Efficiencies and Costs of Various Fuels in Domestic Heating, by R. A. Sherman and R. C. Cross, Technical Report No. 3, Bituminous Coal Research, Inc., (December, 1936).

Automatic Heating Equipment, *American Architect Reference Data*, No. 7, September, 1933.

Chapter 11

HEAT AND FUEL UTILIZATION

Fuel Consumption Records, Calculated Heat Loss Estimation Method, Maximum Rate of Fuel Burning, Degree-Day Method, Unit Fuel Consumption per Degree-Day, Maximum Demands and Load Factors

MANY methods are in use for estimating in advance of actual operation the anticipated heat or fuel consumption of heating plants over long or short periods. With suitable modification in procedure these same general methods are frequently useful in checking the degree of effectiveness with which heat or fuel is utilized during plant operation.

In applying any of these estimating methods to the consumption of a particular building plant it should be noted that (a) reliable records of past heat or fuel consumptions of *this* building will usually produce more trustworthy estimates of future consumptions than will any data obtained by averages or from other similar buildings; (b) where no past records exist useful data can sometimes be obtained from records of similar buildings with similar plants *in the same locality;* (c) records of consumption, which are averages from many types of plants in many types of buildings in various localities, can produce no better than an average estimate which may be far from accurate; (d) estimates based on computed heat losses without the benefit of operating data are wholly dependent on how well the computation represents the actual facts

Where records of past consumptions are available they should be examined for reliability to be sure that the records show fuel or heat for the heating plant only, or else make a suitable allowance for fuel used for other purposes, such as heating service water. Weights and measures shown on invoices may not always agree with fuel used, for residues left in bins or tanks may represent a considerable fraction of the fuel charged to a building. Generally, plant operating records of fuel used are to be preferred to those obtained from accounting or bookkeeping offices from fuel invoices.

Records from similar buildings even in the same locality should be examined with care before being used as the basis of estimates. The type of heating system, the quality of supervision in manual plants, the kind of control in automatic plants, and the attention given to the plant operation are all factors in fixing the consumption in any building. Many times these factors do not show up in superficial examination and are even difficult to evaluate when known to be present. Especially check the

records to be sure that they do not include energy or fuel used for other purposes than heating the building.

Estimates based on computed heat losses alone are frequently the only ones possible to obtain, especially where new equipment is put into unusual buildings and there is a scarcity of records and an absence of experience data. Such estimates also have to be made where direct information is not obtainable as, for example, if a survey is being made without the assistance or knowledge of the building operator and thus without information as to the actual consumption. Estimates of this kind are also useful in some cases where a *relative* standard of performance is desired to serve as a base of comparisons in a campaign of fuel utilization. In such situations it can be plausibly argued that an estimate based on computed heat quantities is to be preferred to one which is related to operating methods.

In interpreting and evaluating heat or fuel consumption estimates as well as in their preparation, it is well to realize that any estimating method used will produce a more reliable result over a long period operation than over a short period. Nearly all of the methods in common use will give trustworthy results over a full *annual* heating season, and in some cases such estimates will prove consistent within themselves for monthly periods. As the period of the estimate is shortened there is more chance that some factor not allowed for in the estimating method will become controlling and thus give discrepant and even ridiculous results.

Of the various estimating methods in use attention is directed in this discussion to but two as they are illustrative of all, viz: (1) calculated heat loss method, and (2) degree-day method.

CALCULATED HEAT LOSS METHOD

This method is theoretical and assumes constant temperatures for very definite hours each day throughout the entire heating season. It does not take into account factors which are difficult to evaluate such as opening of windows, abnormal heating of the building, sun effect, poor heating systems, and many others.

In order to apply this method the hourly heat loss from the building under maximum load, or design condition, is computed following the principles discussed in Chapters 4 and 5 and the method described and illustrated in Chapter 6.

In some cases, however, depending on the presence of interior partitions, the computed heat loss is modified when used for estimating the heat or fuel consumption. If the building has no interior walls or partitions then, by the method of Chapters 5 and 6, the infiltration losses are calculated by using only half the total window crack. In such a building the calculated loss need not be modified in order to prepare heat or fuel estimates by this method. Where the building does contain interior walls or partitions instead of using as the calculated heat loss (H) which is equal to the sum of the transmission losses (H_t) and the infiltration losses (H_i), it is more desirable to let $H = H_t + \frac{H_i}{2}$.

In predicting fuel consumption for heating a building by the Calculated Heat Loss Method, the general formula is:

$$F = \frac{H\,(t - t_a)\,N}{E\,(t_d - t_o)\,C} \tag{1}$$

where

F = quantity of fuel or energy required (in the units in which C is expressed).
H = calculated heat loss, Btu per hour, during the design hour, based on t_o and t_d $\left(\text{generally } H = H_t + H_i \text{ but may on occasion equal } H_t + \frac{H_i}{2}\right)$.
t = average inside temperature maintained over estimate period, degrees Fahrenheit.
t_a = average outside temperature through estimate period, degrees Fahrenheit (Table 2, Chapter 6).
t_d = inside design temperature, degrees Fahrenheit (usually 70 F).
t_o = outside design temperature, degrees Fahrenheit (see Outside Temperatures, Chapter 6).
N = number of heating hours in estimate period (for an Oct. 1—May 1 heating season, 5088).
E = efficiency of utilization of the fuel over the period, expressed as a decimal; not the efficiency at peak or rated load condition.
C = heating value of one unit of fuel or energy.

Example 1. A residence in Philadelphia is to be heated to 70 F from 6 A.M. to 10 P.M. and 55 F from 10 P.M. to 6 A.M. The calculated hourly heat loss is 120,000 Btu per hour based on 70 F inside at −5 F outside. If the building is to be heated by metered steam, how many pounds would be required during an average heating season?

Solution. The heating value of steam may be taken as 1000 Btu per pound, and since it is purchased steam, the efficiency can be assumed as 100 per cent. From Table 2, Chapter 6, t_a = 42.7 F. The average inside temperature is:

$$\frac{(16 \times 70) + (8 \times 55)}{24} = 65 \text{ F.}$$

Substituting in Equation 1:

$$F = \frac{120{,}000\,(65 - 42.7)\,5088}{1.00\,[70 - (-5)]\,1000} = 181{,}239 \text{ lb.}$$

Example 2. How much would the fuel cost to heat the building in Example 1 during an average heating season with coal at \$8 per ton and with a calorific value of 11,000 Btu per pound, assuming that the seasonal efficiency of the plant was 55 per cent?

Solution. Substituting in Equation 1: $F = \dfrac{120{,}000\,(65 - 42.7)\,5088}{0.55\,[70 - (-5)]\,11{,}000} = 30{,}013$ lb = 15 tons, which, at \$8 per ton, costs \$120.

Example 3. What will be the estimated fuel cost per year of heating a building with gas, assuming that the calculated hourly heat loss is 92,000 Btu based on 0 F, which includes 26,000 Btu for infiltration? The design temperatures are 0 F and 72 F. The normal heating season is 210 days, and the average outside temperature during the heating season is 36.4 F. The seasonal efficiency will be 75 per cent. The heating plant will be thermostatically controlled, and a temperature of 55 F will be maintained from 11 P.M. to 7 A.M. Assume that the price of gas is 7 cents per 100,000 Btu of fuel consumption, and disregard the loss of heat through open windows and doors.

Solution. The average hourly temperature is:

$$t_a = \frac{(72 \times 16) + (55 \times 8)}{24} = 66.3 \text{ F.}$$

The maximum hourly heat loss will be:

$$H = 92{,}000 - \frac{26{,}000}{2} = 79{,}000 \text{ Btu.}$$

$$M = \frac{79{,}000\ (66.3 - 36.4) \times 24 \times 210}{100{,}000 \times 0.75 \times (72 - 0)} = 2204.6 \text{ hundred thousand Btu.}$$

$2204.6 \times \$0.07 = \$154.34$ = estimated fuel cost per year of heating building.

Equation 1 can be expressed as:

$$F = \frac{H}{EC} \times \frac{(t - t_a)\ N}{(t_d - t_o)} \tag{2}$$

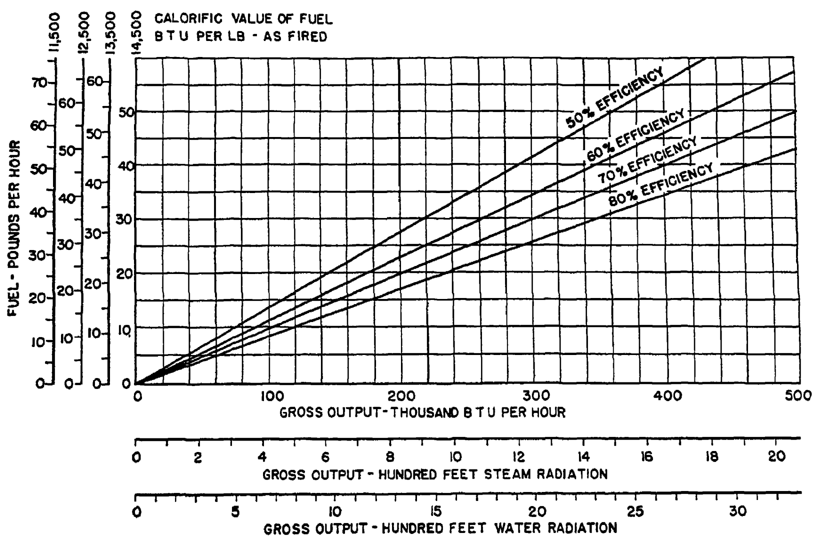


FIG. 1. COAL FUEL BURNING RATE CHART

where the expression $\frac{H}{EC}$ is the rate at which fuel is burned during the design hour. Values of this rate are plotted as ordinates in Figs. 1, 2 and 3 for coal, oil and gas. For a given efficiency, the rate of fuel burning is directly proportional to the load and therefore these charts can be extended by moving the decimal points the same number of digits in both vertical and horizontal scales. Use of these charts thus expedites the estimate.

The charts are plotted so that the load is expressed in three terms: (a) hourly heat loss at design conditions, (b) square feet of steam radiator surface (240 Btu per hour), and (c) square feet of hot water radiator surface (150 Btu per hour). By entering the chart at the correct point on the abscissa corresponding to the calculated heat loss (H), following vertically to the seasonal efficiency assumed and thence horizontally to the fuel rate, the rate of fuel burning during a maximum or design hour will be found along the left hand scale for various calorific values of the fuel.

In the case of gravity warm air heating installations, the load is usually expressed in square inches of leader pipe. This can be converted into hourly heat loss by multiplying by the factors in Table 1.

Example 4. A building located in Salt Lake City with an oil-burning heating plant has a calculated hourly heat loss of 260,000 Btu per hour. The plant is designed to maintain a temperature of 70 F inside during all 24 hours of the day, the outside design temperature is -5 F, the average outside temperature 40 F, the heating season 5088 hours long, the assumed efficiency 60 per cent, and the oil has a calorific value of 143,400 Btu per gallon. What will be the seasonal fuel consumption?

Solution. Enter Fig. 2 at 260,000 on the upper horizontal scale, move vertically to the 60 per cent efficiency curve, and horizontally to the vertical scale where the firing rate $\left(\frac{H}{EC}\right)$ is found as 3.0 gal per hour.

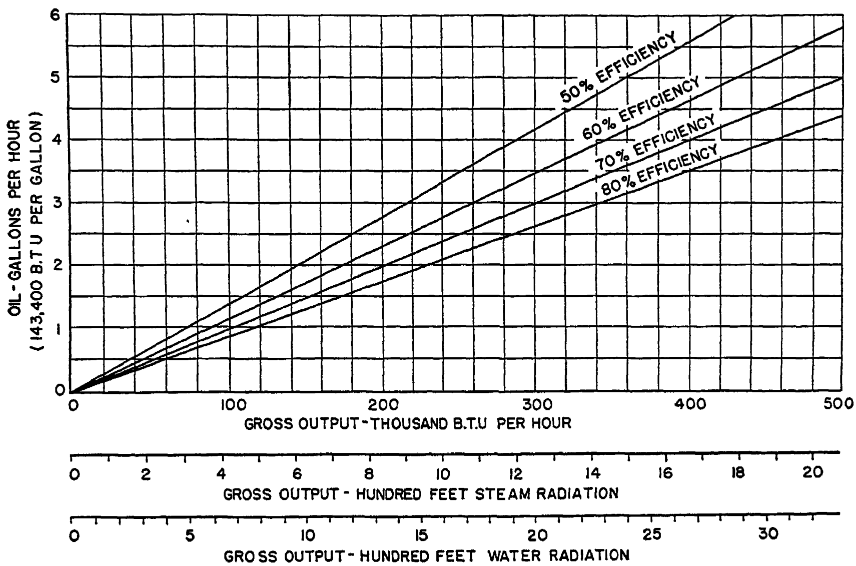


FIG. 2. OIL FUEL BURNING RATE CHART[a]

[a]This chart is based upon No. 3 oil having a heat content of 143,400 Btu per gallon. If other grades of oil are used multiply the value obtained from this chart by the following factors: No. 1 oil (139,000 Btu per gallon) 1.032; No. 2 oil (141,000 Btu per gallon) 1.017; No. 4 oil (144,500 Btu per gallon) 0.992; No. 5 oil (146,000 Btu per gallon) 0.982; and No. 6 oil (150,000 Btu per gallon) 0.956.

Substituting this in Equation 2 for $\left(\frac{H}{EC}\right)$:

$$F = 3.0 \times \frac{(70 - 40)\ 5088}{70 - (-5)} = 6106 \text{ gal.}$$

Example 5. What would be the total gas consumption over a full heating season of a gas-fired gravity warm air furnace designed according to the Code[1], and with four 12 in. and two 8 in. round leaders to the first floor and six 10 in. leaders to the second floor, if the gas has a heating value of 500 Btu per cubic foot, the plant operates at a 70 per cent

[1]Standard Code Regulating the Installation of Gravity Warm Air Heating Systems in Residences (9th edition), and the Technical Code for the Design and Installation of Mechanical Warm Air Heating Systems, may be obtained from the *National Warm Air Heating and Air Conditioning Association*, 145 Public Square, Cleveland, Ohio.

seasonal efficiency and is designed to maintain an average inside temperature of 65 F when it is 10 F outside in a city where the average outside temperature is 45 F and the heating season is 5088 hours long?

Solution. The area of the round leaders is: 12 in., 113 sq in.; 10 in., 79 sq in.; and 8 in., 50 sq in. From Table 1 the total Btu transmitted is:

First Floor:	$[(4 \times 113) + (2 \times 50)] \times 111$	= 61,272 Btu per hour.
Second Floor:	$(6 \times 79) \times 167$	= 79,158 Btu per hour.
	Total	140,430 Btu per hour.

Allowing 10 per cent for duct and furnace losses, the gross output would be 154,500 Btu per hour.

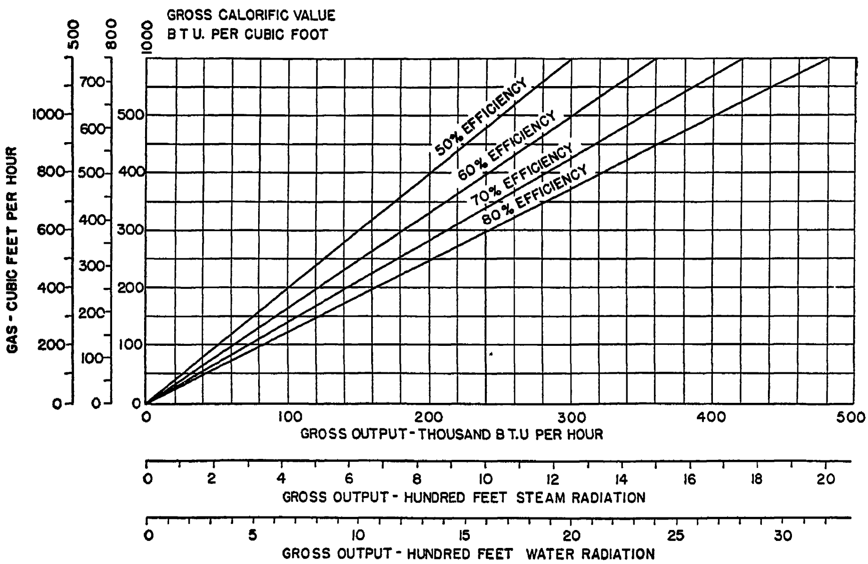


FIG. 3 GAS FUEL BURNING RATE CHART

Enter Fig. 3 at 154.5 on the upper horizontal scale, move to the 70 per cent efficiency curve and thence to the 500 Btu per cubic foot vertical scale and find $\left(\frac{H}{EC}\right)$ to be approximately 440 cu ft per hour.

Substituting in Equation 2:

$$F = 440 \times \frac{(65 - 45)\ 5088}{(70 - 10)} = 746{,}428 \text{ cu ft.}$$

Maximum Rate of Fuel Burning

The rate at which fuel is burned during the maximum, or design hour, is frequently useful in setting, or adjusting, the fuel feed devices attached to stokers, oil-burners, and gas burners. This rate is $\left(\frac{H}{EC}\right)$ and can be found from the charts of Figs. 1, 2 and 3 in the same way as outlined in the Examples 4 and 5. In using the charts for this purpose, however, it should be noted that the efficiency (E) is the overall efficiency of the boiler

or furnace at the time of peak load. This efficiency is generally considerably greater than the value selected for E when the seasonal efficiency of utilization is used in making seasonal fuel estimates. Failure to distinguish between the two essentially different meanings attached to E may result in grossly inaccurate estimates.

The correct fuel burning rate can be determined directly from the several charts for oil or gas burning installations, as these customarily operate on a strictly intermittent basis. These fuel burning devices usually introduce the fuel at a single fixed rate during the *on* periods and this rate should be sufficient to carry the gross or maximum design load. In the case of coal stokers, which are usually capable of variable rates of firing, it is desirable to operate at as low a rate as weather conditions will permit, but the maximum firing rate of the stoker should be sufficient to carry the gross load. This rate may be determined by the same method as used for oil or gas.

TABLE 1. HEAT CARRYING CAPACITY OF GRAVITY WARM AIR FURNACE ROUND LEADER PIPES

180 F Register Temperature

LEADER PIPE	BTU PER HOUR AT DESIGN CONDITIONS PER SQ IN. OF LEADER PIPE
First floor	111
Second floor	167
Third floor	200

Example 6. The estimated net load (including domestic hot water supply) as calculated for a residence is 1500 sq ft of hot water radiation. Determine the firing rates for various mechanically fired fuels assuming an overall boiler efficiency of 70 per cent; using coal with a calorific value of 12,500 Btu per pound; No. 3 fuel oil and natural gas having a gross heating value of 1000 Btu per cubic foot.

Solution. Referring to Fig. 1, Chapter 12, a piping and pick-up factor for a net load of 1500 sq ft is found to be 43 per cent or the gross output is equivalent to 1500 $\times$ 1.43 = 2145 sq ft of hot water radiation.

Using the charts in Figs. 1, 2 and 3 project vertically from the gross output value on the proper horizontal scale to the intersection of the 70 per cent efficiency line. From the intersection of this line proceed horizontally to the proper vertical scale where a direct value of the required fuel burning rate is given. These values are rates of burning while firing device is in operation and are not indicative of hourly fuel consumption.

By use of the respective charts the firing rates for the various fuels will be found to be: coal 36.8 lb per hour, oil 3.2 gal per hour, and gas 460 cu ft per hour.

DEGREE-DAY METHOD

This method is based on consumption data which have been taken from buildings in operation, and the results computed on a degree-day basis. While this method may not be as theoretically correct as the Calculated Heat Loss Method, it is of more value for practical use.

The amount of heat required by a building depends upon the outdoor temperature, if other variables are eliminated. Theoretically it is proportional to the difference between the outdoor and indoor temperatures. Some years ago the *American Gas Association*[2] determined from experi-

[2]See Industrial Gas Series, House Heating, (*third edition*) published by the *American Gas Association*

ment in the heating of residences that the gas consumption varied directly as the difference between 65 F and the outside temperature. In other words, on a day when the temperature was 20 deg below 65 F, twice as much gas was consumed as on a day when the temperature was 10 deg below 65 F. The degree-day is defined in Chapter 47.

Some years ago the *National District Heating Association* studied the metered steam consumption of 163 buildings[3] in 22 different cities and published data substantiating the fact that the 65 F base originally chosen by the gas industry is approximately correct. (See Table 2.)

If the degree-days occurring each day are totaled for a reasonably long period, the fuel consumption during that period as compared with another period will be in direct proportion to the number of degree-days in the two periods. Consequently, for a given installation, the fuel consumption can be calculated in terms of fuel used per degree-day for any sufficiently long period and compared with similar ratios for other periods to determine the relative operating efficiencies with the outside temperature variable eliminated.

Predictions of fuel consumption are generally based on the average number of degree-days which have occurred over a long period of years, and such averages, by months, on a 65 F base, are given for various United States and Canadian cities in Table 3. In general, attempts to apply the degree-day method to fuel consumptions over a period of less than a month are of questionable value.

Formula for Degree-Day Method

The general formula for predicting fuel consumption by the Degree-Day Method is:

$$F = U \times N \times D \qquad (3)$$

where

F = fuel consumption for the estimate period.

U = unit fuel consumption, or quantity of fuel used per degree-day per building load unit.

N = number of building load units.

D = number of degree-days for the estimate period.

Values of D for use in Equation 3 are given in Table 3. Values of N depend on the particular building for which the estimate is being prepared and must be found by surveying plans, by observation, or by measurement of the building. Values of U for use in this equation are the Unit Fuel Consumptions per Degree-Day and are obtained as a result of the collection of operating information. Certain of this information is presented later but before referring to these data attention is directed to the nature of the unit.

Unit Fuel Consumptions per Degree-Day

The quantity of fuel used per degree-day in a given heating plant can be reduced to a unit basis in terms of quantity of fuel (or steam) per degree-day per square foot of radiation, per cubic foot of heated building space, or per thousand Btu hourly heat loss at design conditions. A less frequently used basis is quantity of fuel per degree-day per square foot of

[3]These buildings are all served with steam from a district heating company.

floor area. In fact any convenient unit can be used to relate the consumption to the degree-day and to the building.

The choice of these units requires explanation and some discrimination and judgment. The use of heated space in preference to the gross building cubage used by architects is obviously more accurate for this purpose. The architects' cubage includes the outer walls and certain percentages of attic and basement space which are usually unheated. The net heated space is usually about 80 per cent of the gross cubage and can be calculated from the latter if it cannot be measured. The cubical content is somewhat inaccurate as a basis of comparison due to differences in types of construction, exposure, and ratio of exposed area to cubical contents. Use of equivalent radiator surface figures is fundamentally the same as

TABLE 2. BASE TEMPERATURE FOR THE DEGREE-DAY[a]

Type of Building	No. of Buildings Analyzed	Temperature F Corresponds to Zero Steam Consumption
Office	60	66.2
Office and Bank	4	65.8
Bank	3	66.2
Office and Telephone Exchange	2	65.5
Office and Stores	6	67.4
Stores	11	64.0
Department Stores	12	64.3
Hotels	7	66.5
Apartments	14	68.8
Residences	8	66.9
Clubs	4	65.5
Lodges	5	64.9
Theaters	3	67.6
Churches	2	65.8
Garages	2	64.8
Auto Sales and Service	4	61.2
Newspaper and Printing	3	67.7
Warehouse and Loft	3	67.7
Office and Loft	2	65.2
Manufacturing	8	65.4
Average for 163 Buildings		66.0 F

[a]Report of Commercial Relations Committee, *Proceedings, National District Heating Association*, 1932.

using a calculated heat loss and therefore units in terms of fuel per degree-day per equivalent square foot of calculated radiator surface, per 1000 Btu of calculated heat loss, or per Btu of heat loss at design conditions are all of equal accuracy and desirability. It is doubtful if installed radiator surface as determined by count should be used at all. Radiator units are also of questionable value where there is fan coil surface or warm air systems. In view of all these considerations it is believed that the unit based on *thousands of Btu of hourly calculated heat loss for the design hour* is probably the most desirable although the one most widely used seems to be *units of fuel (or heat) per degree-day per square foot of equivalent direct radiator surface.*

Since this unit is the one most widely used at present the unit fuel consumptions given in succeeding paragraphs of this chapter make use of this unit to a considerable extent, although it should be understood that most of these units of consumption can be transposed as desired.

TABLE 3. DEGREE-DAYS FOR CITIES IN THE UNITED STATES AND CANADA[a]

State	City	Jan.	Feb.	Mar.	Apr.	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec.	Total
Ala.	Birmingham	617	476	298	58						51	333	577	2410
	Mobile	418	288	164	11						3	192	397	1473
Ariz.	Phoenix	428	277	133	4							160	403	1405
Ark.	Fort Smith	790	622	384	97						89	420	710	3112
	Little Rock	732	563	372	92						72	387	645	2863
Calif.	Los Angeles	322	266	232	168	87	3				11	123	260	1472
	San Francisco	468	358	335	300	254	195	202	183	123	140	261	425	3244
Colo.	Denver	1091	904	797	537	273	18			76	428	756	1014	5894
	Grand Junction	1271	899	663	378	123				31	378	771	1162	5676
Conn.	New Haven	1141	1008	905	534	220	10			55	347	690	1008	5918
D. C.	Washington	980	832	694	351	61				3	236	594	880	4631
Fla.	Jacksonville	298	196	76								88	270	928
Ga.	Atlanta	694	552	403	120						80	387	629	2865
	Savannah	422	308	186	15						7	195	391	1524
Idaho	Boise	1091	846	691	438	245	30			102	431	720	1020	5614
Ill.	Chicago	1237	1053	890	519	202	3			17	307	714	1085	6027
	Springfield	1194	994	769	369	71				7	285	684	1032	5405
Ind.	Evansville	976	804	592	249	16					174	552	865	4228
	Indianapolis	1135	949	775	387	77				12	288	681	1017	5321
Iowa	Des Moines	1392	1156	902	447	117				28	360	798	1209	6409
	Sioux City	1463	1232	1001	516	142				68	437	894	1299	7052
Kan.	Dodge City	1116	890	688	342	65				3	276	672	1004	5056
	Topeka	1159	952	694	309	49				3	248	666	1023	5103
Ky.	Lexington	995	829	660	321	46				2	236	606	905	4600
	Louisville	949	778	608	258	16					178	549	849	4185
La.	New Orleans	335	216	71								104	291	1017
	Shreveport	558	395	208	16						24	270	493	1964
Me.	Eastport	1383	1218	1119	780	536	297	143	133	276	542	849	1200	8476
	Portland	1321	1154	1029	660	363	79		5	162	468	810	1159	7210
Md.	Baltimore	967	829	704	342	47				2	211	561	862	4525
Mass.	Boston	1150	1014	911	558	245	13			61	353	690	1008	6003
Mich.	Detroit	1259	1112	980	564	217	4			58	388	771	1107	6460
	Marquette	1510	1364	1246	816	496	183	12	37	225	567	951	1314	8721
Minn.	Duluth	1770	1501	1280	840	549	234	32	74	297	648	1050	1522	9797
	Minneapolis	1621	1375	1097	558	226	9			113	499	978	1407	7883
Miss.	Vicksburg	521	370	202	19						22	252	465	1851
Mo.	Kansas City	1141	946	691	306	43				2	226	639	1008	5002
	St. Louis	1051	846	648	267	15					191	588	933	4539
	Springfield	976	834	614	270	41				2	211	579	893	4420
Mont.	Havre	1615	1439	1175	639	360	94		22	258	636	1014	1383	8635
Neb.	Lincoln	1308	1089	852	405	107				21	335	777	1159	6053
	Omaha	1336	1106	868	414	91				15	332	795	1197	6154
Nev.	Winnemucca	1128	882	775	549	344	76			175	518	798	1085	6330
N. H.	Concord	1345	1182	1060	648	332	68		4	171	474	819	1184	7287
N. J.	Altantic City	1008	879	818	516	214	8			10	251	582	887	5173
N. M.	Santa Fe	1122	893	784	549	288	30			122	453	783	1063	6087
N. Y.	Albany	1299	1145	1001	546	177	1			69	400	771	1132	6541
	Buffalo	1252	1140	1051	666	322	41			81	406	768	1091	6818
	New York	1057	944	846	468	139				12	270	624	930	5290
N. C.	Raleigh	741	610	459	168	1					98	420	682	3179
	Wilmington	574	479	363	94						31	270	493	2304
N. D.	Bismark	1773	1532	1265	687	326	55		5	207	623	1095	1559	9127
Ohio	Cincinnati	1076	902	747	378	71				10	288	675	980	5127
	Cleveland	1194	1053	942	564	220	6			47	353	723	1048	6150
	Columbus	1128	960	803	414	93				15	304	693	1011	5421
Okla.	Oklahoma City	887	711	465	156	3					120	486	797	3625
Ore.	Baker	1243	1008	849	594	412	192	12	27	270	570	870	1169	7216
	Portland	794	641	561	396	251	80			100	335	546	738	4442
Pa.	Philadelphia	1004	871	750	387	77				3	223	579	890	4784

TABLE 3. DEGREE-DAYS FOR CITIES IN THE UNITED STATES AND CANADA[a] (CONCLUDED)

State	City	Jan.	Feb.	Mar.	Apr.	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec.	Total
Pa.	Pittsburgh	1063	916	787	414	91	......	......	......	15	288	654	955	5183
S. C.	Charleston	468	353	236	37	......	......	......	......	......	8	207	412	1721
	Columbia	589	470	304	63	......	......	......	......	......	54	330	552	2362
S. D.	Huron	1665	1420	1119	597	267	18	......	......	112	536	1005	1435	8174
	Rapid City	1333	1165	1004	621	341	49	......	......	140	512	873	1181	7219
Tenn.	Knoxville	812	647	505	210	8	......	......	......	......	160	513	766	3621
	Memphis	747	580	394	98	......	......	......	......	......	76	399	663	2957
	Nashville	818	655	490	180	3	......	......	......	......	130	480	744	3500
Texas	El Paso	620	448	285	59	......	......	......	......	......	72	369	623	2476
	Fort Worth	608	468	226	25	......	......	......	......	......	24	285	542	2178
	Houston	381	255	56	......	......	......	......	......	......	......	122	329	1143
	San Antonio	394	269	70	......	......	......	......	......	......	......	138	350	1221
Utah	Modena	1187	952	831	570	356	62	......	......	150	527	858	1114	6637
	Salt Lake City	1110	874	722	462	236	12	......	......	54	388	717	1026	5601
Vt.	Burlington	1432	1277	1113	651	264	21	......	......	140	490	861	1259	7508
Va.	Lynchburg	852	692	549	231	9	......	......	......	1	202	534	790	3860
	Norfolk	756	624	521	246	19	......	......	......	......	89	408	679	3342
	Richmond	840	711	552	252	15	......	......	......	......	167	501	781	3819
Wash.	Seattle	790	669	623	468	326	180	59	59	207	422	582	722	5107
	Spokane	1162	944	784	498	294	72	......	......	174	518	795	1071	6312
W. Va.	Elkins	1073	935	775	486	180	3	......	......	71	394	741	1001	5659
	Parkersburg	1008	862	688	348	55	......	......	......	10	276	636	924	4807
Wis.	Green Bay	1528	1333	1128	654	313	35	......	......	139	512	930	1324	7896
	LaCrosse	1516	1282	1038	534	177	1	......	......	87	456	894	1324	7309
	Milwaukee	1376	1182	1020	636	338	52	......	......	80	431	831	1206	7152
Wyo.	Cheyenne	1224	1056	989	723	456	138	......	13	240	626	906	1132	7503
	Lander	1448	1190	1011	678	428	135	......	18	279	666	1041	1383	8277

Province	City	Jan.	Feb.	Mar.	Apr.	May	June	July	Aug.	Sept	Oct.	Nov.	Dec.	Total
Alta.	Calgary	1674	1428	1240	750	496	270	124	186	450	744	1170	1395	9,927
	Edmonton	1829	1512	1302	720	434	270	124	186	450	713	1230	1519	10,289
B. C.	Vancouver	899	756	713	510	341	180	62	31	270	496	660	837	5,555
Man.	Winnipeg	2139	1820	1581	810	465	90	......	62	270	744	1320	1829	11,130
N. B.	Moncton	1519	1428	1178	810	465	210	......	93	300	620	930	1333	8,886
N. S.	Halifax	1302	1176	1085	780	496	210	......	......	210	496	780	1147	7,682
Ont.	Ottawa	1674	1484	1271	690	279	30	......	......	210	589	990	1457	8,676
	Port Arthur	1829	1624	1426	900	558	240	62	86	360	713	1140	1550	10,588
	Toronto	1333	1204	1209	720	372	60	......	......	180	558	870	1209	7,715
P.E.I.	Charlottetown	1178	1120	1209	870	529	210	......	......	600	558	870	1240	8,382
Que.	Montreal	1581	1428	1209	720	310	......	......	......	180	558	960	1395	8,341
	Quebec	1705	1484	1333	870	434	120	......	31	270	651	1050	1519	9,467
Sask.	Saskatoon	2108	1820	1581	810	465	210	62	155	450	806	1290	1736	11,493

[a] Figures for United States cities taken from Degree-Day Normals over the United States, by A. G. Topil (*Monthly Weather Review*, U. S. Weather Bureau, July, 1937, p. 266). Figures for Canadian Cities abstracted from *Heating & Ventilating*, October, 1939.

Estimating Gas Consumption

Values of the Unit Fuel Consumption Constant (U) for gas are given in Table 4 for various gas heating values, and different types and sizes of heating plants. *They are based on an inside design temperature of 70 F and an outside design temperature of 0 F and apply only to these conditions.* For other design conditions corrections must be made as given in Table 5. Estimates for industrial buildings where low inside temperatures are maintained cannot be made from this table.

The factors in Table 4, as corrected if necessary, are satisfactory for regions having 3500 to 6500 degree-days per heating season. In regions with less than 3500 degree-days the unit gas consumption is higher than given; where over 6500, the unit is less than given. Ten per cent addition or deduction in these cases is recommended by *AGA* publications.

For gas heating values other than those given in Table 4, simply interpolate or extrapolate. It will also be noted that Table 4 applies only to small installations. In general the larger the installation the smaller the unit gas consumption becomes and the values in the table should be used with care, if at all, in large gas-burning installations.

Example 7. Make an estimate of the gas required to heat a building located in Chicago, Ill., assuming that the calculated heating surface requirements are 1000 sq ft of hot water radiation based on design temperature of 0 F and 70 F. Chicago has 800 Btu mixed gas, and 6027 degree-days.

Solution. Using Equation 3 and Table 4, the fuel consumption for a design temperature of 0 F with 800 Btu gas is found to be 0.085 cu ft of gas per degree-day per square foot of hot water radiation.

$$0.085 \times 1000 \times 6027 = 512{,}295 \text{ cu ft.}$$

Estimating Oil Consumption

Unit fuel consumption factors for oil, similar to those for gas in Table 4 are given in Table 6. *The factors in Table 6 apply only to an inside design*

TABLE 4. FACTORS FOR ESTIMATING GAS CONSUMPTION[a]

Btu Value of Gas per Cu Ft	Hot Water			Steam			Warm Air	
	Cu Ft Gas per Degree-Day per Sq Ft Radiator			Cu Ft Gas per Degree-Day per Sq Ft Radiator			Cu Ft Gas per Degree-Day per 1000 Btu Hourly Design Heat Loss	
	Up to 500 Sq Ft	500 to 1200 Sq Ft	Over 1200 Sq Ft	Up to 500 Sq Ft	300 to 700 Sq Ft	Over 700 Sq Ft	Gravity	Fan Systems
500	0.142	0.135	0.128	0.242	0.231	0.220	0.855	0.820
535	0.132	0.126	0.120	0.226	0.215	0.206	0.800	0.766
800	0.089	0.085	0.081	0.151	0.144	0.137	0.534	0.513
1000	0.071	0.068	0.065	0.121	0.115	0.110	0.428	0.410
1 Therm = 100,000 Btu	Gas Consumption in Therms per Degree-Day							
	0.000708	0.000675	0.000642	0.00121	0.00115	0.00110	0.00428	0.00409

[a]Abstracted from Comfort Heating, *American Gas Association*, 1938.

TABLE 5. CORRECTION FACTORS FOR OUTSIDE DESIGN TEMPERATURES

Outside Design Temp. Deg F	Inside Design Temp. Deg F	Multiply Values in Tables 4, 6 and 7 by
−20	70	0.778
−10	70	0.875
0	70	0.000
+10	70	1.167
+20	70	1.400

temperature of 70 F and an outside design temperature of 0 F. For other outside design temperatures, the constants in Table 6 must be multiplied by the values in Table 5 as explained under Estimating Gas Consumption.

Table 6 assumes the use of oil with a heating value of 140,000 Btu per gallon. For other heating values, multiply the values in Table 6 by the ratio of 140,000 divided by the heating value per gallon of fuel being used.

Example 8. What would be the estimated seasonal oil consumption of a boiler-burner unit in Minneapolis of a building having a calculated heat loss of 192,000 Btu per hour, burning 144,000 Btu per gallon oil and operating at a seasonal efficiency of 60 per cent, if the outside design temperature for Minneapolis is −20 F, and the inside design temperature is 70 F?

Solution. From Table 6, under 60 per cent efficiency and opposite the bottom column, find the uncorrected U to be 0.00476 gal per 1000 Btu hourly heat loss.

From Table 5, the correction multiplier for −20 F outside design temperature is 0.778. Solving, $0.778 \times 0.00476 = 0.00370$. Making a further correction for the heating value:

$0.0037 \times \dfrac{140{,}000}{144{,}000} = 0.0036$ gal per 1000 Btu per hour calculated heat loss per degree-day.

From Table 3, the average degree-days for Minneapolis number 7883, and from the problem $N = 192$. Substituting in Equation 3:

$$F = 0.0036 \times 7883 \times 192 = 5449 \text{ gal.}$$

TABLE 6. UNIT FUEL CONSUMPTION[a] CONSTANTS FOR OIL[b]

Unit	Efficiency in Per Cent				
	40	50	60	70	80
Gal Oil per Sq Ft Steam Radiator	0.00172	0.00137	0.00114	0.00098	0.00086
Gal Oil per Sq Ft Hot Water Radiator	0.00108	0.00086	0.00072	0.00062	0.00054
Gal Oil per 1000 Btu per Hour Heat Loss	0.00715	0.00571	0.00476	0.00409	0.00358

[a]Based on a heating value of 140,000 Btu per gallon.
[b]Abstracted by permission from *Degree-Day Handbook* (Second Edition, 1937), by C. Strock and C. H. B. Hotchkiss.

TABLE 7. UNIT FUEL CONSUMPTION[a] CONSTANTS FOR COAL[b]

Unit	Efficiency in Per Cent				
	40	50	60	70	80
Lb Coal per Sq Ft Steam Radiator	0.0200	0.0160	0.0133	0.0114	0.0100
Lb Coal per Sq Ft Hot Water Radiator	0.0125	0.0100	0.0084	0.0072	0.0063
Lb Coal per 1000 Btu per Hour Heat Loss	0.0825	0.0666	0.0550	0.0471	0.0412

[a]Based on a heating value of 12,000 Btu per pound.
[b]Abstracted by permission from *Degree-Day Handbook*, (Second Edition, 1937), by C. Strock and C. H. B. Hotchkiss.

Estimating Coal or Coke Consumption

Coal or coke consumption estimates can be made in exactly the same way as for oil. The uncorrected values of U are given in Table 7. *These constants apply only to inside design temperatures of 70 F and an outside design temperature of 0 F*, and correction must be made for other conditions by use of the multiplying factors in Table 5. Table 7 is based on 12,000 Btu per pound coal and for other heating values of coal, values in Table 7 must be multiplied by the ratio of 12,000 divided by the heating value of fuel used.

Example 9. A building in Marquette, Mich., has an hourly heat loss at design conditions of 240,000 Btu per hour. If the inside design temperature is to be 70 F and the outside design temperature is −10 F, what will be the estimated normal seasonal coal consumption for heating if 12,000 Btu per pound fuel is burned at a 50 per cent seasonal efficiency, and what part of the total will be used during November, December, and January?

Solution. From Table 7, U uncorrected, is 0.0666 lb of coal per 1000 Btu per hour heat loss. Correcting for the outside design temperature, Table 5, the corrected value of U is $0.875 \times 0.0666 = 0.0583$. From Table 3, D is 8721 and from the problem N is 240.

Substituting in Equation 3:

$$F = 0.0583 \times 240 \times 8721 = 122{,}024 \text{ lb.}$$

Fuel used over any period is, according to the theory of the degree-day, proportional to the number of degree-days during the period. From Table 3, the average number of degree-days for November, December, and January in Marquette are 951, 1314, and 1510, a total of 3775. The yearly total is 8721, so that during these three months the estimated consumption is:

$$\frac{3775}{8721} \times 122{,}024 = 52{,}820 \text{ lb.}$$

Estimating Steam Consumption

In estimating steam consumption the efficiency is not ordinarily a factor and is assumed at 100 per cent. Ordinarily low pressure steam with a heating value of 1000 Btu per pound is used so that no correction is necessary for heating value in the usual case. In comparing values from different cities, correction should be made for design temperature (see Table 5) when the unit figures are in terms of square foot of radiator or 1000 Btu per hour calculated heat loss, but not when the values are in terms of building volume or floor space.

Consideration has been given to the difference in steam utilization of different types of buildings and Table 8[4] shows actual average units for these various types. These figures are obtained from operating results in 196 buildings located in 21 different cities in the United States. Being averages, and for small groups in each type, the figures may need considerable modification to allow for local variations. It should be especially noted that the steam used for heating hot water is included in the values given in Table 8, but in the case of office buildings, the steam for heating only is also shown. Presentation of the unit consumption in three ways permits making the estimate if either the calculated heat loss or the volume of net heated space in the building is known.

[4]The Heat Requirements of Buildings, by J. H. Walker and G. H. Tuttle (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 171).

TABLE 8. STEAM CONSUMPTION FOR VARIOUS CLASSES OF BUILDINGS[a]
(*Heating Season Only*)

Building Classification	No. of Buildings Listed	Steam Consumption Pounds per Degree-Day—65 F Basis[d]		
		Per M Cu Ft of Heated Space	Per M Sq Ft of Radiator[c] Surface	Per M Btu per Hour of Heat Loss[b]
Apartments	16	1.78	97.5	0.359
Hotels	10	1.46	80.6	0.371
Residences	12	1.32	64.2	....
Printing	7	1.25	105.5	....
Clubs and Lodges	10	0.96	77.0	....
Retail Stores	18	0.90	80.6	0.268
Theaters	6	0.90	75.0	0.498
Loft and Mfg.	16	0.89	72.3	0.283
Banks	7	0.88	45.2	....
Auto Sales and Service	8	0.83	62.2	....
Churches	6	0.58	49.4	....
Department Stores	14	0.57	60.7	0.238
Garages (Storage)[e]	6	0.42	72.3	....
Offices (Total)	35	1.09	70.0	0.283
Offices (Heating only)	35	0.975	65.4	0.256

[a]Includes steam for heating domestic water for heating season only.
[b]Heat loss calculated for maximum design condition (in most cases 70 F inside, zero outside).
[c]Equivalent steam radiator surface.
[d]The figures are a numerical—not a weighted—average for the several buildings in each class.
[e]Based on zero consumption at 55 F.

Example 10. A store in Detroit with a heating system designed to maintain 70 F inside in 0 F weather has 1700 sq ft of equivalent direct steam radiator surface, and uses only moderate quantities of hot water. What would be the estimated average yearly steam consumption of purchased steam for heating and for hot water during the heating season?

Solution. According to Table 8, a store (0 to 70 F conditions) would use 80.6 lb of steam per thousand square feet of radiation per degree-day, including winter hot water. From Table 3, Detroit has 6460 degree-days per normal year. Inserting in Equation 3:

$$F = 80.6 \times 1.7 \times 6460 = 885{,}149 \text{ lb of steam.}$$

TABLE 9. BUILDING LOAD FACTORS AND DEMANDS OF SOME DETROIT BUILDINGS[a]

Building Classification	Load Factor	Lb of Demand per Hour per Sq Hour of Equivalent Installed Radiator Surface
Clubs and Lodges	0.318	0.184
Hotels	0.316	0.207
Printing	0.287	0.217
Offices	0.263	0.209
Apartments	0.255	0.225
Retail Stores	0.238	0.182
Auto Sales and Service	0.223	0.248
Banks	0.203	0.158
Churches	0.158	0.152
Department Stores	0.138	0.145
Theaters	0.126	0.151

[a]Report of Commercial Relations Committee, *Proceedings, National District Heating Association*, 1932.

MAXIMUM DEMANDS AND LOAD FACTORS

In one form of district heating rates, a portion of the charge is based upon the maximum demand of the building. The maximum demand may be measured in several different ways. It may be taken as the instantaneous peak or as the rate of use during any specified interval. One method is to take the average of the three highest hours during the winter. These figures are available for a number of buildings in Detroit, as shown in Table 9.

These maximum demands were measured by an attachment on the condensation meter and therefore represent the amounts of condensation passed through the meter in the highest hours, rather than the true rate at which steam is supplied. There might be slight differences in these two quantities due to time lag and to storage of condensate in the system, but wherever this has been investigated it has been found to be negligible.

The load factor of a building is the ratio of the average load to the maximum load and is an index of the utilization habits. Thus, in Table 9, the theaters, operating for short hours, have a load factor of 0.126 as compared with the figure of 0.318 for clubs and lodges.

SEASONAL EFFICIENCY

The task of predicting fuel consumption within reasonably accurate limits is a simple one where sufficient experience data are available for the fuel in question. Such data can be analyzed to the point where average unit factors can be determined and expressed in such terms as, for example, average gallons of oil actually burned per square foot of calculated steam radiator surface per degree-day. The unit U can be inserted directly in Equation 3 without reference to efficiency. Such experience factors are available for gas (see Table 4) and for district steam (Table 8), but not for coal or oil.

Since values of U are not available for oil or coal, an assumed seasonal efficiency E must be used. Selection of a value for this E must be made with caution, for its use implies a meaning not commonly associated with the word efficiency and consequently is frequently misleading.

The input of heat to a building consists not only of the energy in the fuel but that from occupants, the sun, appliances, processes, and all other sources. In many cases these make up, over a period, an important percentage of the total heat required, and if they are not taken into account a calculation of *efficiency* can show a figure over 100 per cent.

For this and other reasons the actual seasonal efficiency is a difficult thing to determine. Published data are widely scattered and insufficient. From the available published material it is found that the seasonal efficiency varies over a wide range, depending on the fuel used, and it varies widely even for a given fuel. For example, in a recent survey of 30 houses in one locality there was found a variation of from 45 to 75 per cent in the utilization efficiency depending on the fuel[5].

[5]Heat Losses and Efficiencies of Fuels in Residential Heating, by R. A. Sherman and R. C. Cross, (A.S.H.V.E. Transactions, Vol. 43, 1937, p. 185).

Chapter 12

HEATING BOILERS

Cast-Iron Boilers, Steel Boilers, Special Heating Boilers, Gas-Fired Boilers, Hot Water Supply Boilers, Furnace Design, Heating Surface, Testing and Rating Codes, Output Efficiency, Selection of Boilers, Connections and Fittings, Erection, Operation and Maintenance, Boiler Insulation

STEAM and hot water boilers for low pressure heating work are built in a wide variety of types, many of which are illustrated in the *Catalog Data Section*, and are classified as (1) cast-iron sectional, (2) steel fire tube, (3) steel water tube, and (4) special.

CAST-IRON BOILERS

Cast-iron boilers usually fall into one of two general classifications (1) rectangular pattern with vertical sections and rectangular grate commonly known as sectional boilers, (2) round pattern with horizontal pancake sections and circular grates commonly known as round boilers. A few boilers of the sectional type use outside header construction where each section is independent of the other and the water and steam connections are made externally through these headers. The majority of boilers, however, both sectional and round, are assembled with push nipples and tie rods at the top and bottom of the sections in which case water and steam connections are internal. The present trend in design of sectional boilers is to use a large top push nipple so that in a steam boiler the water line may be carried through the top push nipple thereby permitting circulation of water between adjacent sections at both the top and bottom of the water content of the boiler. The primary purpose of this construction is to eliminate the necessity of connecting the sections below the water line with an external header to permit circulation of the water from one section to the other which is necessary in the case of a steam boiler equipped with an indirect water heater for summer-winter hot water supply.

Round and sectional boilers may be increased in size by the addition of sections which in the case of sectional boilers also increases the grate area. The grate area of round boilers remains the same as additional sections are added.

Cast-iron boilers are usually shipped knocked down. This facilitates handling at the place of installation where assembly is made in that separate sections can be taken into or out of basements and other places more or less inaccessible after the building is constructed. This feature

is of importance in the original installation of the boiler and also in making repairs to or replacing a damaged boiler at a later date.

Cast-iron boilers may be designed to burn efficiently one kind of fuel only or various kinds of fuel. Practical combustion rates for coal-fired boilers are given in Table 1. Many recent designs of oil burning boilers have been designed exclusively for oil fuel and in some cases for both oil and gas. The present trend in the design of boilers for hand firing is, however, to design them so they will be suitable for ready conversion to and efficient operation with oil and stoker firing even after the boiler has been installed and has been operating for a period of time on one type of fuel.

Table 1. Practical Combustion Rates for Coal-Fired Heating Boilers Operating at Maximum Load on Natural Draft of from 1/8 in. to 1/2 in. Water[a]

Kind of Coal	Sq Ft Grate	Lb of Coal per Sq Ft Grate per Hour
No. 1 Buckwheat Anthracite	Up to 4	3
	5 to 9	3½
	10 to 14	4
	15 to 19	4½
	20 to 25	5
Anthracite Pea	Up to 9	5
	10 to 19	5½
	20 to 25	6
Anthracite Nut and Larger	Up to 4	8
	5 to 9	9
	10 to 14	10
	15 to 19	11
	20 to 25	13
Bituminous	Up to 4	9.5
	5 to 14	12
	15 and above	15.5

[a]Steel boilers usually have higher combustion rates for grate areas exceeding 15 sq ft than those indicated in this table.

Capacities of cast-iron boilers range from that required for small residences up to about 18,000 sq ft of radiation. For larger loads boilers must be installed in multiple. The maximum allowable working pressure for cast-iron boilers is limited by the *A.S.M.E.* Code to 15 lb per square inch for steam boilers. Hot water boilers are usually limited to 30 lb per square inch maximum working pressure but may be designed for higher pressures where required for heating purposes or for hot water supply where the boiler must withstand high local water pressures.

STEEL BOILERS

Steel heating boilers may be classified according to (a) position of combustion gas with respect to tube surface, (b) arrangement and construction of furnaces, and (c) type of fuel and method of firing.

Fire tube boilers are those in which the gases of combustion pass through the tubes and the boiler water circulates around them. In water

tube boilers, the gases circulate around the tubes and the water passes through them.

Steel heating boilers may be furnished with integral water jacketed furnaces or arranged for refractory lined brick or refractory lined jacketed furnaces. Those with integral water jacketed furnaces are called *portable firebox boilers* and are the most commonly used type. They may be either fire tube or water tube and are furnished for any fuel and method of firing used in heating boiler practice. They are usually shipped from the factory in one piece, ready for piping. Bridge-walls and smokeless furnace parts are shipped in place when furnished. Boilers with refractory lined furnaces may be either fire tube or water tube. They also may be arranged for any fuel or method of firing. Refractory furnaces are usually installed in such boilers after they are set.

SPECIAL HEATING BOILERS

A special type of boiler, known as the *magazine feed boiler*, has been developed for the burning of small sizes of anthracite and coke. These are built of both cast-iron and steel, and have a large fuel carrying capacity which results in longer firing periods than would be the case with the standard types using buckwheat sizes of coal. Special attention must be given to insure adequate draft and proper chimney sizes and connections.

Oil-burner boiler units, in which a special boiler has been designed with a furnace shaped to meet the general requirements of oil burners, or are specially adapted to one particular burner, have been developed by a number of manufacturers. These usually are compact units with the burner and all controls enclosed within an insulated steel jacket. Ample furnace volume is provided for efficient combustion, and the heating surfaces are proportioned for effective heat transfer. Consequently, higher efficiencies are obtainable than with the ordinary coal-fired boiler designed primarily for hand firing and converted to oil firing.

GAS-FIRED BOILERS

Gas boilers have assumed a well-defined individuality. The usual boiler is sectional in construction with a number of independent burners placed beneath the sections. In most boilers each section has its own burner. In all cases the sections are placed quite closely together, much closer than would be possible when burning a soot-forming fuel. The effort of the designer is always to break the hot gas up into thin streams, so that all particles of the heat-carrying gases can come as close as possible to the heat-absorbing surfaces. Because there is no fuel bed resistance and because the gas company supplies the motive power to draw in the air necessary for combustion (in the form of the initial gas pressure), draft losses through gas boilers are low. (See Chapter 10.)

Most gas-fired boilers carry the approval of the *American Gas Association.* In order to obtain this approval the boilers must be submitted to the *American Gas Association Testing Laboratory* and meet the Approval Requirements for Central Heating Gas Appliances issued by the *American Gas Association.* The boiler ratings must be such that they meet the limitations as set forth in these Approval Requirements.

HOT WATER SUPPLY BOILERS

Boilers for hot water supply are classified as direct, if the water heated passes through the boiler, and as indirect, if the water heated does not come in contact with the water or steam in the boiler.

Direct heaters are built to operate at the pressures found in city supply mains and are tested at pressures from 200 to 300 lb per square inch. The life of direct heaters depends almost entirely on the scale-making properties of the water supplied. If water temperatures are maintained below 140 F the life of the heater will be much longer than if higher temperatures are used, owing to decreased scale formation and minimized corrosion below 140 F. Direct water heaters in some cases are designed to burn refuse and garbage.

Indirect heaters generally consist of steam boilers in connection with heat exchangers of the coil or tube types which transmit the heat from the steam to the water. This type of installation has the following advantages:

1. The boiler operates at low pressure.
2. The boiler is protected from scale and corrosion.
3. The scale is formed in the heat exchanger in which the parts to which the scale is attached can be cleaned or replaced. The accumulation of scale does not affect efficiency although it will affect the capacity of the heat exchanger.
4. Discoloration of water may be prevented if the water supply comes in contact with only non-ferrous metal.

Where a steam heating system is installed, the domestic hot water usually is obtained from an indirect heater placed below the water line of the boiler. Indirect heaters may also be used with hot water heating systems to obtain domestic hot water and should be located as high as possible with respect to the boiler for most satisfactory performance.

FURNACE DESIGN

Good efficiency and proper boiler performance are dependent on correct furnace design embodying sufficient volume for burning the particular fuel at hand, which requires thorough mixing of air and gases at a high temperature with a velocity low enough to permit complete combustion of all the volatiles. On account of the small amount of volatiles contained in coke, anthracite, and semi-bituminous coal, these fuels can be burned efficiently with less furnace volume than is required for bituminous coal, the combustion space being proportioned according to the amount of volatiles present.

Combustion should take place before the gases are cooled by the boiler heating surface, and the volume of the furnace must be sufficient for this purpose. The furnace temperature must be maintained sufficiently high to produce complete combustion, thus resulting in a higher CO_2 content and the absence of CO. Hydrocarbon gases ignite at temperatures varying from 1000 to 1500 F.

The question of furnace proportions, particularly in regard to mechanical stoker installations, has been given some consideration by various manufacturers' associations. Arbitrary values have been recommended for minimum dimensions. A customary rule-of-thumb method of figuring furnace volumes is to allow 1 cu ft of space for a maximum heat release

of 50,000 Btu per hour. This value is equivalent to allowing approximately 1 cu ft for each developed horsepower, and it is approved by most smoke prevention organizations.

The setting height will vary with the type of stoker. In an overfeed stoker, for instance, all the volatiles must be burned in the combustion chamber and, therefore, a greater distance should be allowed than for an underfeed stoker where a considerable portion of the gas is burned while passing through the incandescent fuel bed. The design of the boiler also may affect the setting height, since in certain types the gas enters the tubes immediately after leaving the combustion chamber, while in others it passes over a bridge wall and toward the rear, thus giving a better opportunity for combustion by obtaining a longer travel before entering the tubes.

To secure suitable furnace volume, especially for mechanical stokers or oil burners, it often is necessary either to pit the stoker or oil burner, or, where water line conditions and headroom permit, to raise the boiler on a brick foundation setting.

Smokeless combustion of the more volatile bituminous coals is furthered by the use of mechanical stokers. (See Chapter 10.) Smokeless combustion in hand-fired boilers burning high volatile solid fuel is aided (1) by the use of double grates with down-draft through the upper grate, (2) by the use of a curtain section through which preheated auxiliary air is introduced over the fire toward the rear of the boiler, and (3) by the introduction of preheated air through passages at the front of the boiler. All three methods depend largely on mixing secondary air with the partially burned volatiles and causing this mixture to pass over an incandescent fuel bed, thus tending to secure more complete combustion than is possible in boilers without such provision.

HEATING SURFACE

Boiler heating surface is that portion of the surface of the heat transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other side. Heating surface on which the fire shines is known as *direct* or radiant surface and that in contact with hot gases only, as *indirect* or convection surface. The amount of heating surface, its distribution and the temperatures on either side thereof influence the capacity of any boiler.

Direct heating surface is more valuable than indirect per square foot because it is subjected to a higher temperature and also, in the case of solid fuel, because it is in position to receive the full radiant energy of the fuel bed. The heat transfer capacity of a radiant heating surface may be as high as 6 to 8 times that of an indirect surface. This is one of the reasons why the water legs of some boilers have been extended, especially in the case of stoker firing where the extra amount of combustion chamber secured by an extension of the water legs is important. For the same reason, care should be exercised in building a refractory combustion chamber in an oil-burning boiler so as not to screen any more of this valuable surface with refractories than is necessary for good combustion.

The effectiveness of the heating surface depends on its cleanliness, its location in the boiler, and the shape of the gas passages. The area of the

gas passages must not be so small as to cause excessive resistance to the flow of gases where natural draft is employed. Inserting baffles so that the heating surface is arranged in series with respect to the gas flow increases boiler efficiency and reduces stack temperature and increases the draft loss through the boiler.

Heat Transfer Rates

Practical rates of heat transfer in heating boilers will average about 3300 Btu per square foot per hour for hand-fired boilers and 4000 Btu per square foot per hour for mechanically fired boilers when operating at *design load.* When operating at *maximum load*[1] these values will run between 5000 and 6000 Btu per square foot per hour. Boilers operating under favorable conditions at the above heat transfer rates will give exit gas temperatures that are considered consistent with good practice.

TESTING AND RATING CODES

The Society has adopted four solid fuel testing codes, a solid fuel rating code and an oil fuel testing code. A.S.H.V.E. Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers—Codes 1 and 2—(Revision of June 1929)[2], are intended to provide a method for conducting and reporting tests to determine heat efficiency and performance characteristics. A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers—Code No. 3—(Edition of 1929)[2] is intended for use with A.S.H.V.E. Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers[3]. The object of this test code is to specify the tests to be conducted and to provide a method for conducting and reporting tests to determine the efficiencies and performance of the boiler. The A.S.H.V.E. Standard Code for Testing Steam Heating Boilers Burning Oil Fuel[4] is intended to provide a standard method for conducting and reporting tests to determine the heating efficiency and performance characteristics when oil fuel is used with steam heating boilers. In 1938 the Society adopted a Standard Code for Testing Stoker-Fired Steam Heating Boilers[5] which is intended to provide a test method for determining the efficiency and performance characteristics of any stoker or boiler combination burning any type of solid fuel such as anthracite or bituminous coal.

The *Steel Heating Boiler Institute* has adopted a method for the rating of low pressure boilers based on their physical characteristics and expressed in square feet of steam or water radiation or in Btu per hour as given in Table 2. The detailed requirements of this code were outlined in Chapter 13 of The Guide 1939. The *Institute of Boiler and Radiator Manufacturers* has also adopted a method of rating cast-iron heating boilers based upon performance obtained under tests. This code became effective August 1, 1939 for sectional boilers of 20 in. width grate or less, but the *Institute* intends eventually to expand the code to apply to all

[1]For definitions of design load and maximum load see page 242.
[2]See A.S.H.V.E. Transactions, Vol. 35, 1929, pp. 322 and 332.
[3]See A.S.H.V.E. Transactions, Vol. 36, 1930, p. 42.
[4]See A.S.H.V.E. Transactions, Vol. 37, 1931, p. 23.
[5]See A.S H.V.E. Transactions, Vol. 44, 1938, p. 366.

TABLE 2. STEEL HEATING BOILER STANDARD RATINGS[a]

Hand-Fired Rating						Mechanically-Fired Rating				
Catalog			Net Load			Catalog			Net Load	
Steam Radiation Sq Ft	Water Radiation Sq Ft	Btu per Hour in Thousands	Steam Radiation Sq Ft	Heating Surface Sq Ft	Grate Area Sq Ft	Steam Radiation Sq Ft	Water Radiation Sq Ft	Btu per Hour in Thousands	Steam Radiation Sq Ft	Furnace Volume, Oil, Gas or Bituminous Coal Cu Ft
1,800	2,880	432	1,389	129	7.9	2,190	3,500	525	1,695	15.7
2,200	3,520	528	1,702	158	8.9	2,680	4,280	643	2,089	19.2
2,600	4,160	624	2,020	186	9.7	3,160	5,050	758	2,461	22.6
3,000	4,800	720	2,335	215	10.5	3,650	5,840	876	2,853	26.1
3,500	5,600	840	2,732	250	11.4	4,250	6,800	1,020	3,335	30.4
4,000	6,400	960	3,135	286	12.2	4,860	7,770	1,166	3,830	34.8
4,500	7,200	1,080	3,540	322	13.4	5,470	8,750	1,312	4,330	39.1
5,000	8,000	1,200	3,945	358	14.5	6,080	9,720	1,459	4,834	43.5
6,000	9,600	1,440	4,770	429	16.4	7,290	11,660	1,749	5,850	52.1
7,000	11,200	1,680	5,608	500	18.1	8,500	13,600	2,040	6,885	60.8
8,500	13,600	2,040	6,885	608	20.5	10,330	16,520	2,479	8,490	73.8
10,000	16,000	2,400	8,197	715	22.5	12,150	19,440	2,916	10,125	86.8
12,500	20,000	3,000	10,417	893	25.6	15,180	24,280	3,643	12,650	108.5
15,000	24,000	3,600	12,500	1,072	28.4	18,220	29,150	4,372	15,183	130.2
17,500	28,000	4,200	14,584	1,250	30.9	21,250	34,000	5,100	17,708	151.8
20,000	32,000	4,800	16,667	1,429	33.2	24,290	38,860	5,829	20,242	173.5
25,000	40,000	6,000	20,834	1,786	37.4	30,360	48,570	7,286	25,300	216.9
30,000	48,000	7,200	25,000	2,143	41.2	36,430	58,280	8,743	30,359	260.3
35,000	56,000	8,400	29,167	2,500	44.7	42,500	68,000	10,200	35,417	303.6

[a]Adopted by the *Steel Heating Boiler Institute* in cooperation with the *Bureau of Standards, United States Department of Commerce Simplified Practice Recommendations R 157-35.*

sizes of boilers. Methods of testing hand-fired and oil-fired boilers are specified and are referred to as *IBR* testing codes.

BOILER OUTPUT

Boiler output as defined in A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3) is the quantity of heat available at the boiler nozzle with the boiler normally insulated. It should be based on actual tests conducted in accordance with this code. This output is usually stated in Btu and in square feet of equivalent heating surface (radiation). According to the A.S.H.V.E. Standard Code for Rating Steam Heating, Solid Fuel Hand-Fired Boilers, the performance data should be given in tabular or curve form on the following items for at least five outputs ranging from maximum down to 35 per cent of maximum: (1) fuel available, (2) combustion rate, (3) efficiency, (4) draft tension, (5) flue gas temperature. The only definite restriction placed on setting the maximum output is that priming shall not exceed 2 per cent. These curves provide complete data regarding the performance of the boiler under test conditions. Certain other pertinent information, such as grate area, heating surface and chimney dimensions, is desirable also in forming an opinion of how the boiler will perform in actual service.

The output of large heating boilers is frequently stated in terms of *boiler horsepower* instead of in Btu per hour or square feet of equivalent radiation.

BOILER EFFICIENCY

The term *efficiency* as used for guarantees of boiler performance is usually construed as follows:

1. *Solid Fuels.* The efficiency of the *boiler* alone is the ratio of the heat absorbed by the water and steam in the boiler per pound of combustible burned on the grate to the calorific value of 1 lb of combustible as fired. The *combined* efficiency of *boiler, furnace and grate* is the ratio of the heat absorbed by the water and steam in the boiler per pound of fuel as fired to the calorific value of 1 lb of fuel as fired.

2. *Liquid and Gaseous Fuels.* The *combined* efficiency of boiler, furnace and burner is the ratio of the heat absorbed by the water and steam in the boiler per pound or cubic foot of fuel to the calorific value of 1 lb or cubic foot of fuel respectively.

Solid fuel boilers usually show an efficiency of 50 to 75 per cent when operated under favorable conditions at their rated capacities. Information on the combined efficiencies of boiler, furnace and burner has resulted from research conducted at Yale University in cooperation with the A.S.H.V.E. Research Laboratory and the *American Oil Burner Association*[6].

SELECTION OF BOILERS

Estimated Design Load: The load, stated in Btu per hour or equivalent direct radiation, as estimated by the purchaser for the conditions of inside and outside temperature for which the amount of installed radiation was determined, is equivalent to the sum of the heat emission of the radiation to be actually installed, the allowance for the heat loss of the connecting piping, and the heat requirement for any apparatus requiring heat connected to the system.

The estimated design load is the sum of the following three items[7]:

1. The estimated heat emission in Btu per hour of the connected radiation (direct, indirect or central fan) to be installed.

2. The estimated maximum heat in Btu per hour required to supply water heaters or other apparatus to be connected to the boiler.

3. The estimated heat emission in Btu per hour of the piping connecting the radiation and other apparatus to the boiler.

Estimated Maximum Load: Construed to mean the load stated in Btu per hour or the equivalent direct radiation that has been estimated by the purchaser to be the greatest or maximum load that the boiler will be called upon to carry.

The estimated maximum load is given by[8]:

4. The estimated increase in the normal load in Btu per hour due to starting up cold radiation. This percentage of increase is to be based on the sum of Items 1, 2 and 3 and the heating-up factors given in Table 3.

Other things to be considered are:

5. Efficiency with hard or soft coal, gas, or oil firing, as the case may be.

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[6]A.S.H.V.E. Research Report No. 907—Study of the Characteristics of Oil Burners and Heating Boilers, by L. E. Seeley and E. J. Tavanlar (A.S.H.V.E. Transactions. Vol. 37, 1931, p. 517). A.S.H.V.E. Research Report No. 925—A Study of Intermittent Operation of Oil Burners, by L. E. Seeley and J. H. Powers (A.S.H V.E. Transactions, Vol. 38, 1932, p. 317).

[7]A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929).

[8]Loc. Cit. Note 7.

6. Grate area with hand-fired coal, or fuel burning rate with stokers, oil, or gas.
7. Combustion space in the furnace.
8. Type of heat liberation, whether continuous or intermittent, or a combination of both.
9. Miscellaneous items consisting of draft available, character of attendance, possibility of future extension, possibility of breakdown and headroom in the boiler room.

Radiation Load

The connected radiation (Item 1) is determined by calculating the heat losses in accordance with data given in Chapters 4, 5 and 6, and dividing by 240 to change to square feet of equivalent radiation as explained in Chapter 13. For hot water, the emission commonly used is 150 Btu per square foot, but the actual emission depends on the temperature of the medium in the heating units and of the surrounding air. (See Chapter 13.)

Although it is customary to use the actual connected load in equivalent square feet of radiation for selecting the size of boiler, this connected load usually represents a reserve in heating capacity to provide for infiltration

TABLE 3. WARMING-UP ALLOWANCES FOR LOW PRESSURE STEAM AND HOT WATER HEATING BOILERS[a, b, c]

Design Load (Representing Summation of Items 1, 2, and 3,[d]		Percentage Capacity to Add for Warming Up
Btu per Hour	Equivalent Square Feet of Radiation[d]	
Up to 100,000	Up to 420	65
100,000 to 200,000	420 to 840	60
200,000 to 600,000	840 to 2500	55
600,000 to 1,200,000	2500 to 5000	50
1,200,000 to 1,800,000	5000 to 7500	45
Above 1,800,000	Above 7500	40

[a]This table is taken from the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings, except that the second column has been added for convenience in interpreting the design load in terms of equivalent square feet of radiation.

[b]See also Time Analysis in Starting Heating Apparatus, by Ralph C. Taggert (A.S.H.V.E. TRANSACTIONS, Vol. 19, 1913, p. 292); Report of A.S.H.V.E. Continuing Committee on Codes for Testing and Rating Steam Heating Solid Fuel Boilers (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 35); Selecting the Right Size Heating Boiler, by Sabin Crocker (*Heating, Piping and Air Conditioning*, March, 1932).

[c]This table refers to hand-fired, solid fuel boilers. A factor of 20 per cent over design load is adequate when automatically-fired fuels are used (see Fig. 1).

[d]240 Btu per square foot.

in the various spaces of the building to be heated, which reserve, however, is not in use at all places at the same time, or in any one place at all times. For a further discussion of this subject see Chapter 5.

Hot Water Supply Load

When the hot water supply (Item 2) is heated by the building heating boiler, this load must be taken into consideration in sizing the boiler. The allowance to be made will depend on the amount of water heated and its temperature rise. A good approximation is to add 4 sq ft of equivalent radiation for each gallon of water heated per hour through a temperature range of 100 F. For more specific information, see Chapter 46.

Piping Tax (Item 3)

It is common practice to add a flat percentage allowance to the equivalent connected radiation to provide for the heat loss from bare and covered pipe in the supply and return lines. The use of a flat allowance of

25 per cent for steam systems and 35 per cent for hot water systems is preferable to ignoring entirely the load due to heat loss from the supply and return lines, but better practice, especially when there is much bare pipe, is to compute the emission from both bare and covered pipe surface in accordance with data in Chapter 43. A chart is shown in Fig. 1 indicating percentage allowances for piping and warming-up which are applicable to automatically-fired heating plants using steam radiation. With direct radiation served by bare supply and return piping the percentages may be higher than those stated, while in the case of unit heaters where the output is concentrated in a few locations, the piping tax may be 10 per cent or less.

Warming-Up Allowance

The warming-up allowance represents the load due to heating the boiler and contents to operating temperature and heating up cold radiation and piping. (See Item 4.) The factors to be used for determining the

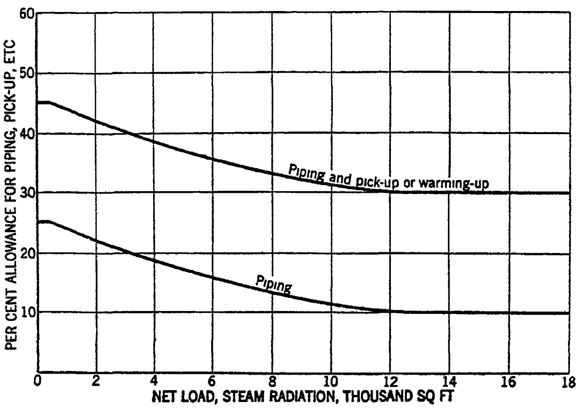


FIG. 1. PERCENTAGE ALLOWANCE FOR PIPING AND WARMING-UP

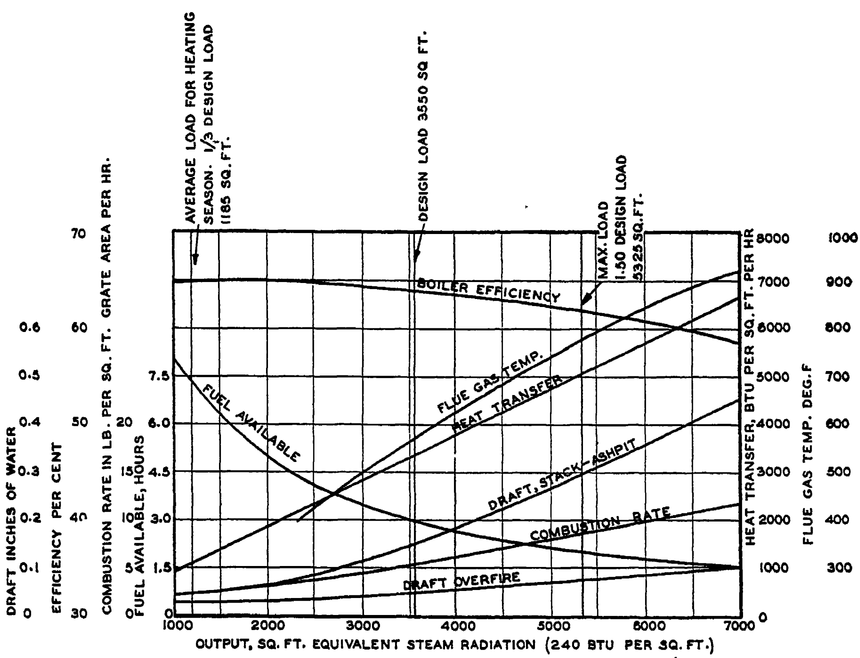
allowance to be made should be selected from Table 3 and should be applied to the estimated design load as determined by Items 1, 2 and 3. While in every case the estimated maximum load will exceed the design load if adequate heating response is to be achieved, there is, however, no object in over-estimating the allowances, as the only effect would be to reduce the time of warming-up by a few minutes. Otherwise, it might result in firing the boiler unduly and increasing the cost of operation.

Performance Curves for Boiler Selection

In the selection of a boiler to meet the estimated load, the A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers recommends the use of performance curves based on actual tests conducted in accordance with the A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3), similar to the typical curves shown in Fig. 2. It should be understood that performance data apply to test conditions and that a reasonable allowance should be made for decreased output resulting from soot deposit, poor fuel or inefficient attention.

Selection Based on Heating Surface and Grate Area

Where performance curves are not available, a good general rule for conventionally-designed boilers is to provide 1 sq ft of boiler heating surface for each 14 sq ft of equivalent radiation (240 Btu per square foot) represented by the design load consisting of connected radiation, piping tax and domestic water heating load. As stated in the section on Boiler Output, this is equivalent to allowing 10 sq ft of boiler heating surface per



BOILER DATA—

GRATE AREA, SQ. FT.	18.0
HEATING SURFACE, SQ. FT.	254
WEIGHT, LB.	9160
FUEL CAPACITY, LB.	651
FUEL AVAILABLE, LB.	414
FUEL DEPTH, IN	10

FUEL—BITUMINOUS 3/4" LUMP

ANALYSIS—

VOLATILE MATTER	34.68%
FIXED CARBON	55.44
ASH	9.87
SULPHUR	2.86
MOISTURE	3.00
BTU PER LB.	13,655

FIG. 2. TYPICAL PERFORMANCE CURVES FOR A 36-IN. CAST-IRON SECTIONAL STEAM HEATING BOILER, BASED ON THE A.S.H.V.E. CODE FOR RATING STEAM HEATING SOLID FUEL HAND-FIRED BOILERS

boiler horsepower. In this case it is assumed that the maximum load including the warming-up allowance will be provided for by operating the boiler in excess of the design load, that is, in excess of the 100 per cent rating on a boiler-horsepower basis.

Due to the wide variation encountered in manufacturers' ratings for boilers of approximately the same capacity, it is advisable to check the grate area required for heating boilers burning solid fuel by means of the following formula:

$$G = \frac{H}{C \times F \times E} \tag{1}$$

where

G = grate area, square feet.

H = required total heat output of the boiler, Btu per hour (see Selection of Boilers, p. 242).

C = combustion rate in pounds of dry coal per square foot of grate area per hour, depending on the kind of fuel and size of boiler as given in Table 1.

F = calorific value of fuel, Btu per pound.

E = efficiency of boiler, usually taken as 0.60.

Example 1. Determine the grate area for a required heat output of the boiler of 500,000 Btu per hour, a combustion rate of 6 lb per hour, a calorific value of 13,000 Btu per pound, and an efficiency of 60 per cent.

$$G = \frac{500{,}000}{6 \times 13{,}000 \times 0.60} = 10.7 \text{ sq ft}$$

The boiler selected should have a grate area not less than that determined by Formula 4. With small boilers where it is desired to provide sufficient coal capacity for approximately an eight-hour firing period plus a 20 per cent reserve for igniting a new charge, more grate area may be required depending upon the depth of the fuel pot.

Selection of Steel Heating Boilers

Ratings obtained from the previously mentioned *Steel Heating Boiler Institute* code are intended to correspond with the estimated design load based on the sum of items 1, 2 and 3 outlined on page 242. Boilers with less than 128 sq ft of heating surface are classified as residence size. An insulated residence boiler for oil or gas, not convertible, may carry a net load expressed in square feet of steam radiation of not more than 17 times the square feet of heating surface in the boiler, provided the boiler manufacturer guarantees it to be capable of operating at a maximum output of not less than 150 per cent of net load rating with overall efficiency of not less than 75 per cent with at least two different makes of each type of standard commercial burner recommended by the boiler manufacturer. If the heat loss from the piping system exceeds 20 per cent of the installed radiation, the excess is to be considered as a part of the net load.

When the estimated heat emission of the piping (connecting radiation, and other apparatus to the boiler) is not known the net load to be considered for the boiler may be determined from Table 2.

Selection of Gas-Fired Boilers

Gas-heating appliances should be selected in accordance with the percentage allowances given in Fig. 1. These factors are for thermostatically-controlled systems; in case manual operation is desired, a warming-up allowance of 100 per cent is recommended by the *A.G.A.* A gas boiler selected by the use of the *A.G.A.* factors will be the minimum size boiler which can carry the load. From a fuel economy standpoint, it may be advisable to select a somewhat larger boiler and then throttle the gas and air adjustments as required. This will tend to give a low stack temperature with high efficiency and at the same time provide reserve capacity in case the load is under-estimated or more is added in the future.

Conversions

The conversion of a coal or oil boiler to gas burning is simpler than the reverse since little furnace volume need be provided for the proper combustion of gas. When a solid fuel boiler of 500 sq ft (or less) capacity is converted to gas burning, the necessary gas heat units should be approximately double the connected load. The presumption for a conversion job is that the boiler is installed and probably will not be made larger; therefore, it is a matter of setting a gas-burning rate to obtain best results with the available surface. Assuming a combustion efficiency of 75 per cent for a conversion installation the boiler output would be $2 \times 0.75 = 1.5$ times the connected load, which allows 50 per cent for piping tax and pickup. In converting large boilers, the determination of the required Btu input should not be done by an arbitrary figure or factor but should be based on a detailed consideration of the requirements and characteristics of the connected load.

An efficient conversion installation depends upon the proper size of flue connection. Often the original smoke breeching between the boiler and chimney is too large for gas firing, and in this case, flue orifices can be used. They are discs provided with an opening of the size for the gas input used in this boiler. The size should be based on 1 sq in. of flue area for each 7500 hourly Btu input.

If dampers are found in the breeching they should be locked in position so that they will not interfere with the normal operation of the gas burners at maximum flow. In the case of large boiler conversions, automatic damper regulators proportion the position of the flue dampers to the amount of gas flowing and may be substituted for existing dampers. Generally in residence conversions automatic dampers are not of the proportioning type but close the flue during the off periods of the gas burners. Automatic shut-off dampers should be located between the back draft diverter and the chimney flue. Automatic dampers are usually designed to operate with electric contact mechanism, but frequently an arrangement is utilized which functions with mechanical fluid or gas pressure.

Physical Limitations

As it will usually be found that several boilers will meet the specifications, the final selection may be influenced by other considerations, as:

1. Dimensions of boiler.
2. Durability under service.
3. Convenience in firing and cleaning.
4. Adaptability to changes in fuel and kind of attention.
5. Height of water line.

In large installations, the use of several smaller boiler units instead of one larger one will obtain greater flexibility and economy by permitting the operation, at the best efficiency, of the required number of units according to the heat requirements.

Space Limitations

Boiler rooms should, if possible, be situated at a central point with respect to the building and should be designed for a maximum of natural

light. The space in front of the boilers should be sufficient for firing, stoking, ash removal and cleaning or renewal of flues, and should be at least 3 ft greater than the length of the boiler firebox.

A space of at least 3 ft should be allowed on at least one side of every boiler for convenience of erection and for accessibility to the various dampers, cleanouts and trimmings. The space at the rear of the boiler should be ample for the chimney connection and for cleanouts. With large boilers the rear clearance should be at least 3 ft in width.

The boiler room height should be sufficient for the location of boiler accessories and for proper installation of piping. In general the ceiling height for small steam boilers should be at least 3 ft above the normal boiler water line. With vapor heating, especially, the height above the boiler water line is of vital importance.

When steel boilers are used, space should be provided for the removal and replacement of tubes.

CONNECTIONS AND FITTINGS

Steam or water outlet connections preferably should be the full size of the manufacturers' tappings in order to keep the velocity of flow through the outlet reasonably low and avoid fluctuation of the water line and undue entrainment of moisture, and should extend vertically to the maximum height available above the boiler.

Particular attention should be given to *fitting connections* to secure conformity with the *A.S.M.E.* Boiler Construction Code for Low Pressure Heating Boilers. Attention is called in particular to pressure gage piping, water gage connections and safety valve capacity.

Steam gages should be fitted with a water seal and a shut-off consisting of a cock with either a tee or lever handle which is parallel to the pipe when the cock is open. *Steam gage connections* should be of copper or brass when smaller than 1 in. *I.P.S.*[9] if the gage is more than 5 ft from the boiler connection, and also in any case where the connection is less than ½ in. *I.P.S.*

Each steam or vapor boiler should have at least one *water gage* glass and two or more *gage cocks* located within the range of the visible length of the glass. The water gage fittings or gage cocks may be directly connected to the boiler, if so located by the manufacturer, or may be mounted on a separate water column. No connections, except for combustion regulators, drains or steam gages, should be placed on the pipes connecting the water column and the boiler. If the water column or gage glass is connected to the boiler by pipe and fittings, a cross, tee or equivalent, in which a cleanout plug or a drain valve and piping may be attached, should be placed in the water connection at every right-angle turn to facilitate cleaning. The water line in steam boilers should be carried at the level specified by the boiler manufacturer.

Safety valves should be capable of discharging all the steam that can be generated by the boiler without allowing the pressure to rise more than 5 lb above the maximum allowable working pressure of the boiler. This

[9]*A.S.M.E.* Code, Identification of Piping Systems.

should be borne in mind particularly in the case of boilers equipped with mechanical stokers or oil burners where the amount of grate area has little significance as to the steam generating capacity of the boiler.

Where a *return header* is used on a cast-iron sectional boiler to distribute the returns to both rear tappings, it is advisable to provide full size plugged tees instead of elbows where the branch connections enter the return tappings. This facilitates cleaning sludge from the bottom of the boiler sections through the large plugged openings. An equivalent clean-out plug should be provided in the case of a single return connection.

Blow-off or drain connections should be made near the boiler and so arranged that the entire system may be drained of water by opening the drain cock. In the case of two or more boilers separate blow-off connections must be provided for each boiler on the boiler side of the stop valve on the main return connection.

Water service connections must be provided for both steam and water boilers, for refilling and for the addition of make-up water to boilers. This connection is usually of galvanized steel pipe, and is made to the return main near the boiler or boilers.

For further data on pipe connections for steam and hot water heating systems, see Chapters 15 and 16 and the *A.S.M.E.* Boiler Construction Code for Low Pressure Heating Boilers.

Smoke Breeching and Chimney Connections. The breeching or smoke pipe from the boiler outlet to the chimney should be air-tight and as short and direct as possible, preference being given to long radius and 45-deg instead of 90-deg bends. The breeching entering a brick chimney should not project beyond the flue lining and where practicable it should be grouted from the inside of the chimney. A thimble or sleeve grout usually is provided where the breeching enters a brick chimney.

Where a battery of boilers is connected into a breeching each boiler should be provided with a tight damper. The breeching for a battery of boilers should not be reduced in size as it goes to the more remote boilers. Good connections made to a good chimney will usually result in a rapid response by the boilers to demands for heat.

ERECTION, OPERATION, AND MAINTENANCE

The directions of the boiler manufacturer always should be read before the assembly or installation of any boiler is started, even though the contractor may be familiar with the boiler. All joints requiring boiler putty or cement which cannot be reached after assembly is complete must be finished as the assembly progresses.

The following precautions should be taken in all installations to prevent damage to the boiler:

1. There should be provided proper and convenient drainage connections for use if the boiler is not in operation during freezing weather.

2. Strains on the boiler due to movement of piping during expansion should be prevented by suitable anchoring of piping and by proper provision for pipe expansion and contraction.

3. Direct impingement of too intense local heat upon any part of the boiler surface, as with oil burners, should be avoided by protecting the surface with firebrick or other refractory material.

4. Condensation must flow back to the boiler as rapidly and uniformly as possible. Return connections should prevent the water from backing out of the boiler.

5. Automatic boiler feeders and low water cut-off devices which shut off the source of heat if the water in the boiler falls below a safe level are recommended for boilers mechanically fired.

Boiler Troubles

A complaint regarding boiler operation generally will be found to be due to one of the following:

1. *The boiler fails to deliver enough heat.* The cause of this condition may be: (*a*) poor draft; (*b*) poor fuel; (*c*) inferior attention or firing; (*d*) boiler too small; (*e*) improper piping; (*f*) improper arrangement of sections; (*g*) heating surfaces covered with soot; and (*h*) insufficient radiation installed.

2. *The water line is unsteady.* The cause of this condition may be: (*a*) grease and dirt in boiler; (*b*) water column connected to a very active section and, therefore, not showing actual water level in boiler; and (*c*) boiler operating at excessive output.

3. *Water disappears from gage glass.* This may be caused by: (*a*) priming due to grease and dirt in boiler; (*b*) too great pressure difference between supply and return piping preventing return of condensation; (*c*) valve closed in return line; (*d*) connection of bottom of water column into a very active section or thin waterway; and (*e*) improper connections between boilers in battery permitting boiler with excess pressure to push returning condensation into boiler with lower pressure.

4. *Water is carried over into steam main.* This may be caused by: (*a*) grease and dirt in boiler; (*b*) insufficient steam dome or too small steam liberating area; (*c*) outlet connections of too small area; (*d*) excessive rate of output; and (*e*) water level carried higher than specified.

5. *Boiler is slow in response to operation of dampers.* This may be due to: (*a*) poor draft resulting from air leaks into chimney or breeching; (*b*) inferior fuel; (*c*) inferior attention; (*d*) accumulation of clinker on grate; and (*e*) boiler too small for the load.

6. *Boiler requires too frequent cleaning of flues.* This may be due to: (*a*) poor draft; (*b*) smoky combustion; (*c*) too low a rate of combustion; and (*d*) too much excess air in firebox causing chilling of gases.

7. *Boiler smokes through fire door.* This may be due to: (*a*) defective draft in chimney or incorrect setting of dampers; (*b*) air leaks into boiler or breeching; (*c*) gas outlet from firebox plugged with fuel; (*d*) dirty or clogged flues; and (*e*) improper reduction in breeching size.

8. *Low carbon dioxide.* This may be due on oil burning boilers to: (*a*) improper adjustment of the burner; (*b*) leakage through the boiler setting; (*c*) improper fire caused by a fouled nozzle; or (*d*) to an insufficient quantity of oil being burned.

Cleaning Steam Boilers

All boilers are provided with flue clean-out openings through which the heating surface can be reached by means of brushes or scrapers. Flues of solid fuel boilers should be cleaned often to keep the surfaces free of soot or ash. Gas boiler flues and burners should be cleaned at least once a year. Oil burning boiler flues should be examined periodically to determine when cleaning is necessary.

The grease used to lubricate the cutting tools during erection of new piping systems serves as a carrier for sand and dirt, with the result that a scum of fine particles and grease accumulates on the surface of the water in all new boilers, while heavier particles may settle to the bottom

of the boiler and form sludge. These impurities have a tendency to cause foaming, preventing the generation of steam and causing an unsteady water line.

This unavoidable accumulation of oil and grease should be removed by blowing off the boiler as follows: If not already provided, install a surface blow connection of at least $1\frac{1}{4}$ in. nominal pipe size with outlet extended to within 18 in. of the floor or to sewer, inserting a valve in line close to boiler. Bring the water line to center of outlet, raise steam pressure and while fire is burning briskly open valve in blow-off line. When pressure recedes, close valve and repeat process adding water at intervals to maintain proper level. As a final operation bring the pressure in the boiler to about 10 lb, close blow-off, draw the fire or stop burner, and open drain valve. After boiler has cooled partly, fill and flush out several times before filling it to proper water level for normal service. The use of soda, or any alkali, vinegar or any acid is not recommended for cleaning heating boilers because of the difficulty of complete removal and the possibility of subsequent injury, after the cleaning process has been completed.

Insoluble compounds have been developed which are effective, but special instructions on the proper cleaning compound and directions for its use in a boiler, as given by the boiler manufacturer, should be carefully followed.

It is common practice when starting new installations to discharge heating returns to the sewer during the first week of operation. This prevents the passage of grease, dirt or other foreign matter into the boiler and consequently may avoid the necessity of cleaning the boiler. During the time the returns are being passed to the sewer, the feed valve should be cracked sufficiently to maintain the proper water level in the boiler.

Care of Idle Heating Boilers

Heating boilers are often seriously damaged during summer months due chiefly to corrosion resulting from the combination of sulphur from the fuel with the moisture in the cellar air. At the end of the heating season the following precautions should be taken:

1. All heating surfaces should be cleaned thoroughly of soot, ash and residue, and the heating surfaces of steel boilers should be given a coating of lubricating oil on the fire side.

2. All machined surfaces should be coated with oil or grease.

3. Connections to the chimney should be cleaned and in case of small boilers the pipe should be placed in a dry place after cleaning.

4. If there is much moisture in the boiler room, it is desirable to drain the boiler to prevent atmospheric condensation on the heating surfaces of the boiler when they are below the dew-point temperature. Due to the hazard of some one inadvertently building a fire in a dry boiler, however, it is safer to keep the boiler filled with water. A hot water system usually is left filled to the expansion tank.

5. The grates and ashpit should be cleaned.

6. Clean and repack the gage glass if necessary.

7. Remove any rust or other deposit from exposed surfaces by scraping with a wire brush or sandpaper. After boiler is thoroughly cleaned, apply a coat of preservative paint where required to external parts normally painted.

8. Inspect all accessories of the boiler carefully to see that they are in good working order. In this connection, oil all door hinges, damper bearings and regulator parts.

BOILER INSULATION

Insulation for cast-iron boilers is of two general types: (1) plastic material or blocks wired on, cemented and covered with canvas or duck; and (2) blocks, sheets or plastic material covered with a metal jacket furnished by the boiler manufacturer. Self-contained steel firebox boilers usually are insulated with blocks, cement and canvas, or rock wool blankets; HRT boilers are brick set and do not require insulation beyond that provided in the setting. It is essential that the insulation on a boiler and adjacent piping be of non-combustible material as even slow-burning insulation constitutes a dangerous fire hazard in case of low water in the boiler.

Chapter 13

RADIATORS AND CONVECTORS

Heat Emission of Radiators and Convectors, Types of Radiators, Output of Radiators, Heating Effect, Heating Up the Radiator and Convector, Enclosed Radiators, Convectors, Selection, Code Tests, Gravity-Indirect Heating Systems

THE accepted terms for heating units are: (1) *radiators*, for direct surface heating units, either exposed, enclosed, or shielded, which emit a large percentage of their heat by radiation; and (2) *convectors*, for heating units having a large percentage of extended fin surface and which emit heat principally by convection. Convectors are dependent upon enclosures to provide the circulation by gravity of large volumes of air.

HEAT EMISSION OF RADIATORS AND CONVECTORS

Most heating units emit heat by *radiation* and *convection*. The resultant heat from these processes depends upon whether or not the heating unit is exposed or enclosed and upon the contour and surface characteristics of the material in the units.

An exposed radiator emits less than half of its heat by radiation, the amount depending upon the size and number of sections. When the radiator is enclosed or shielded, radiation is further reduced. The balance of the emission is by conduction to the air in contact with the heating surface, and the resulting circulation of the air warms by convection.

A convector emits practically all of its heat by conduction to the air surrounding it and this heated air is in turn transmitted by convection to the rooms or spaces to be warmed, the heat emitted by radiation being negligible.

TYPES OF RADIATORS

Present day radiators may be classified as tubular, wall, or window types, and are generally made of cast-iron. There are two types of tubular radiators available, that known as large-tube which has a spacing of 2½ in. per section, and that known as small-tube which has a spacing of 1¾ in. per section. The tubes in the latter type of radiator are materially smaller than those in the large-tube type. Small-tube radiators occupy less space and are particularly suited for installation in recesses.

In 1939, after a complete study of the demand for various sizes of radiators, the *Institute of Boiler and Radiator Manufacturers*, acting for the manufacturers, in cooperation with the Division of Simplified Practice, *National Bureau of Standards*, established Simplified Practice Recom-

TABLE 1. SIZES—LARGE-TUBE CAST-IRON RADIATORS

Sectional, cast-iron, tubular-type radiators of the large-tube pattern, that is, having tubes approximately $1\frac{3}{8}$ in. in diameter

Number of Tubes per Section	Catalog Rating per Section[a]	Section Dimensions[b] A Height	B Width Minimum	B Width Maximum	C Spacing	D Leg Height[c]	Radiator Length (Sections per Radiator)[d]
	Sq Ft	In.	In.	In	In.	In.	
4	$2\frac{1}{2}$	23	$6\frac{1}{4}$	$6\frac{13}{16}$	$2\frac{1}{2}$	$4\frac{1}{2}$	As Required
	$2\frac{3}{4}$	26	$6\frac{1}{4}$	$6\frac{13}{16}$	$2\frac{1}{2}$	$4\frac{1}{2}$	Do
	$3\frac{1}{2}$	32[e]	$6\frac{1}{4}$	$6\frac{13}{16}$	$2\frac{1}{2}$	$4\frac{1}{2}$	Do
	$4\frac{1}{4}$	38[f]	$6\frac{1}{4}$	$6\frac{13}{16}$	$2\frac{1}{2}$	$4\frac{1}{2}$	Do
5	$2\frac{2}{3}$	20	8	$8\frac{9}{16}$	$2\frac{1}{2}$[h]	$4\frac{1}{2}$	Do
	3	23	8	$8\frac{9}{16}$	$2\frac{1}{2}$[h]	$4\frac{1}{2}$	Do
	$3\frac{1}{2}$	26	8	$8\frac{9}{16}$	$2\frac{1}{2}$[h]	$4\frac{1}{2}$	Do
	$4\frac{1}{3}$	32[e]	8	$8\frac{9}{16}$	$2\frac{1}{2}$[h]	$4\frac{1}{2}$	Do
	5	38[f]	8	$8\frac{9}{16}$	$2\frac{1}{2}$[h]	$4\frac{1}{2}$	Do
6	4	26	9	$10\frac{3}{8}$	$2\frac{1}{2}$	$4\frac{1}{2}$	Do
	6	38[f]	9	$10\frac{3}{8}$	$2\frac{1}{2}$	$4\frac{1}{2}$	Do
7	$2\frac{1}{2}$	14[g]	$11\frac{3}{8}$	$12\frac{3}{16}$	$2\frac{1}{2}$	3	Do
	$3\frac{2}{3}$	20	$11\frac{3}{8}$	$12\frac{3}{16}$	$2\frac{1}{2}$	3 or $4\frac{1}{2}$	Do

[a]The square foot of equivalent direct steam radiation is defined as the ability to emit 240 Btu per hour, with steam at 215 F in air at 70 F. These ratings apply only to radiators installed exposed in a normal manner; not to radiators installed behind enclosures, grilles, etc. (See A.S.H.V.E. Code for Testing Radiators.)

[b]See Fig. 1.

[c]Where greater than standard leg heights are required, this dimension shall be 6 in., except for 7-tube sections, in heights from 13 to 20 in., incl. for which this dimension shall be $4\frac{1}{2}$ in. Radiators may be furnished without legs.

[d]Maximum assembly 60 sections.

[e]Alternate height by 1 producer is 30 in.

[f]Alternate height by 2 producers is 36 in.; by another, 37 in.

[g]Alternate height by 1 producer is 13 in ; by 2 producers $13\frac{1}{2}$ in.; by another, 15 in.

[h]For 5-tube hospital-type radiation, this dimension is 3 in.

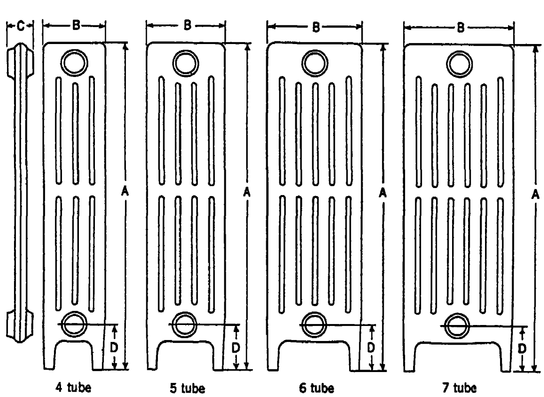


FIG. 1. TYPES AND DIMENSIONS OF LARGE-TUBE CAST-IRON RADIATORS

mendation R174-40 for large-tube cast-iron radiators. A program is now in progress for the establishment of a similar recommendation for small-tube cast-iron radiators. Tables 1 and 2, and Figs. 1 and 2 show the sizes and dimensions of large-tube and small-tube cast-iron radiators which are being manufactured at the present time.

Pipe coils are assemblies of standard pipe or tubing (1 in. to 2 in.) which

TABLE 2. SIZES—SMALL-TUBE CAST-IRON RADIATORS

Number of Tubes per Section	Catalog Rating per Section[a]	Section Dimensions[b]					Radiator Length (Sections per Radiator)[d]
		A Height[c]	B Width Minimum	B Width Maximum	C Spacing	D Leg Height[c]	
	Sq Ft	In	In.	In.	In.	In.	
3[e]	1.6	25	3¼	3½	1¾	2½	As Required
4[e]	1.6	19	4 7/16	4 13/16	1¾	2½	Do
	1.8	22	4 7/16	4 13/16	1¾	2½	Do
	2.0	25	4 7/16	4 13/16	1¾	2½	Do
5[e]	2.1	22	5⅝	6 5/16	1¾	2½	Do
	2.4	25	5⅝	6 5/16	1¾	2½	Do
	3.0	32	5⅝	6 5/16	1¾	2½	Do
6[e]	1.6	14	6 13/16	8	1¾	2½	Do
	2.3	19	6 13/16	8	1¾	2½	Do
	3.0	25	6 13/16	8	1¾	2½	Do
	3.7	32	6 13/16	8	1¾	2½	Do

[a]The square foot of equivalent direct steam radiation is defined as the ability to emit 240 Btu per hour, with steam at 215 F, in air of 70 F. These ratings apply only to radiators installed exposed in a normal manner; not to radiators installed behind enclosures, grilles, etc. (See A.S.H.V.E. Code for Testing Radiators.)

[b]See Fig. 2.

[c]Overall height and leg height, as produced by some manufacturers is one inch (1 in.) greater than shown in columns A and D. Radiators may be furnished without legs.

[d]Even number of sections. Maximum assembly 60 sections.

[e]Or equal.

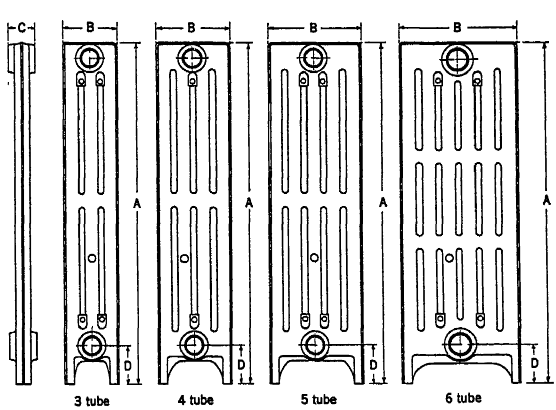


FIG. 2. TYPES AND DIMENSIONS OF SMALL-TUBE CAST-IRON RADIATORS

are used as radiators. In older practice these coils were commonly used in factory buildings, but now wall type radiators are most frequently used for this service. When coils are used, the miter type assembly is to be preferred as it best cares for expansion in the pipe. Cast manifolds or headers, known as branch tees, are available for this construction.

OUTPUT OF RADIATORS

The output of a radiator can be measured only by the heat it emits. The old standard of comparison used to be square feet of *actual surface*, but since the advance in radiator design and proportions, the surface area alone is not a true index of output. (The engineering unit of output is the *Mbh* or 1000 Btu per hour.) However, during the period of transition from the old to the new, radiators may be referred to in terms of *equivalent square feet*. For steam service this is based on an emission of 240 Btu per hour per square foot and for hot water service 150 Btu per hour per square foot.

Wall radiators are now rated in terms of equivalent square feet, the same as large-tube and small-tube radiators. Tests have shown that the heat emitted from a wall-type radiator may be reduced from 5 to 10 per cent if the radiator is placed near the ceiling with the bars horizontal and in an air temperature exceeding 70 F. When radiators are placed near the ceiling, there is usually such a large difference in the temperature between the floor level and the ceiling that it becomes difficult to heat the living zone of the room satisfactorily.

The heat emission of pipe coils placed vertically on a wall with the pipes horizontal is given in Table 3. This has been developed from available data and does not represent definite results of tests. For such coils the heat emission varies as the height of the coil. The heat emission of each pipe of ceiling coils, placed horizontally, is about 126 Btu, 156 Btu, and 175 Btu per linear foot of pipe, respectively, for 1-in., 1¼-in., and 1½-in. coils.

Effect of Paint

The prime coat of paint on a radiator has no material effect on the heat output, but the finishing coat may influence the radiation emission and thus affect the heat output. Within the range of temperatures at which

TABLE 3. HEAT EMISSION OF PIPE COILS PLACED VERTICALLY ON A WALL (PIPES HORIZONTAL) CONTAINING STEAM AT 215 F AND SURROUNDED WITH AIR AT 70 F

Btu per linear foot of coil per hour (not linear feet of pipe)

SIZE OF PIPE	1 IN.	1¼ IN.	1½ IN.
Single row	132	162	185
Two	252	312	348
Four	440	545	616
Six	567	702	793
Eight	651	796	907
Ten	732	907	1020
Twelve	812	1005	1135

TABLE 4. EFFECT OF PAINTING 32-IN. THREE COLUMN, SIX-SECTION CAST-IRON RADIATOR[a]

Radiator No.	Finish	Area Sq Ft	Coefficient of Heat Trans. Btu	Relative Heating Value Per Cent
1	Bare iron, foundry finish	27	1.77	100.5
2	One coat of aluminum bronze	27	1.60	90.8
3	Gray paint dipped	27	1.78	101.1
4	One coat dull black heat resistant paint	27	1.76	100.0

[a]Comparative Tests of Radiator Finishes, by W. H. Severns (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 41).

radiators operate, color has no appreciable influence on the radiation emitted. Thus, finishing coats of oil paints of various colors will give the same results. However, a bronze paint, applied as the finish coat will change the character of the surface and reduce the amount of heat emitted by radiation. No paint has a noticeable effect on the portion of heat which is given off by convection. The larger the proportion of direct radiating surface, the greater will be the effect of any finish coat of paint which changes the character of the surface. Available tests are on old-style column type radiators which give results as shown in Table 4.

Effect of Superheated Steam

Available research data indicate that there is probably a decrease in heat transfer rate for a radiator or gravity convector with superheated steam in comparison with saturated steam at the same temperature. The decrease is probably small for low temperatures of superheats and additional tests are necessary with varying degrees of superheat to establish accurate comparisons for all types of radiators and convectors[1].

HEATING EFFECT

For several years the *heating effect* of radiators has been considered by engineers in order to use it for the rating of radiators and in the design of heating systems. Heating effect is the *useful output* of a radiator, in the comfort zone of a room, as related to the total input of the radiator[2].

No standard method for evaluating the heating effect of radiators and convectors and correlating it with comfort has yet been accepted. One method, with test data[3] on radiators and convectors, and making use of the eupatheoscope for evaluating the environment produced has been suggested by the University of Illinois. The principle underlying the eupatheoscope involves the measurement of the heat loss from a sizable body by radiation and convection, when the surface is maintained at some constant temperature. Through the use of this instrument and its calibration curve, non-uniform environments may be referred to uniform environments in which the air and all surrounding surfaces are at the

[1]Tests of Radiators with Superheated Steam, by R. C. Carpenter (A.S.H.V.E. TRANSACTIONS, Vol. 7, 1901, p. 206).

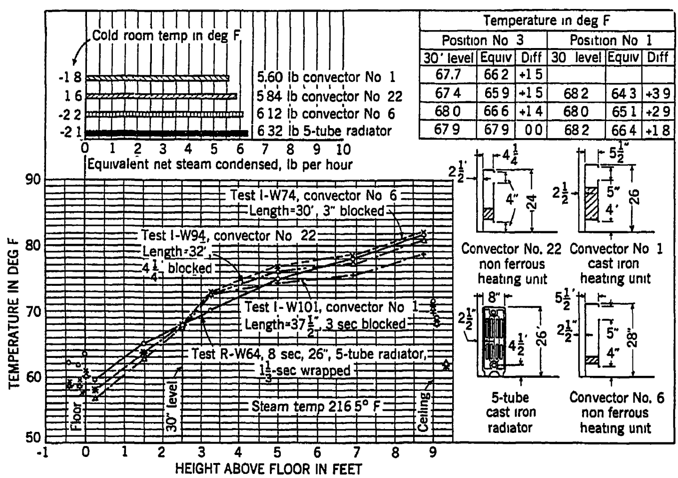
[2]The Heating Effect of Radiators, by Dr. Charles Brabbeé (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 33).

[3]A.S.H.V.E. Research Report No. 962—The Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperature, by A. C. Willard, A. P. Kratz and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 303).

same temperature. The temperatures of the uniform environments are referred to as equivalent temperatures.

Data given in Fig. 3 shows that while the air temperature at the 30 in. level is the same for the three convectors and the one large-tube cast-iron radiator, in position No. 3 in the test room, the equivalent temperature is 1.5 F lower than the air temperature in the case of the three convectors, and the same as the air temperature in the case of the radiator. The difference between the minimum and the maximum amount of heat required to maintain the common air temperature at the 30 in. level is of the order of 13 per cent.

In Fig. 4 are shown the results of tests made with the same three convectors and the one large-tube cast-iron radiator, so adjusted in size that each gave approximately the same equivalent temperature in the



Temperature in deg F					
Position No 3			Position No 1		
30' level	Equiv	Diff	30 level	Equiv	Diff
67.7	66 2	+1 5			
67 4	65 9	+1 5	68 2	64 3	+3 9
68 0	66 6	+1 4	68 0	65 1	+2 9
67 9	67 9	0 0	68 2	66 4	+1 8

FIG. 3. ROOM TEMPERATURE GRADIENTS AND EQUIVALENT TEMPERATURES FOR A DIRECT C. I. RADIATOR AND THREE CONVECTORS WITH A COMMON 30-IN. LEVEL TEMPERATURE

No. 3 position in the test room. The difference between the miminum and the maximum amount of heat required to maintain the common equivalent temperature is of the order of 7 per cent. The following conclusions are given from the results of these tests:

1. The eupatheoscope affords a means of evaluating the combined effect of radiation and convection in a given environment in terms of a standard environment and in some terms related to human comfort.

2. Future investigation should be undertaken to correlate the readings of the eupatheoscope with a more definite standard of comfort based on a consideration of all of the physical and physiological aspects of the problem.

3. The readings of the eupatheoscope serve as a means of ranking various heating units in the order of the relative heating effects produced.

4. The steam condensation obtained when the equivalent temperature is maintained at 68 F at the 30 in. level is a measure of the effectiveness of different types of heating units in providing for human comfort; lower condensations corresponding to greater effectiveness.

5. The proximity of cold walls has an appreciable effect on the degree of comfort as determined by the eupatheoscope.

The Kata thermometer[4], the thermo-integrator[5], [6], and the globe[7] thermometer are other instruments which have been used to measure the influence of air temperature, air movement and radiation in an environment.

The following statements applying to the use of radiators are based on experience and test results:

1. The heating effect of a radiator cannot be judged solely by the amount of steam condensed within the radiator.

2. Smaller floor-to-ceiling temperature differentials can be maintained with long, low, thin, direct radiators, than is possible with high, direct radiators.

3. The larger portion of the floor-to-ceiling temperature differential in a room of average ceiling height heated with direct radiators occurs between the floor and the breathing level.

4. The comfort level (approximately 2 ft-6 in. above floor) is below the breathing line level (approximately 5 ft-0 in. above floor), and temperatures taken at the breathing

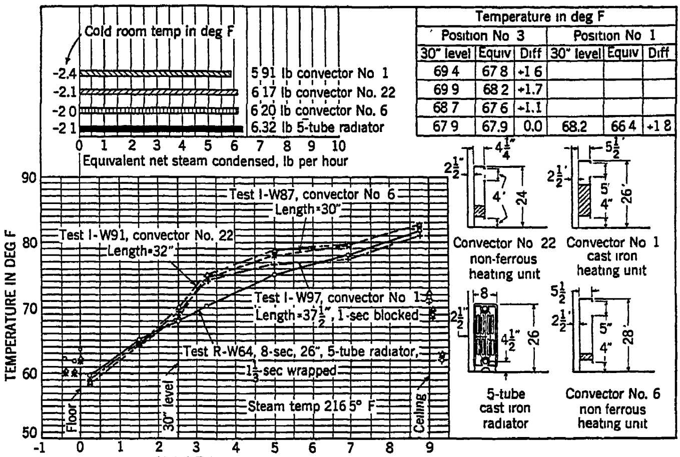


FIG. 4. ROOM TEMPERATURE GRADIENTS AND EQUIVALENT TEMPERATURES FOR A DIRECT C. I. RADIATOR AND THREE CONVECTORS WITH A COMMON EQUIVALENT TEMPERATURE

line may not be indicative of the actual heating effect of a radiator in the room. The comfort-indicating temperature should be taken below the breathing line level.

5. High column radiators placed at the sides of window openings do not produce as comfortable heating effects as long, low, direct radiators placed beneath window openings.

HEATING UP THE RADIATOR AND CONVECTOR

The maximum condensation occurs in a heating unit when the steam is first turned on[8]. Fig. 5 shows a typical curve for the condensation rate

[4]The Kata Thermometer Its Value and Defects, by W. J. McConnell and C. P. Yagloglou. (Reprint No. 953 from *U. S. Public Health Service Report*, pp. 2293-2337, September 5, 1924).

[5]The Thermo-Integrator—A New Instrument for the Observation of Thermal Interchanges, by C.-E. A. Winslow and Leonard Greenburg (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 149).

[6]The Calibration of the Thermo-Integrator, by C.-E. A. Winslow, A. P. Gagge, L. Greenburg, I. M. Moriyama and E. J. Rodee. (*The American Journal of Hygiene*, Vol. 22, No. 1, July, 1935, pp. 137-156.)

[7]The Globe Thermometer in Studies of Heating and Ventilation, by T. Bedford and C. G. Warner. (*The Journal of Hygiene*, Vol. 34, No. 4.)

[8]A.S.H.V.E. RESEARCH REPORT No. 1067—The Cooling and Heating Rates of a Room with Different Types of Steam Radiators and Convectors, by A. P. Kratz, M. K. Fahnestock and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 389).

in pounds per hour for the time elapsing after steam is turned into a cast-iron radiator. The data are from tests on old-style column type radiators. In practice the rate of steam supply to the heating unit while heating up is frequently retarded by controlled elimination of air through air valves or traps. Automatic control valves may also retard the supply of steam. Vacuum types of air venting valves may be used to reduce the length of the venting periods.

ENCLOSED RADIATORS

The general effect of an enclosure placed about a direct radiator is to restrict the air flow, diminish the radiation and, when properly designed, improve the heating effect. Investigations[9] indicate that in the design of the enclosure three things should be considered:

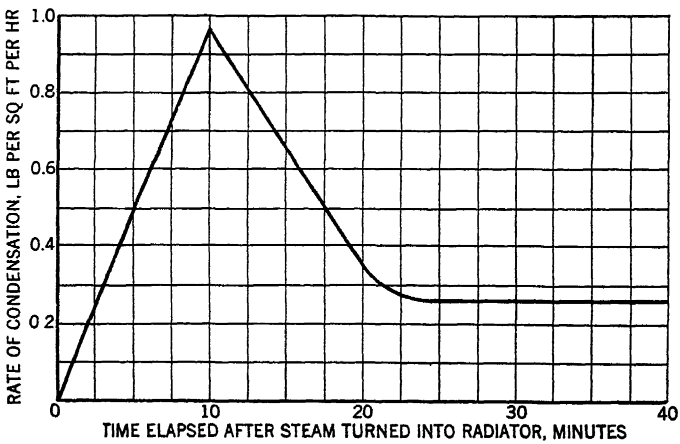


FIG. 5. CHART SHOWING THE STEAM DEMAND RATE FOR HEATING UP A CAST-IRON RADIATOR WITH FREE AIR VENTING AND AMPLE STEAM SUPPLY

1. There should be better distribution of the heat below the breathing line level to produce greater heating comfort and lowered ceiling temperatures.
2. The lessened steam consumption may not materially change the radiator heating performance.
3. The enclosed radiator may inadequately heat the space.

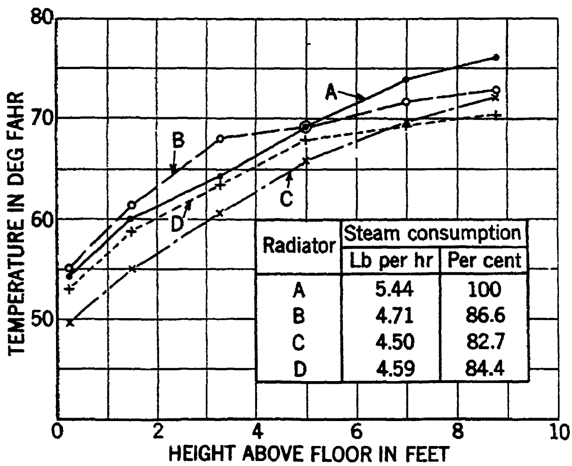
A comparison between a bare or exposed radiator (*A*) and the same radiator with a well-designed enclosure (*B*), with a poorly-designed enclosure (*C*), and with a cloth cover (*D*) will illustrate the relative heating effects. In Fig. 6 the curve (*B*) reveals that the enclosed radiator used less steam than the exposed radiator, but gave a satisfactory heating performance. A well-designed shield placed over a radiator gives about the same heating effect. Curve (*C*) shows the unsatisfactory effects produced by improperly designed enclosures. Curve (*D*) shows that the effect of a cloth cover extending downward 6 in. from the top of the radiator was to make the performance unsatisfactory and inadequate.

[9]University of Illinois, *Engineering Experiment Station Bulletins* Nos. 192 and 223, Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, by A. C. Willard, A. P. Kratz, M. K. Fahnestock and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 77).

Some commercial enclosures and shields for use on direct radiators are equipped with water pans for the purpose of adding moisture to the air in the room. Tests[10] show that an average evaporative rate of about 0.235 lb per square foot of water surface per hour may be obtained from such pans, when the radiator is steam hot and the relative humidity in the room is between 25 and 40 per cent. This source of supply of moisture alone is not adequate to maintain a relative humidity above 25 per cent on a zero day.

CONVECTORS

Although any standard radiator may be concealed in a cabinet or other enclosure so that the greater percentage of heat is conveyed to the room by convection thereby resulting in a form of gravity convector,



Radiator	Steam consumption	
	Lb per hr	Per cent
A	5.44	100
B	4.71	86.6
C	4.50	82.7
D	4.59	84.4

Fig. 6. Steam Consumption of Exposed and Concealed Radiators

generally better results are obtained with specially designed units which permit a free circulation of a larger volume of air at moderate temperatures. Since air stratifies according to temperature, moderate delivery temperatures at the outlet of the enclosure reduce the temperature differential between the floor and ceiling and accordingly accomplish the desired heating effect in the living zone.

A typical recessed convector is shown in Fig. 7. The heating element consisting of a large percentage of fin surface is usually shallow in depth and placed low in the enclosure in order to produce maximum chimney effect in the enclosure. The air enters the enclosure near the floor line just below the heating element, is moderately heated in passing through the core and delivered to the room through an opening near the top of enclosure. Since the air can only enter the enclosure at the floor line, the cooler air in the room which always lies at this level, is constantly being withdrawn and replaced by the warmer air. This air movement

[10] University of Illinois, *Engineering Experiment Station Bulletin* No. 230, p. 20.

accomplishes the desired reduction in temperature differentials and assures maximum comfort in the living zone.

The *Convector Manufacturers Association* has adopted the A.S.H.V.E. Standard[11] in the formulation of its ratings and has compiled a tentative standard of heating effect allowances for various enclosure heights to be included in the ratings by its members.

All published ratings bearing the title *C.M.C.* Ratings (*Convector Manufacturers Certified Ratings*) indicate that the convectors have been tested in accordance with the A.S.H.V.E. Code by an impartial and disinterested laboratory and that the ratings have been approved by the Standardization Committee of the *Convector Manufacturers Association.*

Concealed heaters or convectors are generally sold as completely built-in units. The enclosing cabinet should be designed with suitable

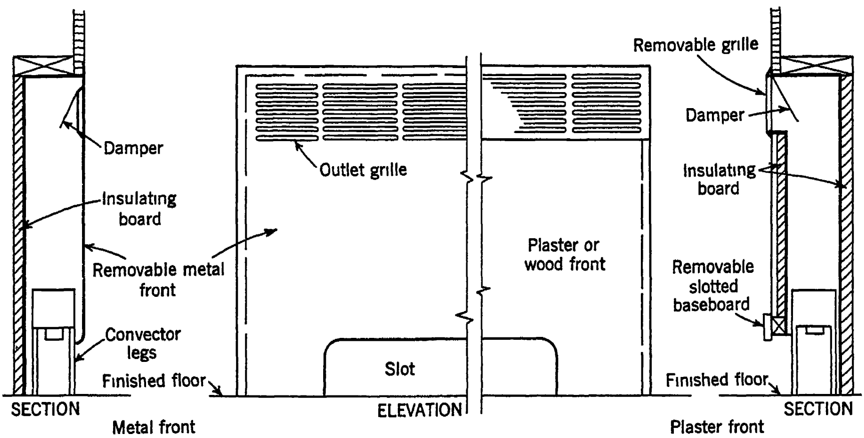


FIG. 7. TYPICAL RECESSED CONVECTOR

air inlet and outlet grilles to give the heating element its best performance. Tables of capacities are catalogued for various lengths, depths and heights, and combinations are available in several styles for installations, such as the wall-hung type, free-standing floor type, recess type set flush with wall or offset, and the completely concealed type. Most of these types may be arranged with a top outlet grille in a plane parallel with the floor, although the front outlet is practically standard. In cases where enclosures are to be used but are not furnished by the heater manufacturer, it is important that the proportions of the cabinet and the grilles be so designed that they will not impair the performance of the assembled convector. It is important that the enclosure or housing for the convector fit as snugly as possible so that the air to be heated must pass through the convector and cannot be by-passed in the enclosure.

The output of a convector, for any given length and depth, is a variable

[11]A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam), (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 367); (Hot Water), (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 237). (See also A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 38, and Vol. 42, 1936, p. 29).

of the height. Published ratings are generally given in terms of equivalent square feet, corrected for heating effect. However, an extended surface heating unit is entirely different structurally and physically from a direct radiator and, since it has no area measurement corresponding to the heating surface of a radiator, many engineers believe that the performance of convectors should be stated in Btu. For steam convectors, as for radiators, 240 Btu per hour may be taken as an equivalent square foot of radiation. When more than one heating unit is used, one mounted above the other in the same cabinet, the output of the upper unit or units will be materially less than that of the bottom unit.

RADIATOR AND CONVECTOR SELECTION

The capacity of a radiator varies as the 1.3 power, and that of a convector[12] as the 1.5 power of the temperature difference between the heating medium and the surrounding air in the case of the radiators, and the entering air in the case of the convector. It is obvious that for conditions other than the basic ones with the heating medium at a temperature of 215 F, and the room temperature at 70 F in the case of a radiator, and the inlet air temperature at 65 F in the case of a convector, the heat emission will be other than 240 Btu per square foot of rating.

Table 5 shows factors by which radiation requirements, as determined by dividing heat load by 240, shall be multiplied to obtain proper radiator or convector sizes from published rating tables for room temperatures ranging between 50 and 80 F as well as for steam or water temperatures from 150 to 300 F. For other room and heating medium temperatures the factor is determined by the following formulae:

For radiators: For convectors:

$$C_s = \left(\frac{215 - 70}{t_s - t_r}\right)^{1.3} \qquad C_s = \left(\frac{215 - 65}{t_s - t_i}\right)^{1.5} \tag{1}$$

where

C_s = correction factor.

t_s = steam temperature, degrees Fahrenheit.

t_r = room temperature, degrees Fahrenheit.

t_i = average inlet air temperature, degrees Fahrenheit.

As previously indicated, the output of radiators and convectors is still designated by the terms of older practice, but this is gradually giving place to an engineering method of designating heat emission. The A.S.H.V.E. has adopted the following standards: Code for Testing Radiators (1927); Codes for Testing and Rating Concealed Gravity Type Radiation (Steam), 1931, and (Hot Water), 1933; (see also A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 38 and Vol. 42, 1936, p. 29).

For steam services the actual condensation weight is taken without any allowance for heating effect; for hot water services the weight of circulated water is used without allowance for heating effect. In all cases the total heat transmission varies as the 1.3 power for radiators[13] and the 1.5 power for convectors[14] of the temperature difference between that inside

[12] A.S.H.V.E. RESEARCH REPORT No. 998—Factors Affecting the Heat Output of Convectors, by A. P. Kratz, M. K. Fahnestock, and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 443).

[13] Loc. Cit. Note 11.

[14] Loc. Cit. Notes 11 and 12.

the radiator and the air in the room, and is expressed in Btu or *Mb* per hour.

Standard test conditions specify either a steam pressure of 1 lb gage 15.6 lb per square inch absolute (215 F) or an average hot water temperature of 170 F and a room temperature of 70 F (5 ft above floor) for radiators, or an inlet air temperature of 65 F for convectors. The heating capacity of a *steam radiator* or *steam convector* is determined as follows:

$$H_t = W_s h_{fg} \tag{2}$$

where

H_t = Btu per hour under test conditions.
W_s = condensation in pounds per hour.
h_{fg} = latent heat in Btu per pound.

TABLE 5. CORRECTION FACTORS FOR DIRECT CAST-IRON RADIATORS AND CONVECTORS[a]

Steam Press. Approx. Gage Vacuum In. Hg.	Steam Press. Approx. Abs. Lb per Sq In.	Heating Medium Temp. F Steam or Water	Factors for Direct Cast-Iron Radiators — Room Temperature F: 80	75	70	65	60	55	50	Factors for Convectors — Inlet Air Temperature F: 80	75	70	65	60	55	50
22.4	3.7	150	2.58	2.36	2.17	2.00	1.86	1.73	1.62	3.14	2.83	2.57	2.35	2.15	1.98	1.84
20.3	4.7	160	2.17	2.00	1.86	1.73	1.62	1.52	1.44	2.57	2.35	2.15	1.98	1.84	1.71	1.59
17.7	6.0	170	1.86	1.73	1.62	1.52	1.44	1.35	1.28	2.15	1.98	1.84	1.71	1.59	1.49	1.40
14.6	7.5	180	1.62	1.52	1.44	1.35	1.28	1.21	1.15	1.84	1.71	1.59	1.49	1.40	1.32	1.24
10.9	9.3	190	1.44	1.35	1.28	1.21	1.15	1.10	1.05	1.59	1.49	1.40	1.32	1.24	1.17	1.11
6.5	11.5	200	1.28	1.21	1.15	1.10	1.05	1.00	0.96	1.40	1.32	1.24	1.17	1.11	1.05	1.00
LbperSqIn.																
1	15.6	215	1.10	1.05	1.00	0.96	0.92	0.88	0.85	1.17	1.11	1.05	1.00	0.95	0.91	0.87
6	21	230	0.96	0.92	0.88	0.85	0.81	0.78	0.76	1.00	0.95	0.91	0.87	0.83	0.79	0.76
15	30	250	0.81	0.78	0.76	0.73	0.70	0.68	0.66	0.83	0.79	0.76	0.73	0.70	0.68	0.65
27	42	270	0.70	0.68	0.66	0.64	0.62	0.60	0.58	0.70	0.68	0.65	0.63	0.60	0.58	0.56
52	67	300	0.58	0.57	0.55	0.53	0.52	0.51	0.49	0.56	0.54	0.53	0.51	0.49	0.48	0.47

[a]To determine the size of a radiator or a convector for a given space, divide the heat loss in Btu per hour by 240 and multiply the result by the proper factor from the above table.

To determine the heating capacity of a radiator or a convector under conditions other than the basic ones with the heating medium at a temperature of 215 F, and the room temperature at 70 F in the case of a radiator, and the inlet air temperature at 65 F in the case of a convector, divide the heating capacities at the basic conditions by the proper factor from the above table.

H_t may be converted to standard conditions of code ratings by using the proper correction factor from the following formulae:

For radiators: For convectors:

$$C_s = \left(\frac{215 - 70}{T_s - T_r}\right)^{1.3} \qquad C_s = \left(\frac{215 - 65}{T_s - T_i}\right)^{1.5} \tag{3}$$

The output under standard conditions will be:

$$H_s = C_s H_t \tag{4}$$

where

C_s = correction factor.
T_s = steam temperature during test, degrees Fahrenheit.
T_r = room temperature during test, degrees Fahrenheit.
T_i = inlet air temperature during test, degrees Fahrenheit.
H_s = heat emission rating under standard conditions, Btu per hour.

Similarly, for *hot water convectors*, the output under test conditions may be determined as follows:

$$H = W(\theta_1 - \theta_2)\frac{3600}{t} \tag{5}$$

where

H = Btu per hour under test conditions.
W = pounds of water handled during test.
θ_1 = average temperature of inlet water, degrees Fahrenheit.
θ_2 = average temperature of outlet water, degrees Fahrenheit.
t = duration of test, seconds.

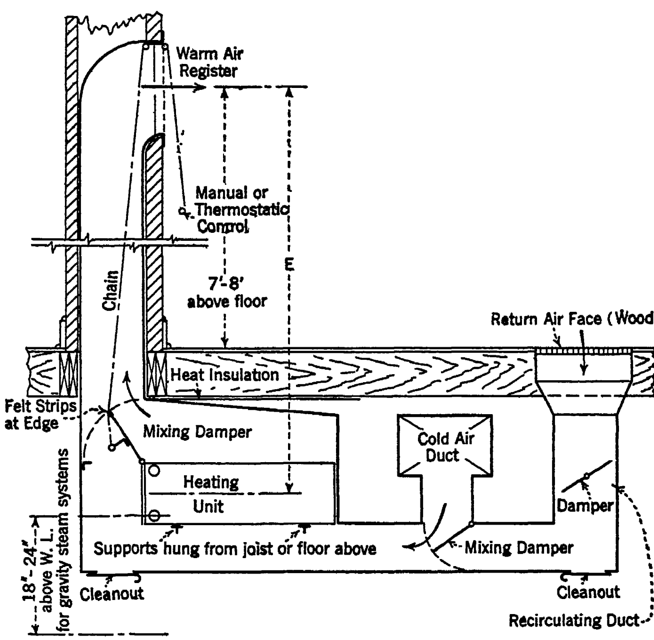


FIG. 8. GRAVITY-INDIRECT HEATING SYSTEM

To convert test results to standard conditions, the following correction factor is used:

$$C = \left(\frac{170 - 65}{\frac{\theta_1 + \theta_2}{2} - T_i}\right)^{1.5} = \left(\frac{105}{\frac{\theta_1 + \theta_2}{2} - T_i}\right)^{1.5} \tag{6}$$

It has been shown that when the exponent 1.5 is used the range of error is less than 3 per cent[15] for convectors.

[15]Loc. Cit. Note 12.

GRAVITY-INDIRECT HEATING SYSTEMS

The heating units for this system are usually of the extended surface type for steam or hot water, and are installed about as shown in Fig. 8. The temperature and volume of the air leaving the register must be great enough so that in cooling to room temperature the heat available will just equal the heat loss during the same time. To establish and maintain a constant heat flow, provision must be made for removing the air in the room, after it has cooled to the desired room temperature, by a system of vent flues or ducts. As the air flow is maintained by natural draft and this gravity head is very slight, it is necessary to make all ducts as short as possible, especially the runs from the heating units to the base of the vertical warm air flues. Gravity-indirect arrangements, such as illustrated in Fig. 8, are not to be generally recommended for hot water systems unless the water temperature can be maintained at a reasonably high temperature and rapid circulation of the water can be obtained.

Chapter 14

STEAM HEATING SYSTEMS

Gravity and Mechanical Return, Gravity One-Pipe Air-Vent, Gravity Two-Pipe Air-Vent, Air Line Heating, One-Pipe Vapor, Two-Pipe Vapor, Atmospheric, Condensation Return, Vacuum, Sub-Atmospheric, Orifice, Zone Control, Condensation Return Pumps, Vacuum Heating Pumps, Traps

STEAM heating systems may be classified according to the pipe arrangement, the accessories used, the method of returning the condensate to the boiler, the method of expelling air from the system, or the type of control employed. Information concerning the design and layout of steam heating systems will be found in Chapter 15.

GRAVITY AND MECHANICAL RETURN

Systems are classified as gravity or mechanical according to the method of returning the condensate from the system to the boiler. In *gravity systems* the condensate is returned by gravity due to the static head of water in the return pipes or mains. The elevation of the boiler water line must be sufficiently below the lowest heating unit, steam pipe or dry return pipe to permit the return by gravity. The *water line difference* forming the *static head* must be sufficient to overcome the maximum pressure drop in the system, including the pressure drop due to the condensing effect of the radiation. When radiator and drip traps are used, as in two-pipe vapor systems, the static pressure must also exceed the operating pressure of the boiler. The pressure drop caused by condensing rate of the radiation is especially important during those portions of the operating periods where changing pressure conditions prevail, as for example, when the system is being initially filled with steam. In systems where the condensate is wasted to the sewer, no water line difference is required as is the case with closed systems. However, the waste of condensate may introduce conditions which warrant the use of an appropriate mechanical system. Whenever the conditions of a heating system are such that the returns from the radiation cannot gravitate to the boiler, they must be returned by some mechanical means.

In *mechanical systems* the condensate flows to a receiver by gravity and is then forced into the boiler against its pressure. In all instances the preferable practice is to provide for gravity flow even where a vacuum pump is used. The lowest parts of the supply side of the system must be kept sufficiently above the water line of the receiver to insure adequate drainage of water from the system.

There are three general types of mechanical return devices in common use, namely, (1) the mechanical return trap, (2) the condensation return pump, and (3) the vacuum return line pump.

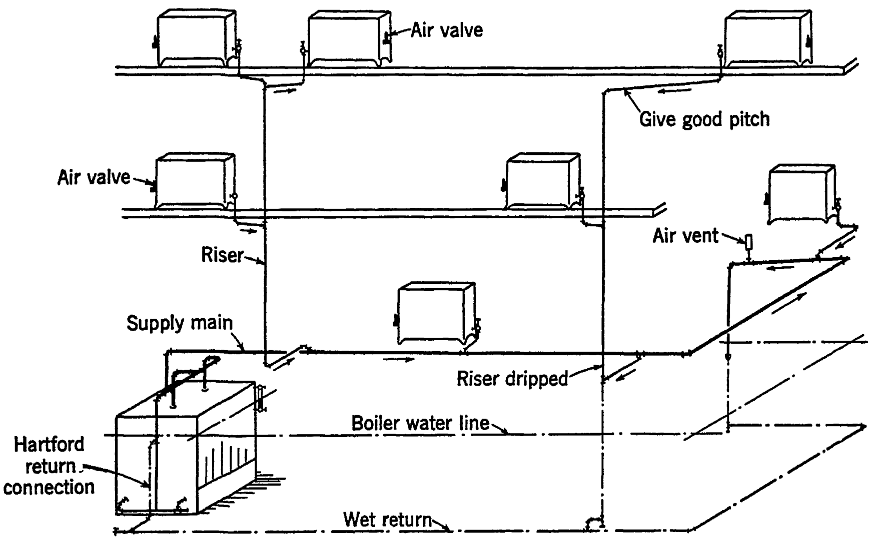


FIG. 1. TYPICAL UP-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

GRAVITY ONE-PIPE AIR-VENT SYSTEM

This system is the most common of all methods of steam heating, especially for small size installations, due largely to its low cost and simplicity.

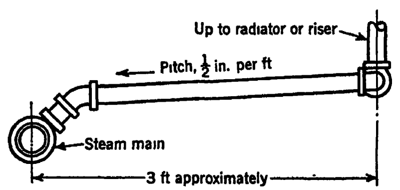


FIG. 2. TYPICAL STEAM RUNOUT WHERE RISERS ARE NOT DRIPPED

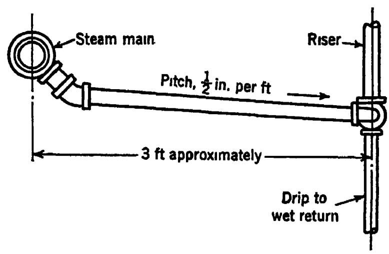


FIG. 3. TYPICAL STEAM RUNOUT WHERE RISERS ARE DRIPPED

The downward pitch of a one-pipe air-vent system is indicated in Fig. 1. Low points and ends of steam mains pitched down from the boiler should be dripped. All drips should be sealed below water line before connecting together. In the risers and radiator connections, steam and

condensation flow in opposite directions. In long steam mains it flows in the same direction as the steam and is removed from the main through the drip. Short mains may be arranged for the condensate to flow in a direction opposite the steam by sizing them so the critical velocity is not exceeded. It is customary to drip the heel of each riser in buildings of several stories to avoid counter-flow of the steam and condensate in the riser branch. In buildings of one or two stories the condensate is returned to the steam main instead of being dripped. Both types of risers are shown in Fig. 1, and riser connections are shown in Figs. 2 and 3. A typical overhead down-feed system is illustrated in Fig. 4. While wet return mains need not be pitched toward the boiler to maintain steam circulation, they should be pitched for drainage.

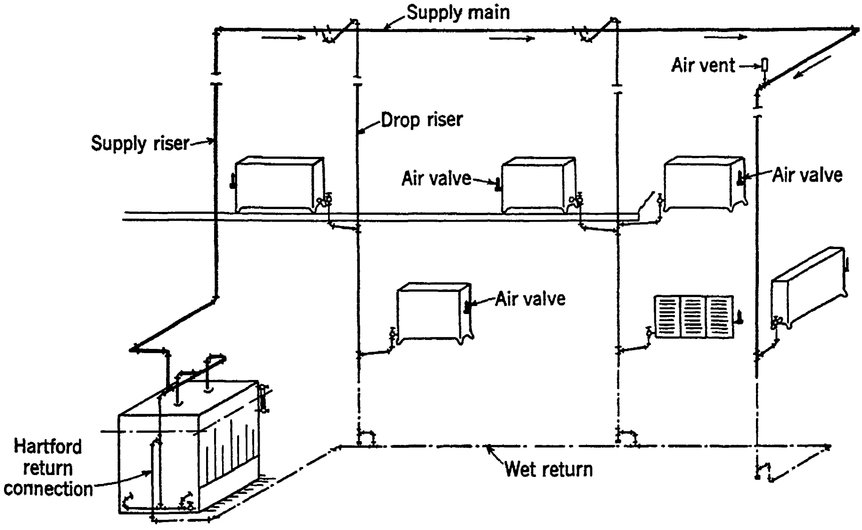


Fig. 4. Typical Down-Feed Gravity One-Pipe Air-Vent System

To improve steam circulation in one-pipe systems quick vent air valves should be provided at the ends and at intermediate points where the steam main is brought to a higher elevation. It is desirable to install the air-vent valves about a foot ahead of the drips, as indicated in Fig. 1, to prevent possible damage to their mechanisms by water.

The radiator valves may be the angle-globe, offset-corner pattern or gate type. Straight-globe and straight-corner type should not be used since the damming effect of the raised valve seat would interfere with the flow of condensation through the valve. Graduated valves cannot be used since the steam valves on this system must be fully open or fully closed to prevent the radiators filling with water and creating a dangerous water line condition. With a one-pipe system the heat cannot be modulated at the radiator, the steam being either *all on* or *all off*. Systems and devices are available which make it possible to obtain a partial modulating effect from one-pipe heating systems.

It is important to keep the lowest points of the steam mains and heating units sufficiently above the water line of the boiler to prevent flooding. The minimum water line difference depends on the initial steam pressure and piping pressure drop plus a safety factor for heating up.

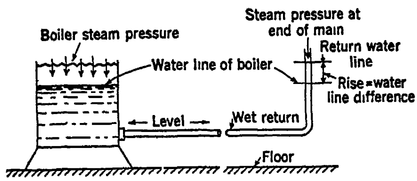


FIG. 5. DIFFERENCE IN STEAM PRESSURE ON WATER IN BOILER AND AT END OF STEAM MAIN

Referring to Fig. 5 it will be noted that the water in the wet return is a U-shaped container, with the boiler steam pressure on the top of the water at one end and the steam main pressure on the top of the water at the other end. The difference between these two pressures is the *pressure drop* in the system, *i.e.*, the friction and resistance to the flow of steam in passing from the boiler to the far end of the main and the pressure reduction in consequence of the condensation occurring in the system. The water in the far end will

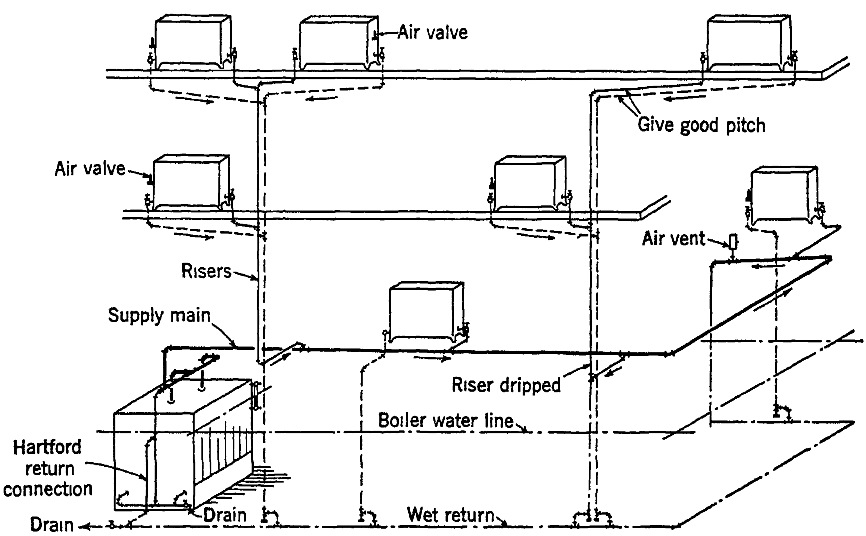


FIG. 6. TYPICAL UP-FEED GRAVITY TWO-PIPE AIR-VENT SYSTEM

rise sufficiently to overcome this difference in order to balance the pressures, and it will rise far enough to produce a flow through the return pipe and overcome the resistance of check valves if installed.

If a one-pipe steam system is designed, for example, for a total pressure drop of $\frac{1}{8}$ lb, and utilizes an Underwriters' Loop instead of a check valve on the return, the rise in the water level at the far end of the return due to the difference in steam pressure would be $\frac{1}{8}$ of 28 in. (28 in. head being equal to one pound per square inch), or $3\frac{1}{2}$ in. Adding 3 in. to overcome the resistance of the return main and 6 in. as a factor of safety for heating up gives $12\frac{1}{2}$ in. as the distance; the bottom of the lowest part of the steam main and all heating units must be above the boiler water line. The same system, however, installed and sized for a total pressure drop of $\frac{1}{2}$ lb, and with a check in the return,

would require ½ of 28 in., or 14 in. for the difference in steam pressure, 3 in. for the flow through the return, 4 in. to operate the check, and 6 in. for a factor of safety, making a total of 27 in. as the required distance. Higher pressure drops would increase the distance accordingly.

GRAVITY TWO-PIPE AIR-VENT SYSTEMS

The gravity two-pipe system indicated in Fig. 6 is now considered obsolete although many of these systems are still in use in older buildings. The same general principles governing its piping design are used when connecting radiators as in other types of gravity systems where they must discharge their condensation to the wet return pipe. Separate supply and return mains and connections are required for each heating unit. Radiator valves are required in both the supply and return connection to the radiator, and air valves are installed on the heating units and the mains. Where the return main has to be located high to function as a

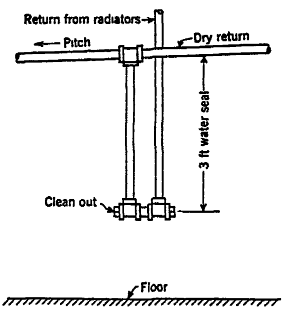


Fig. 7. Method of Connecting Two-Pipe Gravity Returns to Dry Return Main

dry return, it is advisable to connect the return risers to the dry return main through water seals, as shown in Fig. 7, to prevent steam from one riser entering another.

The steam main in the down-feed system is carried to the top of the building, and the piping of the steam side is arranged as in the down-feed one-pipe gravity system. On the return side of the system, the piping is arranged in exactly the same manner as the up-feed gravity two-pipe system.

AIR LINE HEATING SYSTEMS

Both one- and two-pipe systems are at times provided with air valves which, instead of venting to the atmosphere direct, vent to a return pipe system of small size, which in turn is vented to atmosphere or connected to a vacuum pump. These are known as *one-pipe* and *two-pipe* air line systems. Where the air line is exhausted by a vacuum pump they are termed *one-pipe* or *two-pipe vacuum air line systems.*

ONE-PIPE VAPOR SYSTEM

The one-pipe vapor system operates under pressures at or near atmospheric and returns its condensation to the boiler by gravity. In this

system the automatic air valves are of special design to permit the ready release of air and prevent its ready return after it is expelled. The steam radiator valves are a type which, when opened, give a free and unobstructed passageway for water. The piping is the same as for the one-pipe gravity system but sized so as to permit operation at a few ounces pressure.

TWO-PIPE VAPOR SYSTEM

A two-pipe up-feed vapor system using separate supply and return pipes is shown in Fig. 8. The radiators discharge their condensation through thermostatic traps to the dry return pipe. These systems operate at a few ounces pressure and above, but those with mechanical condensate return devices may operate at pressures upward of 10 lb. The simplest

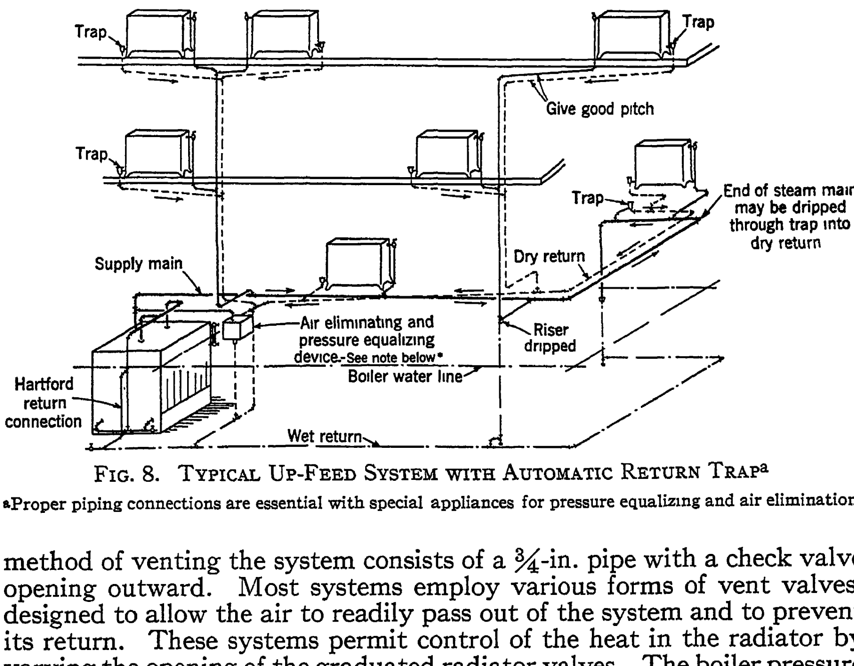


FIG. 8. TYPICAL UP-FEED SYSTEM WITH AUTOMATIC RETURN TRAP[a]

[a]Proper piping connections are essential with special appliances for pressure equalizing and air elimination.

method of venting the system consists of a ¾-in. pipe with a check valve opening outward. Most systems employ various forms of vent valves, designed to allow the air to readily pass out of the system and to prevent its return. These systems permit control of the heat in the radiator by varying the opening of the graduated radiator valves. The boiler pressure is maintained at substantially constant pressure slightly above atmospheric pressure.

These systems may be classified as (1) *closed systems*, consisting of those which have a device to prevent the return of air after it has once been expelled from the system, and which can operate at both super and subatmospheric pressures for a period of four to eight hours depending upon the tightness of the system and rate of firing, and (2) *open systems*, comprising those which have the return line constantly open to the atmosphere without a check or other means to prevent the return of air. The open systems are not so popular because they have the disadvantage of not holding heat when the rate of steam generation is diminishing. Sys-

tems of this design should preferably be equipped with an automatic return trap to prevent water from backing out of the boiler. In installing the return trap a check valve is inserted in the return main at a point near the boiler and a vertical pipe is run up into the bottom of the return trap, which is usually located with the bottom about 18 in. above the boiler water line. Some traps are constructed so that they will operate when they are installed with their bottom as close as 8 in. above the boiler water line. On the other side of this connection a second check valve is installed in the main return just before it enters the boiler. Fig. 9 shows a typical connection for an automatic return trap.

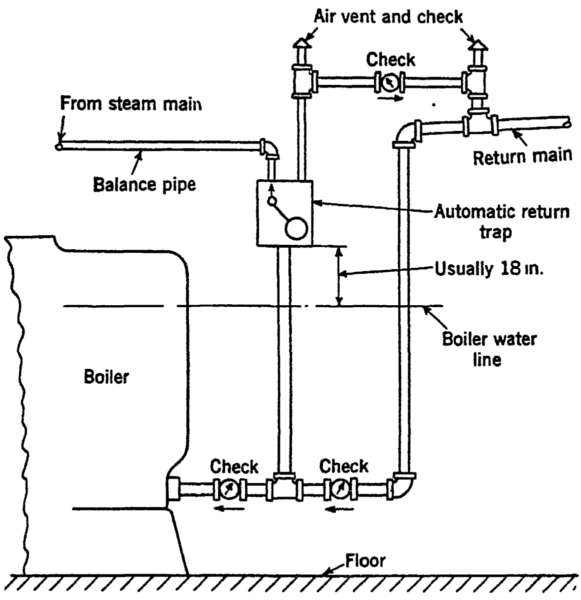


FIG. 9. TYPICAL CONNECTIONS FOR AUTOMATIC RETURN TRAP

Down-Feed Two-Pipe Vapor System

In the down-feed two-pipe vapor system the steam is carried to the top of the building, the top of the vertical riser constituting the high point of the system, and the horizontal supply main is sloped down from this location to the far ends of each branch. The branches are taken off the main from the bottom or at a 45-deg angle downward, with the runouts sloped toward the drops. Thus each branch from the main forms a drip and no accumulation of water is carried down any one drop.

The steam drops are carried down through the building with suitable reductions as the various radiator connections are taken off until the lowest radiator runout is reached. If the drop is only two or three stories high, the portion feeding the bottom radiator should be increased one pipe size to provide for draining the riser, and if the drop is over three stories high it is well to increase the portion feeding the two lowest radiators one or two pipe sizes, especially if the two lowest radiators are small

and the normal size of drop required is 1 in. or less. The bottom of each steam drop should terminate with a dirt pocket and be dripped as shown in Fig. 10. The returns on a down-feed vapor system are the same as on an up-feed system. The runouts to the radiators and the radiator connections of the down-feed system are the same as those for the up-feed system already described.

CONDENSATION RETURN HEATING SYSTEMS

When automatic condensation return pumps are substituted for the gravity return of a two-pipe vapor system they are known as *return systems* or *return pump heating systems*. A typical installation of a motor driven automatic condensation unit is illustrated in Fig. 11. It will be noted that the returns are graded to cause flow by gravity to the vented receiver. As the receiver is filled, the float mechanism operates either a pilot or an across-the-line switch to start the pump and, upon emptying

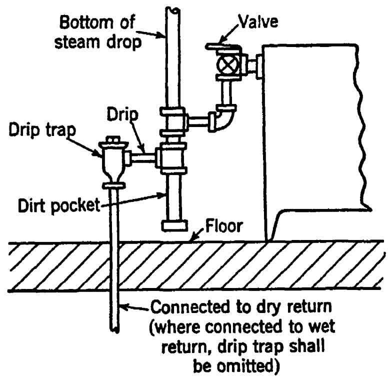


FIG. 10. DETAIL OF DRIP CONNECTIONS AT BOTTOM OF DOWN-FEED STEAM DROP

the tank, to disconnect the power and stop it. The pump may be used to deliver the condensate direct to the boiler, to a feed water heater or to raise the water to any higher elevation or pressure than that of the return line. A useful application is a small condensation unit to handle a remote section of radiation that otherwise would be difficult to grade to the main return.

VACUUM SYSTEMS

In the vacuum system, a vacuum is maintained in the return line practically at all times. The pump is usually controlled by a vacuum regulator which operates the pump to maintain the vacuum within limits and operates in response to a pressure difference between the atmosphere and the return to control the vacuum in the return main, The source of steam supply may be a low pressure boiler as shown in Fig. 12, or a high pressure line through a pressure reducing valve. The piping and other details are the same as for the vapor systems.

The return risers are connected in the basement into a common return main which slopes downward toward the vacuum pump. The vacuum

pump withdraws the air and water from the system, separates the air from the water and expells it to atmosphere and pumps the water back to the boiler, or other receiver, which may be a feed-water heater or hot well. It is essential that no connection be made from the supply side to the return side at any point except through a trap. The desirable practice demands a return flowing to the vacuum pump by an uninterrupted downward slope. In some instances local conditions make it necessary to drop the return below the level of the vacuum pump inlet, before the pump can be reached. In such an event one of the advantages of the vacuum system is the ability to raise the condensate to a considerable height by the suction of the vacuum pump by means of a lift connection or fitting inserted in the return. The height the condensate can be raised depends on the steam pressure and the amount of vacuum maintained. It is

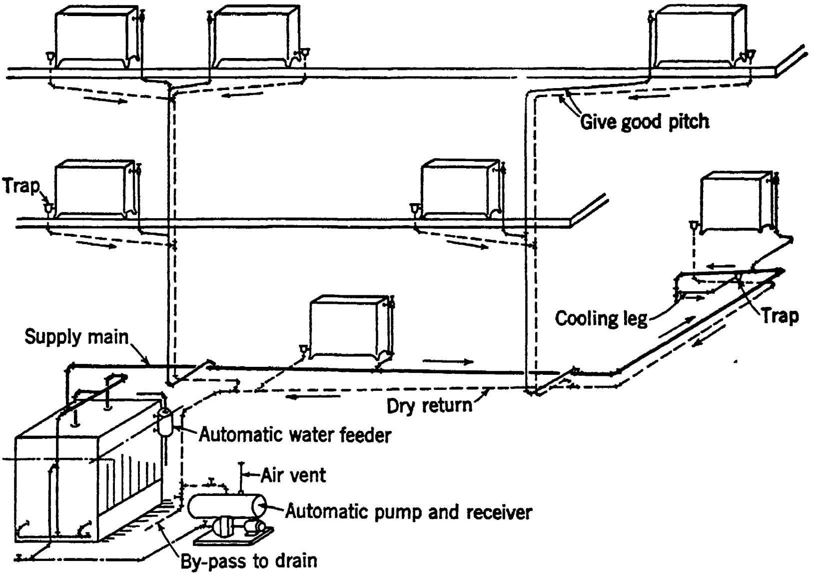


FIG. 11. TYPICAL INSTALLATION USING CONDENSATION PUMP

preferable to limit lift connections to a single lift at the vacuum pump. A still more preferable arrangement is the use of an accumulator tank, or receiver tank, with a float control for the pump at the low point of the return main located adjacent to the vacuum pump.

When the vertical lift is considerable, several lift fittings should be used in steps as shown in Fig. 13. This permits a given lift to be secured with a somewhat lower vacuum than where the vertical distance is served by a single lift. Where several lifts are present in a given system at different locations, the lifting cannot occur until the entire system is filled with steam. A lift connection for location close to the pump, where the size may be above the commercial stock sizes, is shown in Fig. 14. It is desirable that means be provided for manually draining the low point of the lift fittings to eliminate from the return piping all water in danger of freezing in case the system is shut down for a considerable length of time.

Down-Feed Vacuum System

The piping arrangement for the down-feed vacuum system is similar on the supply side to the down-feed vapor system in that it has similar runouts, radiator valves, drips on the bottom of the steam drops, and enlargement of the drops for the lower radiator connections. The return side of the system is exactly the same as the up-feed system except that the steam riser drips at the bottom are connected into the return line through thermostatic traps. It is preferable to take the runouts for the risers from the bottom or at a 45-deg angle down from the steam main so that they may serve as steam main drips. When this is done it is practical to run the steam main level if a runout is located at every change in pipe size, or if eccentric fittings are used (Fig. 15). A slight pitch in the

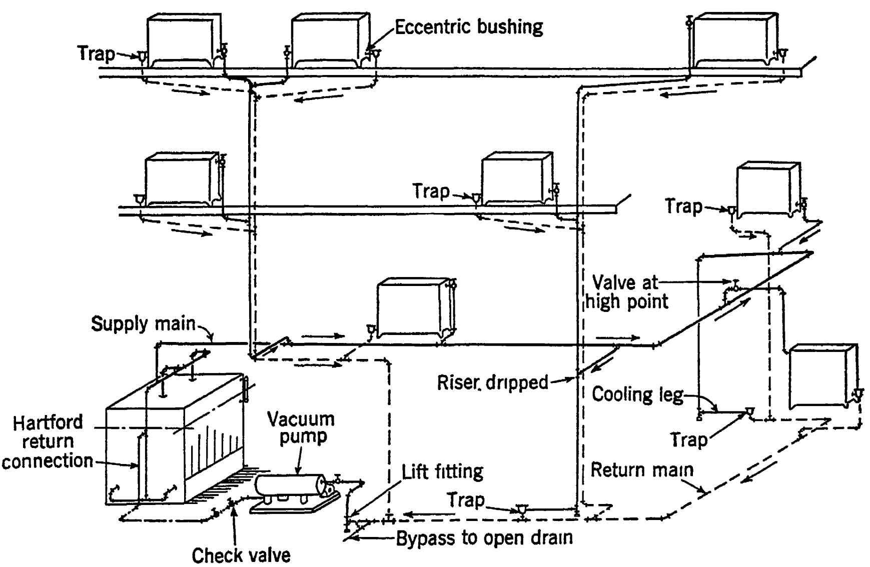


FIG. 12. TYPICAL UP-FEED VACUUM PUMP SYSTEM

steam main, however, should be used when possible. An overhead vacuum down-feed system is shown diagrammatically in Fig. 16.

SUB-ATMOSPHERIC SYSTEMS

Sub-atmospheric systems are similar to vacuum systems but, in contrast, provide control of building temperature by variation of the heat output from the radiators. The radiator heat emission is controlled by varying the pressure, temperature and volume of steam in circulation. These systems differ from the ordinary vacuum system in that they maintain a controllable partial vacuum on both the supply and return sides of the system, instead of only on the return side. In the vacuum system, steam pressure above that of the atmosphere exists in the supply mains and radiators practically at all times. In the sub-atmospheric system, atmospheric pressure or higher exists in the steam supply piping and

radiators only during severe weather. Under average winter temperature the steam is under partial vacuum which in mild weather may reach as high as 25 in. Hg., after which further reduction in heat output is obtained by restricting the quantity of steam.

The rate of steam supply is controlled by a valve in the steam main or by thermostatically controlling the rate of steam production in the boiler. The control valve may be of the automatic *modulating* or *floating* type governed thermostatically from selected control points in the building, or it may be a special pressure reducing valve which will maintain the desired sub-atmospheric pressures by continuous flow into the heating main. All radiator supply valves have incorporated adjustable orifices or are equipped with regulating orifice plates. The sizes of orifices used are larger than for orifice systems because for equal radiator sizes the

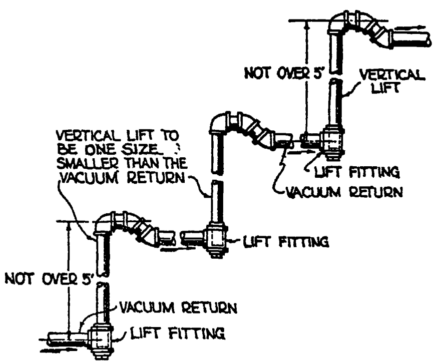


FIG. 13. METHOD OF MAKING LIFTS ON VACUUM SYSTEMS WHEN DISTANCE IS OVER 5 FT

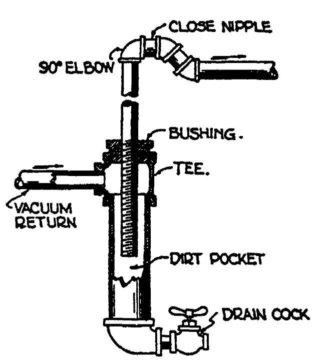


FIG. 14. DETAIL OF MAIN RETURN LIFT AT VACUUM PUMP

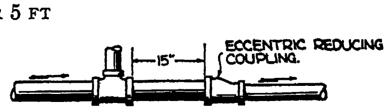


FIG. 15. METHOD OF CHANGING SIZE OF STEAM MAIN WHEN RUNOUTS ARE TAKEN FROM TOP

volume flowing is larger. These orifices are omitted on some systems, depending upon the type of control. Radiator traps and drips are designed to operate at any pressure from 15 lb gage to 26 in. of Hg. A vacuum pump capable of operating at high vacuum is preferable to promote accuracy in the distribution of steam throughout the system, particularly in mild weather. This vacuum is partially self induced by the condensation of the steam in the system under conditions of restricted supply for reduction of the radiator heat emission.

The returns must grade downward constantly and uninterruptedly from the radiator return outlets to the inlet of the receiver of the vacuum pump. One radical difference between this and the ordinary vacuum system is that no lifts should be made in the return line, except at the vacuum pump. The receivers are placed at a lower level than the pump and equipped with float control so the pump may operate as a return pump under night conditions. The system may be operated in the same manner as the ordinary vacuum system when desired.

Steam for heating domestic hot water should be taken from the boiler header back of the control valve so that pressures sufficiently high for heating the water may be maintained on the heater. The sub-atmospheric method of heating can be used for the heating coils of ventilating and air conditioning systems. The flexible control of heat output secured by this method materially reduces the required size of by-pass around the heaters. Sub-atmospheric systems are proprietary.

ORIFICE SYSTEMS

Orifice systems of steam heating may have piping arrangements identical with vacuum systems. Some of these omit the radiator thermostatic

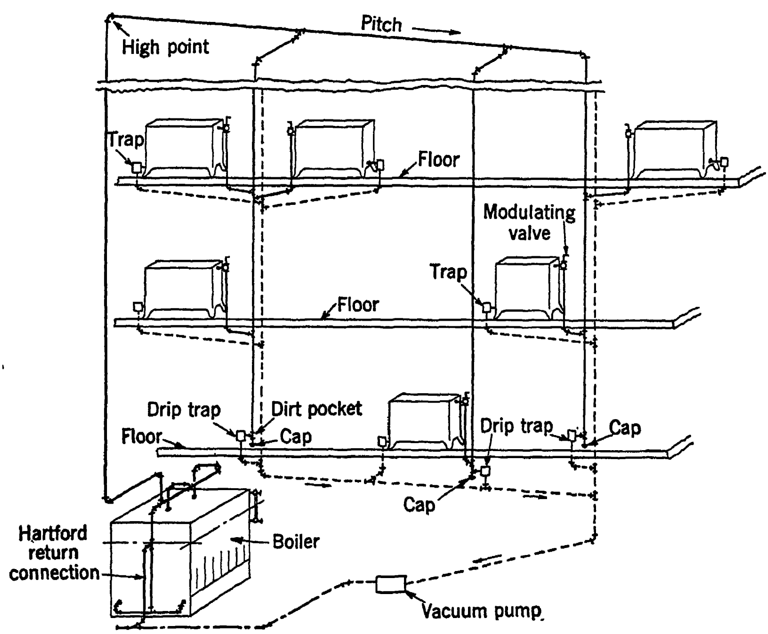


Fig. 16. Typical Down-Feed Vacuum System

traps but use thermostatic or combination float and thermostatic traps on all drip points. A return condensation pump with receiver vented to atmosphere, a return line vacuum pump, or a return trap, is generally used to return the condensation to the boiler or place of similar disposition, such as a feed-water heater or hot well. The heat emission from the radiators is controlled by varying the pressure maintained in the steam supply piping.

The principle on which these systems operate is based on the fact that the steam flow through an orifice will vary when the ratio of the absolute pressures on the two sides of the orifice exceeds 58 per cent. If the absolute pressure on the outlet side is less than 58 per cent of the absolute pressure on the inlet side, no further increase in flow will be obtained as a result of the increased pressure difference. If an orifice is so designed in

size as to exactly fill a radiator with 2 lb gage on one side and ¼ lb gage on the other, the absolute pressure relation is:

$$\frac{14.7 + 0.25}{14.7 + 2.0} = 0.90 \text{ or } 90 \text{ per cent.}$$

Should the steam pressure be dropped to ¼ lb on the supply pipe, the pressure on each side of the orifice would be balanced and no steam flow would take place. From this it will be apparent that if an orifice of a given diameter will fill a given radiator with steam when there is a given pressure on the main, reducing this steam main pressure will permit filling various desired portions of the radiator down to the point where the main pressure equals the back pressure in the radiator provided the supply pipe pressures may be controlled sufficiently close. If orifices are designed on a similar basis for a given system and proportioned to the heating capacity of the radiators they serve, all radiators will heat proportionately to the steam pressure. The range of pressure variation is limited by the permissible noise level of the steam flowing under the pressure difference required for maximum heat output. The control of the steam supply is obtained by a valve placed in the steam main, which maintains a determined pressure; or by a boiler pressure control. The valves are frequently manually set from a remote location, guided by temperature indicating stations in the building; or thermostatically controlled from a thermostat on the roof, which automatically measures the differential of outside and inside temperatures. Since the range through which the pressures may be varied is usually from 0 to 4.0 lb gage, the control should be capable of maintaining close regulation to maintain the desired space temperatures, particularly in mild weather.

Some systems use orifices not only in radiator inlets but also at different points in the steam supply piping for the purpose of balancing the system to a greater extent. In this manner the difference between the initial and terminal pressure in the steam main may be compensated to a great extent. For example, if the initial pressure was 3 lb gage and the pressure at the end of the main was 2 lb, an orifice could be used in each branch for the purpose of obtaining a more uniform pressure throughout the system. Such a provision may be particularly useful in this system for branches close to the boiler where the drop in the main has not yet been produced. Orifice systems are proprietary.

CONDENSATION RETURN PUMPS

Condensation return pumps are used for gravity systems when the local conditions do not permit the condensation to return to the boiler under the existing static head. The return of the condensate permits the water to repeatedly go through the cycle of vaporization, with subsequent condensation and return to the boiler. During such repeated cycles any incrustants or other substances in solution are precipitated and the water de-activated to a considerable extent so that corrosion of a serious nature is seldom ever encountered where the condensate is repeatedly used. Serious corrosion is more frequently found in systems where the condensation is not repeatedly used but is wasted and fresh make-up water is continually being introduced.

The most generally accepted condensation pump unit for low pressure

heating systems consists of a motor-driven centrifugal pump with receiver and automatic float control. Other types in use include rotary, screw and reciprocating pumps with steam turbine or motor drive, and direct-acting steam reciprocating pumps.

The receiver capacities of these automatic units should be sized so as not to cause too great a fluctuation of the boiler water line if fed directly to the boiler and at the same time not so small as to cause too frequent operation of the unit. The usual unit provides storage capacity between stops in the receiver of approximately 1.5 times the amount of condensate returned per minute and the pump generally has a delivery rate of 3 to 4 times the normal flow. This relation of receiver and pump size to heating system condensing capacity takes account of the peak condensation rate.

VACUUM HEATING PUMPS

On vacuum systems, where the returns are under a vacuum, and sub-atmospheric systems, where the supply piping, radiation and the returns are under a vacuum, it is necessary to use a vacuum pump to discharge the air and non-condensable gases to atmosphere and to dispose of the condensation. Direct-acting steam driven reciprocating vacuum pumps are sometimes used where high pressure steam is available or where the exhaust steam from the pump can be utilized. In general, however, these have been replaced by the automatic motor-driven return line heating pump especially developed for this service. Steam turbine drive is also frequently used where steam at suitable pressures is available, the steam being used afterward for building heating. The usual vacuum pump unit consists of a compact assembly of exhausting unit for withdrawing the air-vapor mixture and discharging the air to atmosphere and a water removal unit which discharges the condensate to the boiler. They are furnished complete with receiver, separating tank and automatic controls mounted as an integrated unit on one base. There are also special steam turbine driven units which are operated by passing the steam to be used in heating the building through the turbine with only a 2 to 3 lb drop across the turbine required for its operation. Under special conditions such as installations where it is necessary to return the condensate to a high pressure boiler, auxiliary water pumps may be supplied. In some instances separate air and water pumps may be used.

Practically all automatic motor-driven return line vacuum heating pumps make use of a portion of the condensate to operate either as a liquid piston pump or as a kinetic exhauster (which operate on a modified ejector principle) to withdraw the air and condensate from the system, discharge the air to atmosphere and return the condensate to the boiler. Some type of hydraulic action is utilized to produce the suction. Such hydraulic evacuating devices may be classified as:

a. Water ring centrifugal displacement pumps.
b. Water piston pumps.
c. Stationary kinetic exhauster pumps.
d. Rotary kinetic ejector pumps.

The evacuating element is generally combined with a centrifugal water impeller for the delivery of the condensate to the boiler or feed-water heater.

The assembled units may be further grouped under two general classifications:

a. Those which perform the function of air separation under atmospheric pressure.

b. Those which perform the function of air separation under a partial vacuum.

Pumps coming under the first classification remove both the air and condensate from the returns by means of the hydraulic evacuator and deliver both to a separating tank under atmospheric pressure. From this tank the air and non-condensable vapors are vented to atmosphere while the condensate is removed and delivered to the boiler by means of the built-in boiler feed pump impeller.

In the second classification, the air and condensate are first separated under vacuum by means of the receiver which is directly connected to the returns. The hydraulic evacuator withdraws only the air and non-condensable vapors from the top of the receiver and delivers them to atmosphere. The built-in condensate pump impeller removes the condensate from the bottom of the receiver and delivers it direct to the boiler or feed-water heater.

Under special conditions such as returning the condensate to a high pressure boiler or the furnishing of large air removal units for high vacuum systems, it is customary to supply separate motor-driven air and water pumps.

For rating purposes[1] vacuum pumps are classified as *low vacuum* and *high vacuum*. Low vacuum pumps are those rated for maintaining 5½ in. Hg. vacuum on the system, and high vacuum pumps are those rated to maintain vacuums above 5½ in.

The vacuum that can be maintained on a system depends upon the relationship of the air leakage rate into the system to the operating air capacity of the hydraulic evacuator when operating at any given return line temperature. The hotter the returns, the lower will be the possible vacuum for a given air leakage rate into the system. It is particularly essential on high vacuum installations to see that the entire system is tight in order to reduce the amount of inward air leakage and, furthermore, to see that relatively higher temperature steam is prevented from entering the vacuum return lines through leaky traps, high pressure drips, etc. It is for this reason that the condensate from equipment using steam at high pressures should not be connected directly to a vacuum return line but should drain to a receiver through a high pressure trap. The receiver should have an equalizing connection to a low pressure steam main and drain through a low pressure trap to the vacuum return main as indicated in Fig. 17.

Vacuum Pump Controls

In the ordinary vacuum system, the vacuum pump is controlled by a vacuum regulator which cuts in when the vacuum drops to the lowest point desired and cuts out when it has been increased to the highest point, these points being varied to suit the particular system or operating

[1]A.S.H.V.E. Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps, (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 33).

conditions. In addition to this vacuum control, a float control is included which will automatically start the pump whenever sufficient condensation accumulates in the receiver, regardless of the vacuum on the system. A selector switch is usually provided to allow operation at night as a condensation pump only, also to give manual or continuous operation when desired.

There are several variations in the control of the vacuum maintained on the system by the pump. In some sub-atmospheric systems where orifices are used, the vacuum pump control maintains a pressure difference between the supply and the return piping, which is held within relatively close limits. There are other sub-atmospheric systems which utilize special temperature-pressure actuated controls for maintaining the desired conditions in the return lines. Where various zones are connected to the

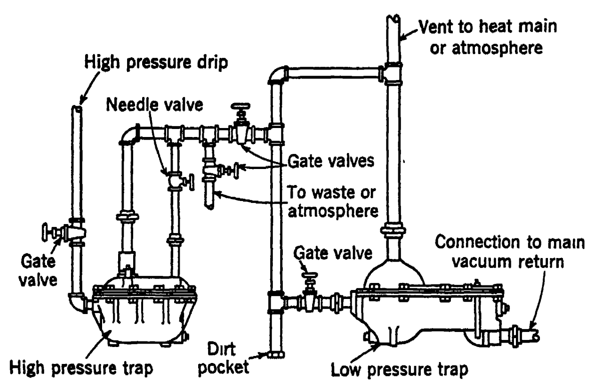


FIG. 17. METHOD OF DISCHARGING HIGH-PRESSURE APPARATUS INTO LOW-PRESSURE HEATING MAINS AND VACUUM RETURN MAINS THROUGH A LOW-PRESSURE TRAP

same return main, the return vacuum must be controlled to meet the requirements of the zone operating at the lowest steam supply pressure.

Piston Displacement Vacuum Pumps

Piston displacement return vacuum heating pumps may be either electric or steam driven. They should be provided with mechanical lubricators and their piston speed in feet per minute should not exceed 20 times the square root of the number of inches in their stroke. They are usually supplied with an air separating tank, open to atmosphere, placed on the discharge side of the pump and at an elevation sufficiently high to allow gravity flow of the condensate to the boiler. If the boiler pressure is too high for such gravity feed then an additional steam pump for feeding the boiler is desirable. The extra pump is sometimes avoided by using a closed separating tank with a float controlled vent. In both arrangements, the air taken from the system must be discharged against the full discharge pressure of the vacuum pump. In the case of high or medium pressure boilers, it is better to use the atmospheric separator and the second pump.

In figuring the required displacement for such pumps, a value of from 6 to 10 times the volumetric flow of condensation is used for average vacuums and systems.

TRAPS

Traps are generally classified as to function as (*a*) separating traps, (*b*) return, lifting or vacuum traps, and (*c*) air traps. Separating traps may be either float operated, thermostatically operated, or float and thermostatically operated. Return traps for low pressure service are referred to later as *alternating receivers* in this chapter. Return traps may also operate to receive condensate under a vacuum and return it to atmosphere or a higher pressure. Air traps are generally float operated.

Separating traps are used to release water of condensation but to retain steam. The thermostatic, and float and thermostatic types release both condensate and air but retain steam. Separating traps are used for draining condensate from radiators, indirect air heaters, steam piping systems, kitchen equipment, laundry equipment, hospital equipment, drying equipment and many other kinds of apparatus. Air traps release air but retain water. Devices known as air vents are, in principle, traps which allow the passage of air but prevent the passage of either water or steam.

Return traps are used for returning condensate either by gravity, by steam pressure, or by both, to a boiler or other point of disposal, and for lifting condensate from a lower to a higher elevation, or for handling condensate from a lower to a higher pressure.

The fundamental principle upon which the operation of practically all traps depends is that the pressure within the trap at the time of discharge shall be equal to, or slightly in excess of, the pressure against which the trap must discharge, including the friction head, velocity head and static head on the discharge side of the trap. If the static head is in favor of the trap discharge it is a minus quantity and may be deducted from the other factors of the discharge head.

Traps may also be classified according to the principle of operating device which supplies the power to cause them to function as (1) float, (2) bucket, (3) thermostatic, (4) float and thermostatic, (5) impulse, or (6) tilting traps.

Float Traps. A discharge valve is operated by the rise and fall of a float due to the change of water level in the trap. When the trap is empty the float is in its lowest position, and the discharge valve is closed. A gage glass indicates the height of water in the chamber.

Unless float traps are well made and proportioned there is danger of considerable steam leakage through the discharge valve due to unequal expansion of the valve and seat and the sticking of moving parts. The discharge from a float trap is usually continuous since the height of the float, and consequently the area of the outlet, is proportional to the amount of water present.

Bucket Traps. Bucket traps are of two types, the upright and inverted, and although they are both of the open float construction, their operating principle is entirely different. In the ***upright bucket*** trap, the water of condensation enters the trap and fills the space between the bucket and the walls of the trap. This causes the bucket to float and forces the valve against its seat, the valve and its stem usually being fastened to the bucket. When the water rises above the edges of the bucket it flows into it and causes it to sink,

thereby withdrawing the valve from its seat. This permits the steam pressure acting on the surface of the water in the bucket to force the water to a discharge opening. When the bucket is emptied it rises and closes the valve and another cycle begins. The discharge from this type of trap is intermittent.

In the *inverted bucket* trap, steam floats the inverted submerged bucket and closes the valve. Water entering the trap fills the bucket, which sinks and through compound leverage opens the valve, and the trap discharges. It is impossible to install a water gage glass on an inverted bucket trap, but if visual inspection is necessary, a gage glass can be placed on the line leading to the trap. No air relief cocks can be used, but this is unnecessary, as the elimination of air is automatically taken care of by air passing through the vent in the top of the inverted bucket regardless of temperature.

Thermostatic Traps. Thermostatic traps are of two types, those in which the discharge valve is operated by the relative expansion of metals, and those in which the action of a volatile liquid is utilized for this purpose. Thermostatic traps of large capacity for draining blast coils or very large radiators are called *blast* traps.

Float and thermostatic traps have both a thermostatic element to release air and a float element to release the water.

Impulse traps operate with a moving valve actuated by a control cylinder. When the trap is handling condensate, the pressure required to lift the valve is greater than the reduced pressure in the control cylinder and consequently the valve opens allowing a free discharge of condensate. As the remaining condensate approaches steam temperature, flashing results, flow through the valve orifice is choked and the pressure builds up in the control chamber closing the valve.

Automatic Return Traps

In the general heating plant, where thermostatic traps are installed on the heating units, it becomes necessary to provide a means for returning the water of condensation to the boiler, if a condensation or vacuum pump is not used. When the return main can be kept sufficiently high above the boiler water line for all operating conditions, the water of condensation will flow back by gravity, and no mechanical device is required. But actually this does not work out in practice. It follows, therefore, that a direct-return trap is needed for the handling of the condensation even though it may not be called into action except under some operating condition where the pressure differential exceeds the static head provided. The installation of a direct-return trap assures safety for such systems, and guarantees the operation of the plant under varying conditions.

Automatic return traps, sometimes called alternating receivers, may be of the counter-balanced, tilting type, or spring actuated. These consist of a small receiver with an internal float, and when the condensate will not flow into the boiler under pressure, it will feed into the receiver of the trap, and in so doing, raise or tilt the float or mechanism which actuates a steam valve automatically. This admits steam to the receiver, at boiler pressure, and the equalizing of the pressures which follows allows the water to flow into the boiler.

Tilting Traps. With this type of trap, water enters a bowl and rises until its weight overbalances that of a counter-weight, and the bowl sinks to the bottom. As the bowl sinks, a valve is opened thus admitting live steam pressure on the surface of the water and the trap then discharges. After the water is discharged, the counter-weight sinks and raises the bowl, which in turn closes the valve and the cycle begins again. Tilting traps are necessarily intermittent in operation. They are not ordinarily equipped with glass water gages, as the action of the trap shows when it is filling or emptying. The air relief of tilting traps is taken care of by the valves of the trap.

Chapter 15

PIPING FOR STEAM HEATING SYSTEMS

Operating Characteristics, Steam Flow, Pipe Sizes, Tables for Pipe Sizing, One-Pipe Gravity Air-Vent Systems, Two-Pipe Gravity Air-Vent Systems, Two-Pipe Vapor Systems, Vacuum, Orifice, Atmospheric and Sub-Atmospheric Systems, Boiler and Radiator Connections, Piping for Indirect Heating Units, Dripping

IT is important that steam piping systems distribute steam not only at full *design* load but during excess and partial loads. Usually the average winter steam demand is less than half of the demand at the *design* outside temperature. Moreover, in rapidly warming up a system even in moderate weather, the load on the steam main and returns may exceed the maximum operating load for severe weather due to the necessity of raising the temperature of the metal in the system to the steam temperature and the building to the *design* indoor temperature. Investigations of the return of condensation have revealed that as high as 143 per cent of the design condensation rate may exist under conditions of actual operation.

The functions of the piping system are the distribution of the steam, the return of the condensate and in systems where no local air vents are provided, the removal of the air. The distribution of the steam should be rapid, uniform and without noise, and the release of air should be facilitated as much as possible, as an air bound system will not heat readily nor properly. In designing the piping arrangement it is desirable to maintain equivalent resistances in the supply and return piping to and from a radiator. Arranging the piping so the total distance from the boiler to the radiation is the same as the return piping distance from the heating unit back to the boiler tends to obtain such a result. The condensation which occurs in steam piping as well as in radiators must be drained to prevent impeding the ready flow of the steam and air. The effect of back pressure in the returns and excessive revaporization, such as occurs where condensation is released from pressures considerably higher than the vacuum or pressure in the return, must be avoided.

The piping design of a heating system is greatly influenced by its operating characteristics. Heating systems do not operate under constant conditions as they are continually changing due to variation in load. As the system is being filled with steam the pressure existing in various

Table 1. Flow of Steam in Pipes

P = loss in pressure in pounds.
D = inside diameter of pipe in inches.
L = length of pipe in feet.
d = weight of 1 cu ft of steam.
W = pounds of steam per hour.

$$W = 5220\sqrt{\frac{PdD^5}{\left(1+\frac{3.6}{D}\right)L}}$$

$$P = 0.0000000367\left(1+\frac{3.6}{D}\right)\frac{W^2L}{dD^5}$$

Pressure Loss in Ounces	Col. 1 $5220\sqrt{\frac{P}{100}}$	Pipe Size Nominal	Pipe Size Actual Internal Diameter	Internal Area of Pipe Sq Inches	Col. 2 $\sqrt{\frac{D^5}{1+\frac{3.6}{D}}}$	Steam Press. by Gage	Col. 3 $\sqrt{d}$	Length of Pipe in Feet	Col. 4 $\sqrt{\frac{100}{L}}$
0.25	65.28	1	1.049	0.864	0.536	−1.0[a]	0.187	20	2.240
0.50	92.28	1¼	1.380	1.496	1.178	−0.5[a]	0.190	40	1.580
1.00	130.5	1½	1.610	2.036	1.828	0.0	0.193	60	1.290
2	184.6	2	2.067	3.356	3.710	0.3	0.195	80	1.120
3	226.0	2½	2.469	4.788	6.109	1.3	0.201	100	1.000
4	261.0	3	3.068	7.393	11.183	2.3	0.207	120	0.912
5	291.8	3½	3.548	9.887	16.705	5.3	0.223	140	0.841
6	319.7	4	4.026	12.730	23.631	10.3	0.248	160	0.793
7	345.3	4½	4.506	15.947	32.134	15.3	0.270	180	0.741
8	369.1	5	5.047	20.006	43.719	20.3	0.290	200	0.710
10	412.7	6	6.065	28.886	71.762	30.3	0.326	250	0.632
12	452.0	7	7.023	38.743	106.278	40.3	0.358	300	0.578
14	488.3	8	7.981	50.027	149.382	50.3	0.388	350	0.538
16	522.0	9	8.941	62.786	201.833	60.3	0.415	400	0.500
20	583.6	10	10.020	78.854	272.592	75.3	0.452	450	0.477
24	639.3	12	12.000	113.098	437.503	100.3	0.507	500	0.447
28	690.5	14	13.250	137.880	566.693	125.3	0.557	600	0.407
32	738.2	16	15.250	182.655	816.872	150.3	0.603	700	0.378
40	825.4					175.3	0.645	800	0.354
48	904.1					200.3	0.685	900	0.333
80	1167.2							1000	0.316
160	1650.7							1200	0.289
320	2334.5							1500	0.258
480	2859.1							2000	0.224

Column 1 × 2 × 3 × 4 = lb of steam per hour that will flow through a straight pipe for a given condition.

Example 1: 1 oz drop — 2 in. pipe — 1.3 lb press. — 100 ft equivalent length:

130.5 × 3.710 × 0.201 × 1 = 97.2 lb per hour.
97.2 × 4[b] = 388.8 sq ft equivalent radiation.

Table 1 does not allow for entrained water in low-pressure steam, condensation in covered pipe and roughness in commercial pipe as found in practice.

[a]Pounds per square inch gage = 2.04 in. Vacuum, Mercury Column.
[b]The factor 4 is the approximate equivalent in square feet of steam radiation of 1 lb of steam per hour.

locations may be different than those which exist for appreciable periods at other locations and which under constant pressure may have conditions that are approximately the same. In designing piping it is of especial importance to arrange the system to preclude trouble caused by such pressure differences. The systems which readily release the air permit uniform pressures to be attained in much shorter time intervals than those which are sluggish. Results are given in Fig. 1 from investigations[1] to determine the rate of condensate and air return from a two-pipe gravity heating system. Variations in the steam pressure during the warming up period when the rate of air elimination and condensation is high are clearly indicated in these curves.

It is evident that the condensation flow during the initial warming up

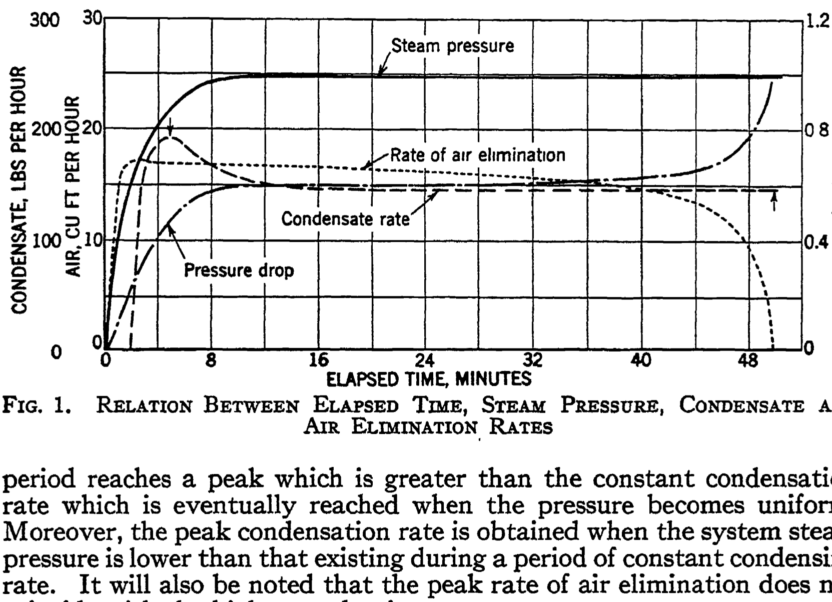


FIG. 1. RELATION BETWEEN ELAPSED TIME, STEAM PRESSURE, CONDENSATE AND AIR ELIMINATION RATES

period reaches a peak which is greater than the constant condensation rate which is eventually reached when the pressure becomes uniform. Moreover, the peak condensation rate is obtained when the system steam pressure is lower than that existing during a period of constant condensing rate. It will also be noted that the peak rate of air elimination does not coincide with the higher condensing rate.

STEAM FLOW

The rate of flow of dry steam or steam with a small amount of water flowing in the same direction is in accordance with the general laws of gas flow and is a function of the length and diameter of the pipe, the density of the steam, and the pressure drop through the pipe. This relationship has been established by Babcock in the formula given at the top of Table 1. In Columns 1, 2, 3, and 4 of this table, the numerical values of the factors for different pressure losses, pipe diameters, steam densities and lengths of pipe have been worked out in convenient form so that the steam flowing in any pipe may be calculated by multiplying together the proper factors in each column as shown in the example at the bottom of the table.

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[1]A.S.H.V.E. RESEARCH REPORT No. 954—Condensate and Air Return in Steam Heating Systems, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 199).

Table 2. Maximum Allowable Capacities of Up-Feed Risers for One-Pipe Low Pressure Steam

Based on A. S. H. V. E. Research Laboratory Tests

Pipe Size Inches	Velocity Feet Per Second	Pressure Drop Ounces per 100 Ft	Capacity		
			Sq Ft Radiation	Btu per Hour	Lb Steam per Hour
A	*B*	*C*	*D*	*E*	*F*
1	14.1	0.68	45	10.961	11.3
1¼	17.6	0.66	98	23,765	24.5
1½	20.0	0.66	152	36,860	38.0
2	23.0	0.57	288	69,840	72.0
2½	26.0	0.54	464	112,520	116.0
3	29.0	0.48	799	193,600	199.8
3½	31.0	0.44	1144	277,000	286.0
4	32.0	0.39	1520	368.000	380.0

INSTRUCTIONS FOR USING TABLE 2

1. Capacities given in Table 2 should never be exceeded on one-pipe risers.
2. Capacities are based on ¼-lb condensation per square foot equivalent radiation and actual diameter of standard pipe.
3. All pipe should be well reamed and free from constrictions. Fittings should be up to size.

Table 3. Maximum Allowable Capacities of Up-Feed Risers for Two-Pipe Low Pressure Steam

Based on A. S. H. V. E. Research Laboratory Tests

Pipe Size Inches	Velocity Feet per Second	Pressure Drop Ounces per 100 Ft	Capacity		
			Sq Ft Radiation	Btu per Hour	Lb Steam per Hour
A	*B*	*C*	*D*	*E*	*F*
¾	20	------	40	9,550	10.0
1	23	1.78	74	17,900	18.45
1¼	27	1.57	151	36,500	37.65
1½	30	1.48	228	55,200	57.0
2	35	1.33	438	106,100	109.5
2½	38	1.16	678	164,100	169.4
3	41	0.95	1129	273,500	282.2
3½	42	0.81	1548	375,500	387.0
4	43	0.71	2042	495,000	510.5

INSTRUCTIONS FOR USING TABLE 3

1. The capacities given in this table should never be exceeded on two-pipe risers.
2. Capacities are based on ¼-lb condensation per square foot equivalent radiation and actual diameter of standard pipe.
3. All pipe should be well reamed and free from constrictions. Fittings should be up to size.

PIPE SIZES

The determination of pipe sizes for a given load in steam heating depends on the following principal factors:

1. The initial pressure and the total pressure drop which may be allowed between the source of supply and the end of the return system.

2. The maximum velocity of steam allowable for quiet and dependable operation of the system, taking into consideration the direction of condensate flow.

3. The equivalent length of the run from the boiler or source of steam supply to the farthest heating unit.

Initial Pressure and Pressure Drop

Theoretically there are several factors to be considered, such as initial pressure and pressure required at the end of the line, but it is most important that (1) the total pressure drop does not exceed the initial pressure of the system; (2) the pressure drop is not so great as to cause excessive velocities; (3) there is a constant initial pressure, except on systems specially designed for varying initial pressures, such as the sub-atmospheric which normally operate under controlled partial vacua, the orifice, and the vapor systems which at times operate under such partial vacua as may be obtained due to the condition of the fire; and (4) the equivalent head due to pressure drop does not exceed the difference in level, for gravity return systems, between the lowest point on the steam main, the heating units, or the dry return, and the boiler water line.

All systems should be designed for a low initial pressure and a reasonably small pressure drop for two reasons: *first*, the present tendency in steam heating unmistakably points toward a constant lowering of pressures even to those below atmospheric; *second*, a system designed in this manner will operate under higher pressures without difficulty. When a system designed for a relatively high initial pressure and a relatively high pressure drop is operated at a lower pressure, it is likely to be noisy and have poor circulation.

The total pressure drop should never exceed one-half of the initial pressure when condensate is flowing in the same direction as the steam. Where the condensate must flow counter to the steam, the governing factor is the velocity permissible without interfering with the condensate flow. A.S.H.V.E. Research Laboratory experiments limit this to the capacities given in Tables 2 and 3 for vertical risers and in Table 4 for horizontal pipes at varying grades.

Maximum Velocity

The capacity of a steam pipe in any part of a steam system depends upon the quantity of condensation present, the direction in which the condensate is flowing, and the pressure drop in the pipe. Where the quantity of condensate is limited and is flowing in the same direction as the steam, only the pressure drop need be considered. When the condensate must flow against the steam, even in limited quantity, the velocity of the steam must not exceed limits above which the disturbance between the steam and the counter-flowing water may produce objectionable sounds, such as water hammer, or may result in the retention of

TABLE 4. COMPARATIVE CAPACITY OF STEAM LINES AT VARIOUS PITCHES FOR STEAM AND CONDENSATE FLOWING IN OPPOSITE DIRECTIONS[a]

Pitch of Pipe in Inches per 10 Ft

PITCH OF PIPE	¼ IN.		½ IN.		1 IN.		1½ IN.		2 IN.		3 IN.		4 IN.		5 IN.	
Pipe Size Inches	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.
¾	25.0	12	30.3	14	37.3	18	40.4	19	42.5	20	46.1	21	47.5	22	49.3	23
1	45.8	12	52.6	15	63.0	17	70.0	20	75.2	22	83.0	23	87.9	25	90.2	26
1¼	104.9	18	117.2	20	133.0	23	144.5	25	154.0	27	165.0	28	172.6	29	178.2	31
1½	142.6	18	159.0	21	181.0	23	196.5	25	209.3	27	224.0	28	234.8	30	242.6	31
2	236.0	19	263.5	20	299.5	23	325.5	25	346.5	27	371.5	28	388.4	29	401.1	30

[a]Data from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory.

water in certain parts of the system until the steam flow is reduced sufficiently to permit the water to pass. The velocity at which such disturbances take place is a function of (1) the pipe size, whether the pipe runs horizontally or vertically, (2) the pitch of the pipe if it runs hori-

TABLE 5. LENGTH IN FEET OF PIPE TO BE ADDED TO ACTUAL LENGTH OF RUN—OWING TO FITTINGS—TO OBTAIN EQUIVALENT LENGTH

SIZE OF PIPE INCHES	LENGTH IN FEET TO BE ADDED TO RUN				
	Standard Elbow	Side Outlet Tee	Gate Valve[a]	Globe Valve[a]	Angle Valve[a]
½	1.3	3	0.3	14	7
¾	1.8	4	0.4	18	10
1	2.2	5	0.5	23	12
1¼	3.0	6	0.6	29	15
1½	3.5	7	0.8	34	18
2	4.3	8	1.0	46	22
2½	5.0	11	1.1	54	27
3	6.5	13	1.4	66	34
3½	8	15	1.6	80	40
4	9	18	1 9	92	45
5	11	22	2 2	112	56
6	13	27	2 8	136	67
8	17	35	3 7	180	92
10	21	45	4 6	230	112
12	27	53	5 5	270	132
14	30	63	6 4	310	152

[a]Value in full open position.

Example of length in feet of pipe to be added to actual length of run.

Measured Length	=	132.0 ft
4 in. Gate Valve	=	1.9 ft
4—4 in. Elbows	=	36.0 ft
Equivalent Length	=	169.9 ft

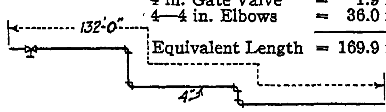

Table 6. Steam Pipe Capacities

Capacity Expressed in Square Feet of Equivalent Direct Radiation

(Reference to this table will be by column letter *A* through *L*)

This table is based on pipe size data developed through the research investigations of the American Society of Heating and Ventilating Engineers.

	Capacities of Steam Mains and Risers								Special Capacities for One-Pipe Systems Only		
	Direction of Condensation Flow in Pipe Line										
Pipe Size In	With the Steam in One-Pipe and Two-Pipe Systems						Against the Steam Two-Pipe Only		Supply Risers Up-Feed	Radiator Valves and Vertical Connections	Radiator and Riser Run-outs
	1/32 lb or ½ Oz Drop	1/24 lb or ⅔ Oz Drop	1/16 lb or 1 Oz Drop	⅛ lb or 2 Oz Drop	¼ lb or 4 Oz Drop	½ lb or 8 Oz Drop	Vertical	Horizontal			
A	*B*	*C*	*D*	*E*	*F*	*G*	*H*[a]	*I*[c]	*J*[b]	*K*	*L*[c]
¾	......	......	30	......	......	......	30	......	25	......	......
1	39	46	56	79	111	157	56	26	45	20	20
1¼	87	100	122	173	245	346	122	58	98	55	55
1½	134	155	190	269	380	538	190	95	152	81	81
2	273	315	386	546	771	1,091	386	195	288	165	165
2½	449	518	635	898	1,270	1,797	635	395	464	......	260
3	822	948	1,163	1,645	2,326	3,289	1,129	700	799	......	475
3½	1,228	1,419	1,737	2,457	3,474	4,913	1,548	1,150	1,144	......	745
4	1,738	2,011	2,457	3,475	4,914	6,950	2,042	1,700	1,520	......	1,110
5	3,214	3,712	4,546	6,429	9,092	12,858	......	3,150	......	......	2,180
6	5,276	6,094	7,462	10,553	14,924	21,105	......	......	......	......	......
8	10,983	12,682	15,533	21,967	31,066	43,934	......	......	......	......	......
10	20,043	23,144	28,345	40,085	56,689	80,171	......	......	......	......	......
12	32,168	37,145	45,492	64,336	90,985	128,672	......	......	......	......	......
16	60,506	69,671	84,849	121,012	169,698	242,024	......	......	......	......	......
	All Horizontal Mains and Down-Feed Risers						Up-Feed Risers	Mains and Undripped Run-outs	Up-Feed Risers	Radiator Connections	Run-outs Not Dripped

Note.—All drops shown are in pounds per 100 ft of equivalent run—based on pipe properly reamed.

[a]Do not use Column *H* for drops of 1/24 or 1/32 lb; substitute Column *C* or Column *B* as required.

[b]Do not use Column *J* for drop of 1/32 lb except on sizes 3 in. and over; below 3 in. substitute Column *B*.

[c]On radiator runouts over 8 ft long increase one pipe size over that shown in Table 6.

zontally, (3) the quantity of condensate flowing against the steam, and (4) freedom of the piping from water pockets which under certain conditions act as a restriction in pipe size.

Three factors of uncertainty always exist in determining the capacity of any steam pipe. The first is variation in manufacture, which apparently cannot be avoided. The second is the care used in reaming the ends of the pipe after cutting. The effect of both of these factors increases as the pipe size decreases. According to A.S.H.V.E. Research Laboratory tests, either of these factors may affect the capacity of a 1-in. pipe as much as 20 per cent. The third factor is the uniformity in grading the pipe line. All of the capacity tables given in this chapter include a factor of safety. However, the factor of safety referred to does not cover abnormal defects or constrictions *nor does it cover pipe not properly reamed.*

Table 7. Return Pipe Capacities

Capacity Expressed in Square Feet of Equivalent Direct Radiation

(Reference to this table will be by column letter *M* through *EE*)

This table is based on pipe size data developed through the research investigations of the American Society of Heating and Ventilating Engineers.

CAPACITY OF RETURN MAINS AND RISERS

Pipe Size Inches	1/32 Lb or ½ Oz Drop per 100 Ft			1/24 Lb or ⅔ Oz Drop per 100 Ft			1/16 Lb or 1 Oz Drop per 100 Ft			⅛ Lb or 2 Oz Drop per 100 Ft			¼ Lb or 4 Oz Drop per 100 Ft			½ Lb or 8 Oz Drop per 100 Ft		
	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.
M	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	*X*	*Y*	*Z*	*AA*	*BB*	*CC*	*DD*	*EE*
Mains																		
¾	......	......	......	......	......	326	......	......	400	......	......	568	......	......	800	......	......	1,130
1	500	248	......	580	285	570	700	320	700	1,000	412	994	1,400	460	1,400	......	......	1,977
1¼	850	520	......	990	595	976	1,200	670	1,200	1,700	868	1,704	2,400	962	2,400	......	......	3,390
1½	1,350	822	......	1,570	943	1,547	1,900	1,058	1,900	2,700	1,362	2,696	3,800	1,512	3,800	......	......	5,370
2	2,800	1,880	......	3,240	2,140	3,256	4,000	2,300	4,000	5,600	2,960	5,680	8,000	3,300	8,000	......	......	11,300
2½	4,700	3,040	......	5,300	3,470	5,453	6,700	3,800	6,700	9,400	4,900	9,510	13,400	5,450	13,400	......	......	18,925
3	7,500	5,840	......	8,500	6,250	8,710	10,700	7,000	10,700	15,000	9,000	15,190	21,400	10,000	21,400	......	......	30,230
3½	11,000	7,880	......	13,200	8,800	13,020	16,000	10,000	16,000	22,000	12,900	22,710	32,000	14,300	32,000	......	......	45,200
4	15,500	11,700	......	18,300	13,400	17,910	22,000	15,000	22,000	31,000	19,300	31,220	44,000	21,500	44,000	......	......	62,180
5	......	......	......	......	......	31,500	......	......	38,700	......	......	54,920	......	......	77,400	......	......	109,300
6	......	......	......	......	......	50,450	......	......	62,000	......	......	88,000	......	......	124,000	......	......	175,100
Risers																		
¾	......	190	......	......	190	570	......	190	700	......	190	994	......	190	1,400	......	......	1,977
1	......	450	......	......	450	976	......	450	1,200	......	450	1,704	......	450	2,400	......	......	3,390
1¼	......	990	......	......	990	1,547	......	990	1,900	......	990	2,696	......	990	3,800	......	......	5,370
1½	......	1,500	......	......	1,500	3,256	......	1,500	4,000	......	1,500	5,680	......	1,500	8,000	......	......	11,300
2	......	3,000	......	......	3,000	5,453	......	3,000	6,700	......	3,000	9,510	......	3,000	13,400	......	......	18,925
2½	......	......	......	......	......	8,710	......	......	10,700	......	......	15,910	......	......	21,400	......	......	30,230
3	......	......	......	......	......	13,020	......	......	16,000	......	......	22,710	......	......	32,000	......	......	45,200
3½	......	......	......	......	......	17,910	......	......	22,000	......	......	31,220	......	......	44,000	......	......	62,180
4	......	......	......	......	......	31,500	......	......	38,700	......	......	54,920	......	......	77,400	......	......	109,300
5	......	......	......	......	......	50,450	......	......	62,000	......	......	88,000	......	......	124,000	......	......	175,100

Equivalent Length of Run

All tables for the flow of steam in pipes, based on pressure drop, must allow for the friction offered by the pipe as well as for the additional resistance of the fittings and valves. These resistances generally are stated in terms of straight pipe; in other words, a certain fitting will produce a drop in pressure equivalent to so many feet of straight run of the same size of pipe. Table 5 gives the number of feet of straight pipe usually allowed for the more common types of fittings and valves. In all pipe sizing tables in this chapter the *length of run* refers to the *equivalent length of run* as distinguished from the *actual length* of pipe in feet. The length of run is not usually known at the outset; hence it is necessary to assume some pipe size at the start. Such an assumption frequently is considerably in error and a more common and practical method is to assume the length of run and to check this assumption after the pipes are sized. For this purpose the length of run usually is taken as double the actual length of pipe.

TABLES FOR PIPE SIZING[2]

Factors determining the size of a steam pipe and its allowable limit of capacity are the direction of the flow of condensate, whether against or with the steam.

Tables 6 and 7 are based on the actual inside diameters of the pipe and the condensation of ¼ lb (4 oz) of steam per square foot of equivalent direct radiation[3] (*abbreviated EDR*) per hour. The drops indicated are drops in pressure per 100 ft of equivalent length of run. The pipe is assumed to be well reamed without unusual or noticeable defects.

Table 6 may be used for sizing piping for steam heating systems by determining the allowable or desired pressure drop per 100 equivalent feet of run and reading from the column for that particular pressure drop. This applies to all steam mains on both one-pipe and two-pipe systems, vapor systems, and vacuum systems. Columns *B* to *G*, inclusive, are used where the steam and condensation flow in the same direction, while Columns *H* and *I* are for cases where the steam and condensation flow in opposite directions, as in risers and runouts that are not dripped. Columns *J*, *K*, and *L* are for one-pipe systems and cover riser, radiator valve, and vertical connection sizes, and radiator and runout sizes, all of which are based on the critical velocities of the steam to permit the counter flow of condensation without noise.

Sizing of return piping may be done with the aid of Table 7 where pipe capacities for wet, dry, and vacuum return lines are shown for the pressure drops per 100 ft corresponding to the drops in Table 6. *It is customary to use the same pressure drop on both the steam and return sides of a system.*

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[2]Pipe size tables in this chapter have been compiled in simplified and condensed form for the convenience of the user; at the same time all of the information contained in previous editions of THE GUIDE has been retained. Values of pressure drops, formerly expressed in ounces, are now expressed in fractions of a pound.

[3]As steam system design has materially changed in recent years so that 240 Btu no longer expresses the heat of condensation from a square foot of radiator surface per hour, and as present day heating units have different characteristics from older forms of radiation, it is the purpose of THE GUIDE to gradually eliminate the empirical expression *square foot of equivalent direct radiation*, *EDR*, and to substitute a logical unit based on the Btu. The new terms to express the equivalent of 1000 Btu (Mb), and 1000 Btu per hour (Mbh), have been approved by the A.S.H.V.E.

Example 2. What pressure drop should be used for the steam piping of a system if the measured length of the longest run is 500 ft and the initial pressure is not to be over 2-lb gage?

Solution. It will be assumed, if the measured length of the longest run is 500 ft, that when the allowance for fittings is added the equivalent length of run will not exceed 1,000 ft. Then, with the pressure drop not over one half of the initial pressure, the drop could be 1 lb or less. With a pressure drop of 1 lb and a length of run of 1,000 ft, the drop per 100 ft would be 1/10 lb, while if the total drop were 1/2 lb, the drop per 100 ft would be 1/20 lb. In the first instance the pipe could be sized according to Column *D* for 1/16 lb per 100 ft, and in the second case, the pipe could be sized according to Column *C* for 1/24 lb. On completion of the sizing, the drop could be checked by taking the longest line and actually calculating the equivalent length of run from the pipe sizes determined. If the calculated drop is less than that assumed, the pipe size is all right; if it is more, it is probable that there are an unusual number of fittings involved, and either the lines must be straightened or the column for the next lower drop must be used and the lines resized. Ordinarily resizing will be unnecessary.

ONE-PIPE GRAVITY AIR-VENT SYSTEMS

One-pipe gravity air-vent systems in which the equivalent length of run does not exceed 200 ft should be sized as follows:

1. *For the steam main and dripped runouts to risers* where the steam and condensate flow in the same direction, use 1/16-lb drop (Column *D*).

2. *Where the riser runouts are not dripped* and the steam and condensation flow in opposite directions, and *also in the radiator runouts* where the same condition occurs, use Column *L*.

3. *For up-feed steam risers* carrying condensation back from the radiators, use Column *J*.

4. *For down-feed systems the main risers* of which do not carry any radiator condensation, use Column *H*.

5. *For the radiator valve size and the stub connection*, use Column *K*.

6. For the *dry return main*, use Column *U*.

7. For the *wet return main* use Column *T*.

On systems exceeding an equivalent length of 200 ft, it is suggested that the total drop be not over 1/4 lb. The return piping sizes should correspond with the drop used on the steam side of the system. Thus, where 1/24-lb drop is being used, the steam main and dripped runouts would be sized from Column *C;* radiator runouts and undripped riser runouts from Column *L;* up-feed risers from Column *J*; the main riser on a down-feed system from Column *C* (it will be noted that if Column *H* is used the drop would exceed the limit of 1/24 lb); the dry return from Column *R;* and the wet return from Column *Q*.

With a 1/32-lb drop the sizing would be the same as for 1/24 lb except that the steam main and dripped runouts would be sized from Column *B*, the main riser on a down-feed system from Column *B*, the dry return from Column *O*, and the wet return from Column *N*.

Notes on Gravity One-Pipe Air-Vent Systems

1. Pitch of mains should not be less than 1/4 in. in 10 ft.

2. Pitch of horizontal runouts to risers and radiators should not be less than 1/2 in. in 10 ft. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

3. In general, it is not desirable to have a main less than 2 in. The diameter of the far end of the supply main should not be less than half its diameter at its largest part.

4. Supply mains, branches to risers, or risers, should be dripped where necessary.
5. Where supply mains are decreased in size they should be dripped, or be provided with eccentric couplings, flush on bottom.

Example 3. Size the one-pipe gravity steam system shown in Fig. 2 assuming that this is all there is to the system or that the riser and run shown involve the longest run on the system.

Solution. The total length of run actually shown is 215 ft. If the equivalent length of run is taken at double this, it will amount to 430 ft, and with a total drop of $\frac{1}{4}$ lb the drop per 100 ft will be slightly less than $\frac{1}{16}$ lb. It would be well in this case to use $\frac{1}{24}$ lb, and this would result in the theoretical sizes indicated in Table 8. These theoretical sizes, however, should be modified by not using a wet return less than 2 in. while the main supply, *g-h*, if from the uptake of a boiler, should be made the full size of the main, or 3 in. Also the portion of the main *k-m* should be made 2 in. if the wet return is made 2 in.

TABLE 8. PIPE SIZES FOR ONE-PIPE UP-FEED SYSTEM SHOWN IN FIG. 2

PART OF SYSTEM	SECTION OF PIPE	RADIATION SUPPLIED (SQ FT)	THEORETICAL PIPE SIZE (INCHES)	PRACTICAL PIPE SIZE (INCHES)
Branches to radiators	....	100	2	2
Branches to radiators	....	50	$1\frac{1}{4}$	$1\frac{1}{4}$
Riser	*a* to *b*	200	2	2
Riser	*b* to *c*	300	$2\frac{1}{2}$	$2\frac{1}{2}$
Riser	*c* to *d*	400	$2\frac{1}{2}$	$2\frac{1}{2}$
Riser	*d* to *e*	500	3	3
Riser	*e* to *f*	600	3	3
Branch to riser	*f* to *g*	600	$3\frac{1}{2}$	$3\frac{1}{2}$
Supply main	*g* to *h*	600	3	3
Branch to supply main	*h* to *j*	600	$2\frac{1}{2}$	3
Dry return main	*f* to *k*	600	$1\frac{1}{4}$	2
Wet return main	*k* to *m*	600	1	2
Wet return main	*m* to *n*	600	1	2
Wet return main	*n* to *p*	600	1	2

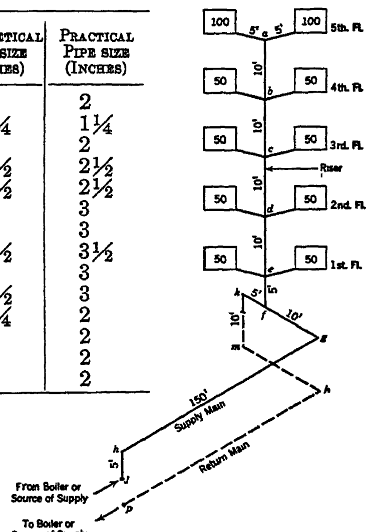


FIG. 2. RISER, SUPPLY MAIN AND RETURN MAIN OF ONE-PIPE SYSTEM

TWO-PIPE GRAVITY AIR-VENT SYSTEMS

The method employed in determining pipe sizes for two-pipe gravity air-vent systems is similar to that described for one-pipe systems except that the steam mains never carry radiator condensation. The drop allowable per 100 ft of equivalent run is obtained by taking the equivalent length to the farthest radiator as double the actual distance, and then dividing the allowable or desired total drop by the number of hundreds of feet in the equivalent length. Thus in a system measuring 400 ft from the boiler to the farthest radiator, the approximate equivalent length of run would be 800 ft. With a total drop of $\frac{1}{2}$ lb the drop per 100 ft would be $\frac{\frac{1}{2}}{8}$ or $\frac{1}{16}$ lb; therefore, Column *D* would be used for all steam mains where the condensation and steam flow in the same direction. If a total drop of $\frac{1}{4}$ lb is desired, the drop per 100 ft would be $\frac{1}{32}$ lb

and Column *B* would be used. If the total drop were to be 1 lb, the drop per 100 ft would be $\frac{1}{8}$ lb and Column *E* would be used.

For mains and riser runouts that are not dripped, and for radiator runouts where in all three cases the condensation and steam flow in opposite directions, Column *I* should be used, while for the steam risers Column *H* should be used unless the drop per 100 ft is $\frac{1}{24}$ lb or $\frac{1}{32}$ lb, when Columns *B* or *C* should be substituted so as not to exceed the drop permitted.

On an overhead down-feed system the main steam riser should be sized by reference to Column *H*, but the down-feed steam risers supplying the radiators should be sized by the appropriate Columns *B* through *G*, since the condensation flows downward with the steam through them. The riser runouts, if pitched down toward the riser as they should be, are sized the same as the steam mains, and the radiator runouts are made the same as in an up-feed system.

In either up-feed or down-feed systems the returns are sized in the same manner and on the same pressure drop basis as the steam main; the return mains are taken from Columns *O*, *R*, *U*, *X*, or *AA* according to the drop used for the steam main; and the risers are sized by reading the lower part of Table 7 under the column used for the mains. The horizontal runouts from the riser to the radiator are not usually increased on the return lines although there is nothing incorrect in this practice. The same notes apply that are given for one-pipe gravity systems.

TWO-PIPE VAPOR SYSTEMS

While many manufacturers of patented vapor heating accessories have their own schedules for pipe sizing, an inspection of these sizing tables indicates that in general as small a drop as possible is recommended. The reasons for this are: (1) to have the condensation return to the boiler by gravity, (2) to obtain a more uniform distribution of steam throughout the system, especially when it is desirable to carry a moderate or low fire, and (3) because with large variation in pressure the value of graduated valves on radiators is destroyed.

For small vapor systems where the equivalent length of run does not exceed 200 ft, it is recommended that the main and any runouts to risers that may be dripped should be sized from Column *D*, while riser runouts not dripped and radiator runouts should employ Column *I*. The up-feed steam risers should be taken from Column *H*. On the returns, the risers should be sized from Column *U* (lower portion) and the mains from Column *U* (upper portion). It should again be noted that the pressure drop in the steam side of the system is kept the same as on the return side except where the flow in the riser is concerned.

On a down-feed system the main vertical riser should be sized from Column *H*, but the down-feed risers can be taken from Column *D* although it so happens that the values in Columns *D* and *H* correspond. This will not hold true in larger systems.

For vapor systems over 200 ft of equivalent length, the drop should not exceed $\frac{1}{8}$ lb to $\frac{1}{4}$ lb, if possible. Thus, for a 400 ft equivalent run the drop per 100 ft should be not over $\frac{1}{8}$ lb divided by 4, or $\frac{1}{32}$ lb. In this case the steam mains would be sized from Column *B*; the radiator and

undripped riser runouts from Column *I*; the risers from Column *B*, because Column *H* gives a drop in excess of 1/32 lb. On a down-feed system, Column *B* would have to be used for both the main riser and the smaller risers feeding the radiators in order not to increase the drop over 1/32 lb. The return risers would be sized from the lower portion of Column *O* and the dry return main from the upper portion of the same column, while any wet returns would be sized from Column *N*. The same pressure drop is applied on both the steam and the return sides of the system.

Notes on Vapor Systems

1. Pitch of mains should not be less than 1/4 in. in 10 ft.

2. Pitch of horizontal runouts to risers and radiators should not be less than 1/2 in. in 10 ft. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

3. In general it is not desirable to have a supply main smaller than 2 in.

4. When necessary, supply main, supply risers, or branches to supply risers should be dripped separately into a wet return, or may be connected into the dry return through a thermostatic drip trap.

VACUUM, ORIFICE, SUB-ATMOSPHERIC SYSTEMS

Vacuum, atmospheric, sub-atmospheric and orifice systems are usually employed in large installations and have total drops varying from 1/4 to 1/2 lb. Systems where the maximum equivalent length does not exceed 200 ft preferably employ the smaller pressure drop while systems over 200 ft equivalent length of run more frequently go to the higher drop, owing to the relatively greater saving in pipe sizes. For example, a system with 1200 ft longest equivalent length of run would employ a drop per 100 ft of 1/2 lb divided by 12, or 1/24 lb. In this case the steam main would be sized from Column *C*, and the risers also from Column *C* (Column *H* could be used as far as critical velocity is concerned but the drop would exceed the limit of 1/24 lb). Riser runouts, if dripped, would use Column *C* but if undripped would use Column *I*; radiator runouts, Column *I*; return risers, lower part of Column *S*; return runouts to radiators, one pipe size larger than the radiator trap connections.

Notes on Vacuum Systems

1. It is not generally considered good practice to exceed 1/8 lb drop per 100 ft of equivalent run nor to exceed 1 lb total pressure drop in any system.

2. Pitch of mains should not be less than 1/4 in. in 10 ft.

3. Pitch of horizontal runouts to risers and radiators should not be less than 1/2 in. in 10 ft. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

4. In general it is not considered desirable to have a supply main smaller than 2 in.

5. When necessary, the supply main, supply riser, or branch to a supply riser should be dripped separately through a trap into the vacuum return. A connection should not be made between the steam and-return sides of a vacuum system without interposing a trap to prevent the steam from entering the return line.

6. Lifts should be avoided if possible, but when they cannot be eliminated they should be made in the manner described in Chapter 14.

7. No lifts can be used in orifice and atmospheric systems. In sub-atmospheric systems the lift must be at the vacuum pump.

BOILER CONNECTIONS

Steam

Cast-iron, sectional heating boilers usually have several outlets in the top. Two or more outlets are sometimes used to reduce the velocity of the steam in the vertical uptakes from the boiler and thus to prevent water being carried over into the steam main.

Return

Cast-iron boilers are generally provided with return tappings on both sides, while steel boilers are generally equipped with only one return tapping. Where two tappings are provided, both should be used to effect proper circulation through the boiler. The return connection should

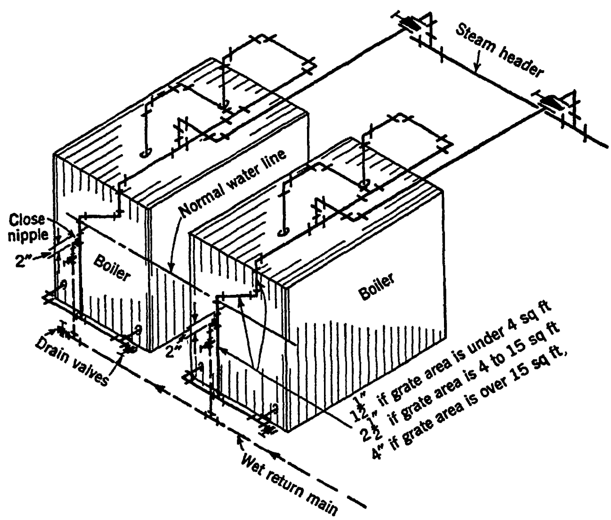


FIG. 3. THE HARTFORD RETURN CONNECTION

include either a Hartford Loop or a check valve to prevent the accidental loss of boiler water to the returns with consequent danger of boiler damage. The Hartford Loop connection is to be preferred over the check valve because the latter is apt to stick or not close tightly and, furthermore, because the check valve offers additional resistance to the condensate coming back to the boiler, which in gravity systems would raise the water line in the far end of the wet return several inches[4].

Hartford Return Connection

In order to prevent the boiler from losing its water under any circumstances, the use of the Hartford connection, or the Underwriters Loop, is recommended. This connection for a one- or two-boiler installation is shown in Fig. 3. The essential features of construction of a Hartford

[4]See method of calculating height above water line for gravity one-pipe systems in Chapter 14

Loop connection are: (1) a direct connection (made without valves) between the steam side of the boiler and the return side of the boiler, and (2) a close nipple connection about 2 in. below the normal boiler water line from the return main to the boiler steam and return pressure balance connection. Equalizing pipe connections between the steam and return are given in Fig. 3, based on grate areas, but in no case shall this pipe size be less than the main return piping from the system.

Sizing Boiler Connections

Little information is available on the sizing of boiler runouts and steam headers. Although some engineers prefer an enlarged steam header to serve as additional steam storage space, there ordinarily is no sudden demand for steam in a steam heating system except during the heating-up period, at which time a large steam header is a disadvantage rather than an advantage. The boiler header may be sized by first computing the maximum load that must be carried by any portion of the header under any conceivable method of operation, and then applying the same schedule of pipe sizing to the header as is used on the steam mains for the building. The horizontal runouts from the boiler, or boilers, may be sized by calculating the heaviest load that will be placed on the boiler at any time, and sizing the runout on the same basis as the building mains. The difference in size between the vertical uptakes from the boiler and the horizontal main or runout is compensated for by the use of reducing ells.

Return connections to boilers in gravity systems are made the same size as the return main itself. Where the return is split and connected to two tappings on the same boiler, both connections are made the full size of the return line. Where two or more boilers are in use, the return to each may be sized to carry the full amount of return for the maximum load which that boiler will be required to carry. Where two boilers are used, one of them being a spare, the full size of the return main would be carried to each boiler, but if three boilers are installed, with one spare, the return line to each boiler would require only half of the capacity of the entire system, or, if the boiler capacity were more than one-half the entire system load, the return would be sized on the basis of the maximum boiler capacity. As the return piping around the boiler is usually small and short, it should not be sized to the minimum.

With returns pumped from a vacuum or receiver return pump, the size of the line may be calculated from the water rate on the pump discharge when it is operating, and the line sized for a very small pressure drop, the size being obtained from the Chart for Pressure Drop for Various Rates of Flow of Water, Chapter 46. The relative boiler loads should be considered, as in the case of gravity return connections. Boiler header and piping sizes should be based on the total load.

HIGH PRESSURE STEAM

When high pressure steam is being supplied and lower steam pressures are required for use in heating, domestic hot water, utility services, etc., one or more pressure reducing valves, or pressure regulators, as they are sometimes called, are required.

These are used in two classes of service, one where the steam must be

shut off tight to prevent the low pressure building up at time of no load, and the other where the low pressure lines will condense enough steam to offset normal leaking through the valve. In the latter case, double seated valves may be used in a manner that reduces the work required of the diaphragm in closing the valve and consequently the size of the diaphragm. These valves also control the low pressures more closely under conditions of varying high pressures.

Valves that shut off all steam are called *dead end* type. They are single seated, and some of them have pilot operation that provides close control of the reduced pressure. If a thermostatically controlled valve is installed after, and near, a reducing valve in such a manner as to cut off the passage of steam, the dead end type should be used.

It is common practice when the initial steam pressure is 100 lb or higher to install two-stage reduction. This makes a quieter condition of steam flow, as it is apparent that with one reduction, as for example, from 150 to 2 lb, there is a smaller opening with greater velocity across the reducing valve, and consequently, more noise. A two-stage reduction also introduces a source of safety, since if one reducing valve were to build up its discharge pressure, this excess pressure would not be as great as the case might be in a one-stage reduction.

If an installation requires single seated valves, and the pilot type cannot be used, it is necessary to use two-stage reduction, as single seated valves require sufficient diaphragm area to overcome the unbalanced pressure underneath the single valve. In many cases the large diameter of diaphragm required would make it impractical in construction. With a two-stage reduction the diaphragm diameter required would be reduced. If a one-stage reduction is desired, it is necessary to use a pilot controlled pressure reducing valve, where low pressures are to be maintained closely.

In making two-stage reduction, allowance should be made, by increasing the pipe size, for expansion of steam on the low pressure side of the valve. This also allows steam flow to be at a more nearly uniform velocity. Separating the valves by a distance up to 20 ft is recommended to reduce excessive hunting action of the first valve.

When the reduced pressure is approximately 15 lb or lower, the weight and lever diaphragm valve gives the best results with minimum maintenance. Above 15 lb, spring loaded diaphragm valves should be used, because of the extra weights required on weight and lever type. Equalizing line connections should be made not too close to the valve, and into the bottom of the reduced pressure steam main, to allow maximum condensation to exist in this equalizing line, or the connection is made into the top of the main and a water accumulator used to reduce the variation of the head of water on the diaphragm.

Care should be exercised in selecting the size of a reducing valve. The safest method is to consult the manufacturer. It is essential that sizes of piping to and from the reducing valve be such that they will pass the desired amount of steam with the maximum velocity desired. A common error is to make the size of the reducing valve the same size as that of the service, or outlet pipe size. Generally, this will make the reducing valve oversized, and bring about wire-drawing of valve and seat, due to small lift of the valve seat.

On installations where the steam requirements are relatively large and variable in mild weather or reduced demand periods, wire-drawing may occur. To overcome this condition, two reducing valves are installed in parallel, with the sizes selected on a 70 and 30 per cent proportion of maximum flow. For example, if 50,000 lb of steam per hour are required, the size of one valve is on the basis of 0.7 + 50,000 lb, or 35,000 lb, and the other on the basis of 0.3 + 50,000 lb, or 15,000 lb. During the mild or reduced demand periods, steam will flow through the smaller valve only. During the remainder of the season, the larger valve is set to control at whatever low pressure is desired, and the smaller one at a somewhat lower pressure. Thus, when steam flow is not at its maximum, the

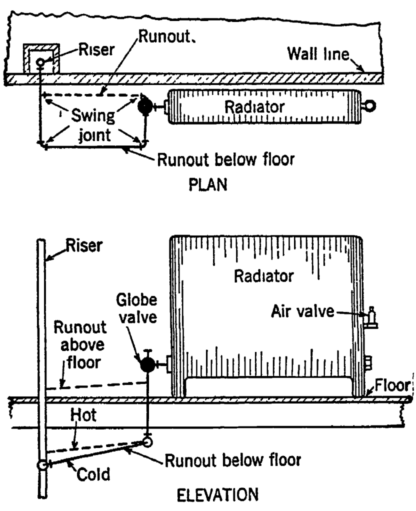


Fig. 4. One-Pipe Radiator Connections

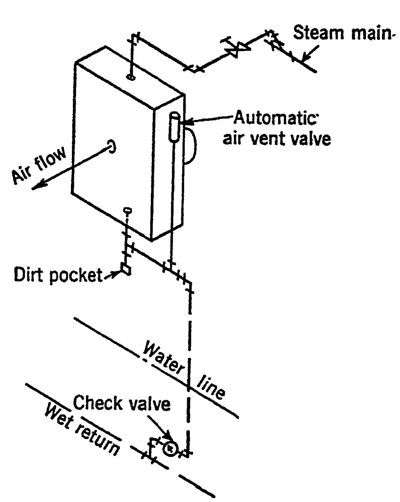


Fig. 5. Unit Heater Connected to One-Pipe Air-Vent System

smaller valve is shut, and automatically opens when the maximum steam demand occurs, since this maximum demand of steam creates a slight pressure drop in the service line.

The installation of reducing valves in pipe lines requires detailed planning. They should be installed to give ease of access for inspection and repair, and wherever possible with diaphragm downward, except in cases of pilot operated valves.

There should be a by-pass around each reducing valve of size equal to one half the size of reducing valve. The globe valve in by-pass line should be of a better type of construction, and must shut off absolutely tight. A steam pressure gage, graduated up to the initial pressure should be installed on the low pressure side. Safety valves located on the low pressure side should be set 5 lb higher than the final pressure but may be 10 lb higher than the reduced pressure if this reduced pressure is that of the first stage reduction of a double reduction. Strainers should always be installed on the inlet to the reducing valve but are not required before

a second-stage reduction. If a two-stage reduction is made, it is well to install a pressure gage immediately before the reducing valve of the second-stage reduction also. In sizes 3 in. and above, it is advisable to tap the bodies of the reducing valve on inlet side for purposes of draining condensate accumulation through steam traps.

Control Valves

Gate valves are recommended in all cases where service demands that the valve be either entirely open or entirely closed, but they should never

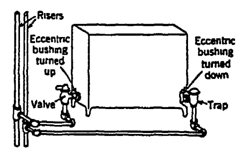


FIG. 6. TYPICAL CONNECTIONS FOR TWO-PIPE SYSTEM

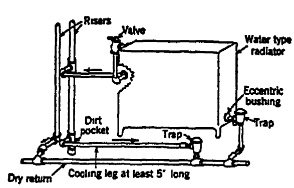


FIG. 7. TOP AND BOTTOM OPPOSITE END RADIATOR CONNECTIONS

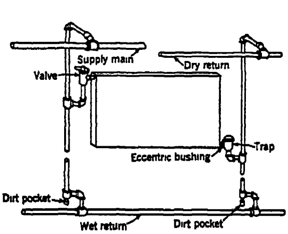


FIG. 8. CONNECTIONS TO RADIATOR HUNG ON WALL

be used for throttling. Angle globe valves and straight globe valves should be used for throttling, as done on by-passes around pressure reducing valves or on by-passes around traps.

Connection to Heating Units

Riser, radiator and convector connections must not only be properly pitched at the time they are installed but must be arranged so that the pitch will be maintained under the strains of expansion and contraction. These connections may be made by swing joints which permit the expansion or contraction to occur under heating and cooling without bending of pipes. To take care of expansion in long risers, either expansion joints of commercial construction or pipe swing joints are used. Anchoring of pipes between expansion joints is desirable.

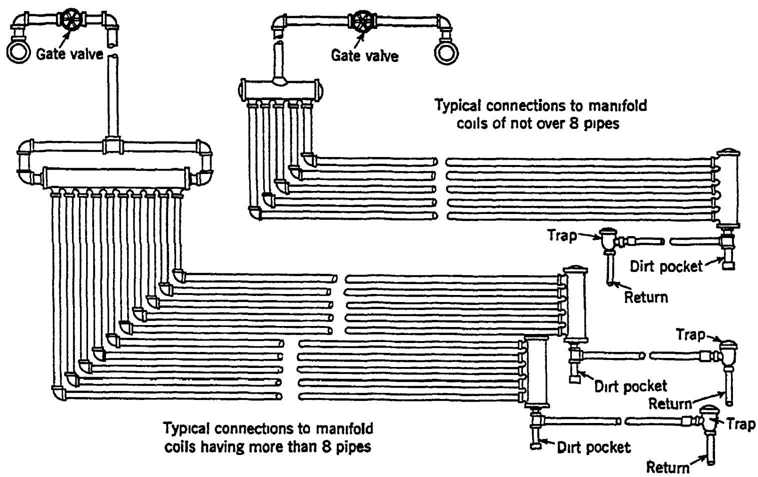


FIG. 9. TYPICAL PIPE COIL CONNECTIONS

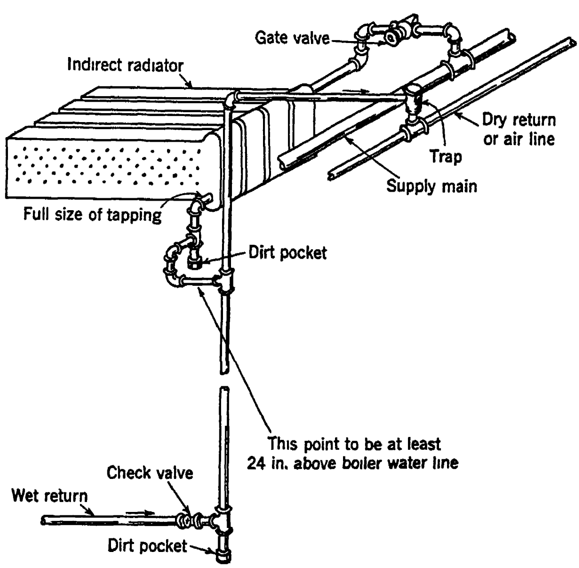


FIG. 10. TYPICAL PIPING CONNECTIONS TO CONCEALED HEATING UNITS WITH WET RETURNS

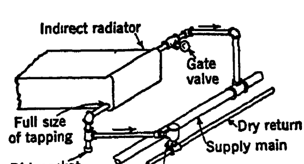


FIG. 11. PIPING CONNECTIONS TO INDIRECT RADIATORS

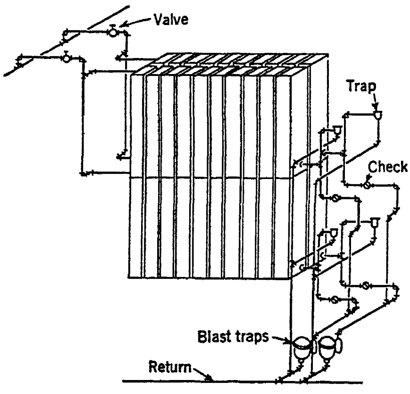


FIG. 12. SUPPLY AND RETURN CONNECTIONS FOR HEATING UNITS OF CENTRAL FAN SYSTEMS

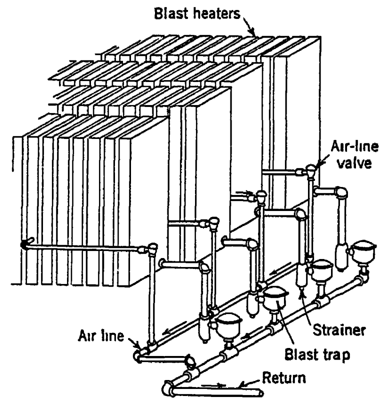


FIG. 13. TYPICAL CONNECTIONS TO CENTRAL FAN SYSTEM HEATING UNITS EXCEEDING 12 SECTIONS

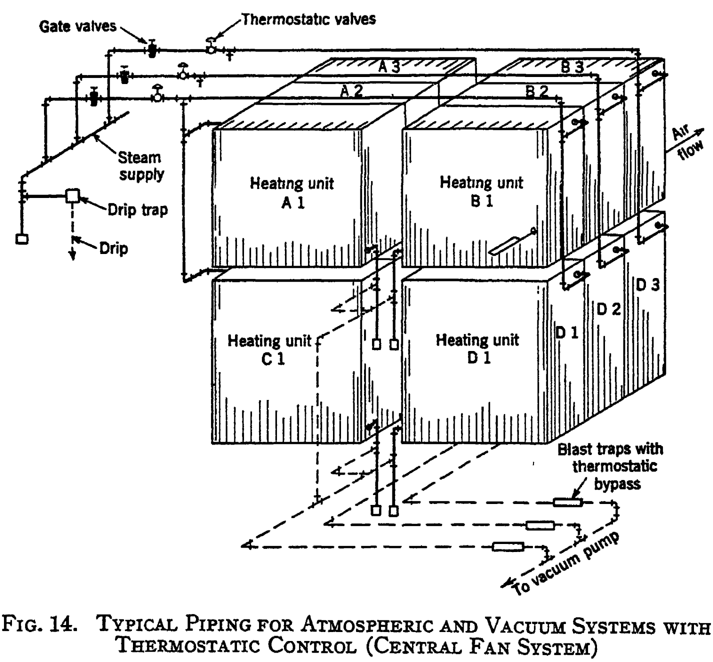


FIG. 14. TYPICAL PIPING FOR ATMOSPHERIC AND VACUUM SYSTEMS WITH THERMOSTATIC CONTROL (CENTRAL FAN SYSTEM)

Two satisfactory methods of making runouts for one-pipe systems for either the up-feed or the down-feed type are shown in Fig. 4. Where the vertical distance is limited and the runouts must run above the floor the radiator may be set on pedestals or of the high leg type. A method of connecting a unit heater to a one-pipe steam heating system is illustrated in Fig. 5.

Typical two-pipe radiator or convector connections are shown in Figs. 6, 7 and 8. While the top is the preferred location for the control valve, it may be located at the bottom. Short radiators may be top supply and bottom return on same end. With convectors the control valve is sometimes omitted and a damper in outlet grille used for heat control. The typical method of connecting pipe coils is shown in Fig. 9 and is suitable for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

Typical pipe connections for indirect radiators and tempering or heating stacks are shown in Figs. 10, 11, 12 and 13.

Where a building is served by a vacuum system or a sub-atmospheric system the stacks should be piped in the usual manner and traps of large capacity, preferably of the combination float and thermostatic type, should be used. In the orifice and *closed* two-pipe systems, traps should be used on the returns so that a pressure above that of the atmosphere may be secured on the heaters.

Each stack should have a separate steam and return connection. Wide stacks are more evenly heated with two steam connections, one at each end, the stacks being divided and a return connection provided for each steam connection. For stacks of large capacity it is sometimes desirable to run a separate steam main direct from the boiler to the stacks.

PIPE SIZING FOR INDIRECT HEATING UNITS

Pipe connections and mains for indirect heating units are sized in a manner similar to radiators, but the equivalent direct radiation must be ascertained for each row of heating unit stacks and then must be divided into the number of stacks constituting that row and into the number of connections to each stack.

$$EDR = \frac{Q \times 60 \times (t_l - t_e)}{55.2 \times 240} = \frac{Q \times (t_l - t_e)}{220.8} \quad (1)$$

where

EDR = equivalent direct radiation, square feet.

Q = volume of air, cubic feet per minute.

t_e = the temperature of the air entering the row of heating units under consideration, degrees Fahrenheit.

t_l = the temperature of the air leaving the row of heating units under consideration, degrees Fahrenheit.

60 = the number of minutes in one hour.

55.2 = the number of cubic feet of air heated 1 F by 1 Btu.

240 = the number of Btu in 1 sq ft of EDR.

Example 4. Assume that the heating units shown in Fig. 14 are handling 50,000 cfm of air and that the rise in the first row is from 0 to 40 F, in the second row from 40 to 65 F, and in the third row from 65 to 80 F. What is the load in EDR on each supply and return connection?

Solution. For row 1,

$$R = \frac{50{,}000 \times (40 - 0)}{220.8} = 9058 \text{ sq ft.}$$

For row 2,

$$R = \frac{50{,}000 \times (65 - 40)}{220.8} = 5661 \text{ sq ft.}$$

For row 3,

$$R = \frac{50{,}000 \times (80 - 65)}{220.8} = 3397 \text{ sq ft.}$$

Each row of heating units consists of four stacks and each stack has two connections so that the load on each stack and each connection of the stack is as follows:

Row	Total Load (EDR)	Stack Load[a] (EDR)	Connection Load[b] (EDR)
1	9058	2265	2265 or 1132
2	5661	1415	1415 or 708
3	3397	849	849 or 425

[a]One quarter of total row load.
[b]One half of stack load if two steam connections are made; otherwise, same as stack load.

The pipe sizes would then be based on the length of the run and the pressure drop desired, as in the case of radiators. It generally is considered desirable to place the indirect heating units on a separate system and not on supply or return lines connected to the general heating system.

DRIPS

A steam main in any type of steam heating system may be dropped to a lower level without dripping if the pitch is downward with the direction of steam flow. Any steam main in any heating system can be elevated if dripped. Fig. 15 shows a connection where the steam main is raised and the drain is to a wet return. If the elevation of the low point is above a dry return it may be drained through a trap to the dry return in two-pipe vapor, vacuum and sub-atmospheric systems. Horizontal steam pipes may also be run over obstructions without a change in level if a small pipe is carried below the obstruction to care for the condensation (Fig. 16). Horizontal return pipes may be carried past doorways and other obstructions by using the scheme illustrated in Fig. 17. It will be noted that the large pipe, in this case, runs below the obstruction and the smaller one over it.

Branches from steam mains in one-pipe gravity steam systems should use the *preferred connection* shown in Fig. 18, but where radiator condensation does not flow back into the main the *acceptable* method shown in the same figure may be used. This acceptable method has the advantage of giving a perfect swing joint when connected to the vertical riser or radiator connection, whereas the preferred connection does not give this swing

without distorting the angle of the pipe. Runouts from the steam main are usually made about 5 ft long to provide flexibility for movement in the main.

Offsets in steam and return piping should preferably be made with 90-deg ells but occasionally fittings of other angles are used, and in such cases the length of the diagonal offset will be found as shown in Fig. 19.

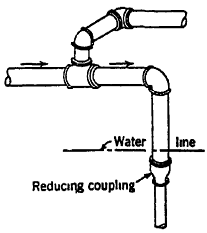


Fig. 15. Dripping Main Where it Rises to Higher Level

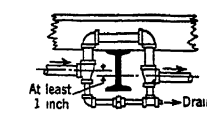


Fig. 16. Looping Main Around Beam

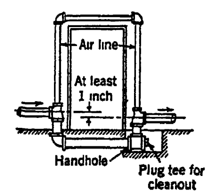


Fig. 17. Looping Dry Return Main Around Opening

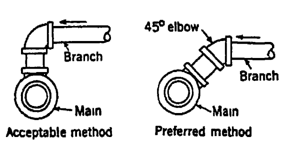
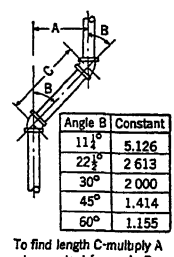


Fig. 18. Methods of Taking Branch from Main



Angle B	Constant
11¼°	5.126
22½°	2.613
30°	2.000
45°	1.414
60°	1.155

To find length C-multiply A by constant for angle B

Fig. 19. Constants for Determining Length Offset Pipe

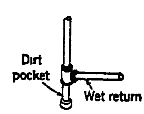


Fig. 20. Dirt Pocket Connection

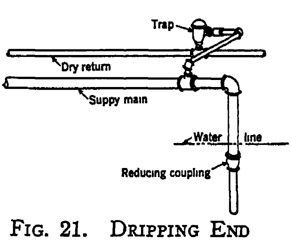


Fig. 21. Dripping End of Main into Wet Return

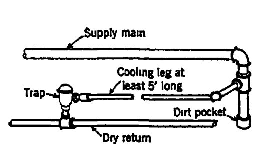


Fig. 22. Dripping End of Main into Dry Return

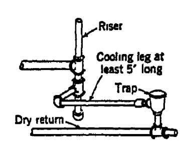


Fig. 23. Dripping Heel of Riser into Dry Return

Dirt pockets, desirable on all systems employing thermostatic traps, should be so located as to protect the traps from scale and muck which will interfere with their operation. Dirt pockets are usually made 8 in. to 12 in. deep and serve as receivers for foreign matter which otherwise would be carried into the trap. They are constructed as shown in Fig. 20.

On vapor systems where the end of the steam main is dripped down into the wet return, the air venting at the end of the main is accomplished

by an air vent passing through a thermostatic trap into the dry return line as shown in Fig. 21. On vacuum systems the ends of the steam mains are dripped and vented into the return through drip traps opening into the return line. The same method may be used in atmospheric systems. A float type trap is preferable to a thermostatic trap for dripping steam mains and large risers. If thermostatic traps are used, a cooling leg (Fig. 22) should always be provided. The cooling leg is for cooling the condensation sufficiently before it reaches the trap so the trap will not be held shut by too high a temperature. On down-feed systems of atmospheric, vapor, and vacuum types, the bottom of the steam risers are dripped in the manner shown in Fig. 23. On large systems it is desirable to install a gate valve in the cooling leg ahead of the trap.

Chapter 16

HOT WATER HEATING SYSTEMS AND PIPING

One- and Two-Pipe Systems, Selecting Pipe Sizes, Forced Circulation, Gravity Circulation, Expansion Tanks, Installation Details

SYSTEMS for heating with hot water radiators may be divided into two general classes, the first known as gravity systems in which circulation is caused by the difference in density of the water in the flow and return lines, and the second known as forced circulation systems in which circulation is caused by a pump. Flow water temperatures vary from 150 to 220 F and the higher temperatures are generally used with the forced system.

For the sizing and selecting of boilers, radiators and piping, refer to Chapters 12, 13 and 18 respectively.

SYSTEMS OF PIPING

There are two general systems of piping used for either gravity or forced hot water systems:

(a) Two-pipe system.
(b) One-pipe system.

With either of these piping systems the distributing mains may be located in the cellar with up-feed to the radiators and risers or the supply main may be located in the attic with the return main located in the cellar. With the latter system of piping, the one-pipe system would be only one pipe for the risers. For basement radiators located on the floor the mains may be run at the ceiling, as one of the advantages of a forced hot water heating system is that the returns need not be below the radiators as required with a steam system. With some one-pipe systems there is one main in the cellar but separate flow and return risers and connections to the radiators.

In the two-pipe system there is separate supply and return pipes throughout so that the same water flows only through one radiator, resulting in the same water temperature in all radiators. With the one-pipe system part of the water flows through more than one radiator, so that the water temperature toward the end of the main is not as hot as near the boiler. However, with the one-pipe system, by maintaining a rapid circulation and small difference in temperature between the water leaving and returning to the boiler or other heat generator, the tendency to have variable temperatures in the radiators is much reduced.

The two-pipe system for larger buildings should, if possible, be arranged for reversed return. The direct and reversed return systems are shown

in Figs. 1 and 2. With the reversed return system, the water has to travel approximately the same distance from and back to the boiler for any one radiator as for any other radiator and, therefore, the friction and temperature losses to all radiators should be nearly the same.

In some cases the reversed return system involves no more piping than the direct return system. In the case of large buildings, it might be advisable to zone the piping.

Mechanical Circulators

Circulating pumps are usually of the centrifugal type. The capacity of the pump is figured from the Mbh (symbol representing 1000 Btu per hour) required for heating and the drop in temperature selected. For example, for 100 Mbh and 20 F drop a pump having a capacity of 5000 lb water per hour or 10 gpm. The resistance head is based on the system as designed. In large systems the economical size of pump may be determined by comparing the cost of power for operation, with the annual charges on the capital cost of the piping system, as larger pipe sizes mean less pump power. Velocities through piping in excess of 4 fps are likely to cause disturbing noises in buildings other than factories. In large

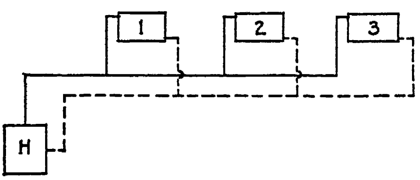


Fig. 1. A Direct Return System

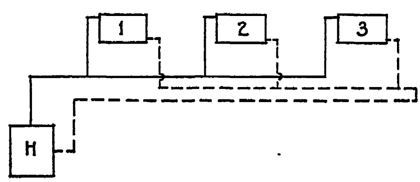


Fig. 2. A Reversed Return System

systems the pumps are run continuously while in small ones they are run either continuously or intermittently depending on the type of automatic temperature control selected. Small circulating pumps are usually driven by direct-connected electric motors. Under certain conditions a valved by-pass should be provided and the piping so designed that in case of breakdown of the pump or failure of electric current there will be sufficient gravity circulation to keep the building reasonably warm. In large buildings or groups of buildings, it is often advisable to have two pumps, each of about 70 per cent of the total capacity to take care of breakdown service. During mild weather, variations in water temperature may be utilized to balance the required heat loss. In the larger systems steam turbines are sometimes used to drive the pumps, the exhaust steam being used for heating the water, and in such buildings as hospitals this is usually the most economical method.

As the average pump used for water circulation is not over 60 per cent efficient, the cost of power on a large job should be figured and comparisons made between the savings made in capital cost of piping and the annual cost of power.

FORCED CIRCULATION PIPE SIZES

The pressure heads available in forced circulation systems are much greater than those in gravity circulation systems, consequently, higher

velocities may be used in designing the system, with the result that smaller pipes may be selected and the first cost of the installation reduced. As the pipe sizes of a heating system are reduced, the necessary increase in the velocity of the water increases the friction losses and thus the cost of operation and the initial cost of the circulating equipment. The increased velocity of a forced circulation system offers a number of advantages, such as a much shorter heating-up period and a more flexible control of hot water circulation. This improved performance merits the small increase in operating cost necessary to circulate the water mechanically. The velocities required should be determined by calculation for the particular system under consideration.

Since forced circulation velocities are higher than those in gravity systems, and since the friction heads in a heating system vary almost as the squares of the velocities, a given error in the calculation or assumption of a velocity is less important in a forced circulation system than in a gravity circulation system, and, consequently, it is easier to design a satisfactory forced circulation system than a satisfactory gravity circulation system.

In forced hot water systems, it is customary to use a temperature drop of 20 or 30 F between the water entering and leaving the boiler or other heater. The head against which the system is to operate must then be decided. This varies from 2 to 5 ft for small systems and may rise to 100 ft on large jobs with a group of buildings. For iron pipe, the sizes can be figured using Fig. 3 and Tables 1 and 3. For copper tubing Tables 2 and 3 are to be used. In systems designed with reversed returns, it will generally be found that very little adjustment is necessary to secure even distribution to all radiators. However, orifices may be used to control the flow and the capacities are given in Table 5. In large buildings provision should be made for quickly draining radiators in case of breakage, and it is often advisable to install a lock shield valve on one end of each radiator and a hand controlled valve on the other. In case of breakage the two valves can be closed and the radiator removed without affecting the rest of the system. The lock shield valve can also be used for balancing the water circulation.

The following examples will illustrate the procedure to be followed in designing forced circulation systems.

Example 1. From the plan of Fig. 4 note that the longest circuit consists of 151 ft of iron pipe; 1 boiler; 1 radiator; 1 radiator valve; 1 stop cock; 10 ells and 3 tees; and the shortest circuit consists of 127 ft of pipe; 4 tees; 1 boiler; 1 radiator; 1 radiator valve; 1 stop cock; and 6 ells. Design the piping for this system.

Solution. The friction in the various fittings can be expressed in terms of the friction in a 90-deg elbow from the values given in Table 3. The longest circuit consists of 151 ft of pipe and 44 elbow equivalents. The short circuit consists of 127 ft of pipe and 39 elbow equivalents.

The friction head in one elbow is approximately equal to the friction produced by the same sized pipe 25 diameters in length. Assume that the average pipe size for this system is 1 in. The equivalent length of the longest circuit will be 151 ft plus 100 ft or 251 ft of pipe. The equivalent length of the short circuit will be 217 ft.

Having determined the equivalent length of the circuits, the next step is to assume the rate at which the water is to be circulated in the system. The water may flow through the system so that it will cool any reasonable number of degrees. For the most economical average system a 20 F drop seems to be a satisfactory rate. This entails a slower water flow from the pumping equipment with a reasonable relationship between pipe

Table 1. Equivalent Length of Pipe *vs.* Pressure Head at Various Friction Losses Steel Pipe

Head Loss, Ft	Milinch Friction Loss per Foot of Pipe													
	720	480	360	300	240	180	160	144	120	96	90	80	70	60
	Equivalent Length of Pipe in Feet													
2	33	50	66	80	100	133	150	167	200	250	270	300	340	400
2½	42	62	84	100	125	167	188	208	250	312	333	375	428	500
3	50	75	100	120	150	200	225	250	300	375	400	450	510	600
3½	59	87	117	140	175	233	263	291	350	437	463	525	593	700
4	67	100	133	160	200	266	300	333	400	500	533	600	685	800
4½	75	112	149	180	225	300	338	374	450	562	593	675	758	900
5	83	125	167	200	250	333	375	416	500	625	666	750	860	1000
5½	92	137	183	220	275	366	413	457	550	687	713	825	923	1100
6	100	150	200	240	300	400	450	500	600	750	800	900	1030	1200
6½	108	162	217	260	325	433	488	540	650	812	843	975	1088	1300
7	116	175	233	280	350	465	525	580	700	875	933	1050	1200	1400
7½	124	187	249	300	375	500	563	623	750	937	973	1125	1252	1500
8	133	200	266	320	400	533	600	666	800	1000	1070	1200	1370	1600
8½	142	212	283	340	425	566	638	706	850	1062	1103	1275	1417	1700
9	150	225	300	360	450	600	675	750	900	1125	1200	1350	1540	1800
9½	159	237	317	380	475	633	713	789	950	1187	1233	1425	1577	1900
10	167	250	333	400	500	666	750	833	1000	1250	1333	1500	1715	2000
10½	175	262	349	420	525	700	788	872	1050	1312	1363	1575	1737	2100
11	183	275	366	440	550	733	825	916	1100	1375	1466	1650	1885	2200
11½	192	287	383	460	575	766	863	955	1150	1437	1533	1725	1897	2300
12	200	300	400	480	600	800	900	1000	1200	1500	1600	1800	2030	2400

Nominal Pipe Size, In.		Capacity of Pipes *Mbh* with a 20 F[a] Drop *A = Carrying Capacity, B = Velocity, inches per second* Friction Head of Pipe Milinches per foot													
		720	480	360	300	240	180	160	144	120	96	90	80	70	60
½	A	20	16	14	13	11	10	9	9	8	7	7	6	6	5
	B	*27*	*22*	*19*	*17*	*15*	*13*	*12*	*11*	*10*	*9*	*9*	*8*	*8*	*7*
¾	A	43	35	30	27	24	21	19	18	17	15	14	13	12	11
	B	*33*	*25*	*23*	*21*	*18*	*16*	*15*	*14*	*13*	*11*	*11*	*10*	*9*	*9*
1	A	85	70	60	54	48	41	39	36	33	30	28	27	25	23
	B	*39*	*32*	*27*	*25*	*22*	*19*	*18*	*17*	*15*	*13*	*13*	*12*	*11*	*10*
1¼	A	180	145	125	115	98	85	80	75	68	60	58	55	51	47
	B	*48*	*39*	*33*	*30*	*27*	*23*	*21*	*20*	*19*	*16*	*15*	*15*	*14*	*12*
1½	A	285	230	195	180	160	135	125	120	110	96	92	88	82	75
	B	*54*	*44*	*38*	*34*	*30*	*26*	*24*	*23*	*21*	*19*	*18*	*17*	*15*	*14*
2	A	540	435	370	340	300	255	240	230	205	180	175	165	150	140
	B	*64*	*52*	*45*	*40*	*36*	*30*	*29*	*27*	*24*	*22*	*21*	*20*	*19*	*17*
2½	A	890	720	610	550	480	420	390	370	330	300	280	270	250	230
	B	*74*	*60*	*50*	*46*	*41*	*35*	*33*	*31*	*28*	*24*	*24*	*22*	*21*	*19*
3	A	1650	1340	1130	1000	900	760	720	670	600	540	520	480	450	410
	B	*88*	*70*	*60*	*54*	*48*	*41*	*38*	*36*	*33*	*29*	*28*	*26*	*24*	*22*
3½	A	2500	2000	1700	1500	1350	1150	1080	1000	900	800	760	720	670	620
	B	*99*	*78*	*66*	*60*	*54*	*46*	*43*	*40*	*36*	*32*	*31*	*29*	*27*	*25*
4	A	3500	2800	2400	2200	1900	1600	1520	1440	1300	1150	1100	1050	960	880
	B	*110*	*87*	*74*	*66*	*58*	*50*	*47*	*45*	*40*	*35*	*34*	*32*	*30*	*27*
5	A	7000	5600	4700	4300	3700	3200	3000	2750	2500	2200	2100	2000	1800	1700
	B	*132*	*106*	*90*	*80*	*70*	*60*	*56*	*53*	*48*	*42*	*41*	*38*	*35*	*32*
6	A	12,000	9200	7800	7000	6200	5200	4800	4600	4100	3600	3500	3300	3000	2800
	B	*156*	*124*	*104*	*94*	*82*	*69*	*64*	*61*	*55*	*48*	*46*	*44*	*41*	*37*

[a]For other temperature drops the pipe capacities may be changed correspondingly. For example, with a temperature drop of 30 F, the capacities shown in this table are to be multiplied by 1.5.

Table 2. Equivalent Length of Tube *vs.* Pressure Head at Various Friction Losses Type L Copper Tube

Head Loss, Ft	Milinch Friction Loss per Foot of Tube											
	720	600	480	360	300	240	180	150	120	90	75	60
	Equivalent Length of Tube in Feet (Longest Circuit)											
2	33	40	50	67	80	100	133	160	200	267	320	400
2½	42	50	63	83	100	125	167	200	250	333	400	500
3	50	60	75	100	120	150	200	240	300	400	480	600
3½	58	70	88	117	140	175	233	280	350	467	560	700
4	67	80	100	133	160	200	267	320	400	533	640	800
4½	75	90	113	150	180	225	300	360	450	600	720	900
5	83	100	125	167	200	250	333	400	500	667	800	1000
5½	92	110	138	183	220	275	367	440	550	733	880	1100
6	100	120	150	200	240	300	400	480	600	800	960	1200
6½	108	130	163	217	260	325	433	520	650	867	1040	1300
7	117	140	175	233	280	350	467	560	700	933	1120	1400
7½	125	150	188	250	300	375	500	600	750	1000	1200	1500
8	133	160	200	267	320	400	533	640	800	1067	1280	1600
8½	142	170	213	283	340	425	567	680	850	1133	1360	1700
9	150	180	225	300	360	450	600	720	900	1200	1440	1800
9½	159	190	238	317	380	475	633	760	950	1267	1520	1900
10	167	200	250	333	400	500	667	800	1000	1333	1600	2000
10½	175	210	263	350	420	525	700	840	1050	1400	1680	2100
11	183	220	275	367	440	550	733	880	1100	1467	1760	2200
11½	192	230	288	383	460	575	767	920	1150	1533	1840	2300
12	200	240	300	400	480	600	800	960	1200	1600	1920	2400

Nominal Tube Size, In.	Capacity of Tubes *Mbh* With a 20 F[a] Drop *A* = *Carrying Capacity*, *B* = *Velocity, inches per second* Friction Head of Pipe Milinches per Foot											
	720	600	480	360	300	240	180	150	120	90	75	60
⅜ A	10	9	8	6 8	6.2	5.4	4.6	4	3.6	3	2.8	2.4
⅜ B	*27*	*24*	*21*	*18*	*16.5*	*14*	*13*	*11*	*10*	*8.5*	*8*	*7*
½ A	20	18	16	13.5	12	10.8	9	8	7	6	5.4	4.7
½ B	*33*	*30*	*25*	*21*	*19*	*17*	*15*	*13*	*12*	*10*	*9*	*8*
⅝ A	36	30	26	22.1	20	17.8	15	13.1	11.8	9.9	9	7.9
⅝ B	*37*	*34*	*30*	*24*	*21*	*19*	*17*	*15*	*13*	*11*	*10*	*9*
¾ A	51	46	40	34	31	28	23.2	20.5	18.1	15.3	13.9	12.1
¾ B	*42*	*38*	*33*	*27*	*24*	*21*	*19*	*17*	*14*	*12*	*11.5*	*10*
1 A	104	94	82	70	63	56	47	42	37	32	28	25
1 B	*48*	*45*	*39*	*34*	*30*	*25*	*22*	*19*	*17*	*14.5*	*13*	*12*
1¼ A	185	169	149	125	112	100	84	75	66	56	50	44
1¼ B	*55*	*51*	*45*	*39*	*35*	*30*	*25*	*22*	*19*	*17*	*15*	*13*
1½ A	300	270	235	200	180	160	134	120	105	90	81	71
1½ B	*62*	*57*	*51*	*43*	*39*	*35*	*30*	*25*	*22*	*19*	*17*	*15*
2 A	625	560	495	420	375	335	280	250	200	188	170	150
2 B	*76*	*68*	*59*	*51*	*47*	*42*	*36*	*32*	*27*	*22*	*20*	*18*
2½ A	1130	1010	890	750	680	600	500	450	395	335	305	270
2½ B	*90*	*80*	*69*	*58*	*49*	*47*	*42*	*37*	*33*	*26*	*23*	*21*
3 A	1840	1650	1450	1210	1100	980	820	740	650	550	490	420
3 B	*98*	*90*	*80*	*66*	*59*	*52*	*47*	*42*	*36*	*30*	*27*	*23*
3½ A	2750	2480	2170	1840	1650	1450	1210	1100	980	820	740	650
3½ B	*110*	*100*	*89*	*75*	*66*	*57*	*51*	*45*	*40*	*35*	*30*	*26*
4 A	3900	3505	3100	2600	2350	2090	1760	1580	1390	1180	1080	950
4 B	*120*	*108*	*96*	*83*	*73*	*63*	*55*	*49*	*44*	*37*	*34*	*29*

[a]For other temperature drops the pipe capacities may be changed correspondingly. For example, with temperature drop of 30 F, the capacities shown in this table are to be multiplied by 1.5.

size and flow. Assume 20 F drop for this system. One gallon of water per minute with a density of 7.99 at 215 F will deliver approximately 9600 Btu per hour with a 20 F drop.

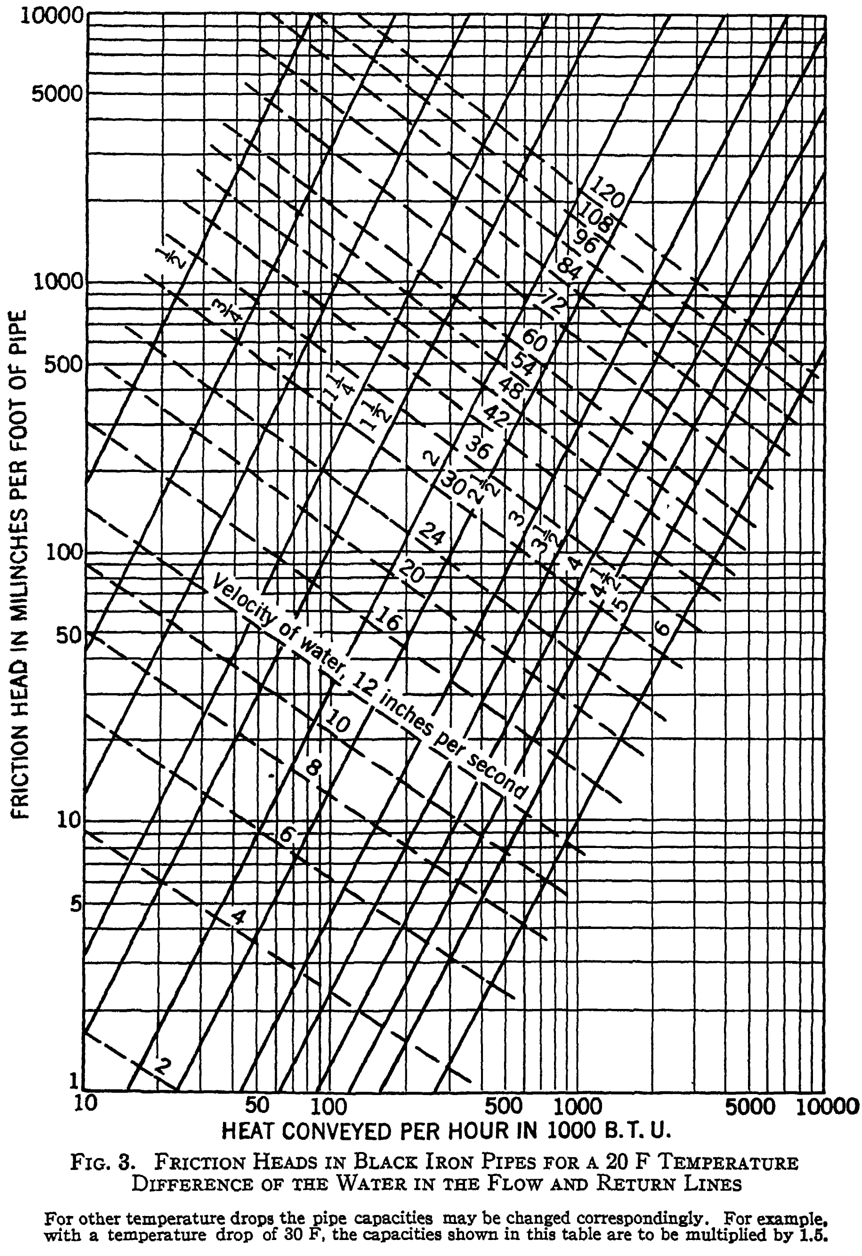


FIG. 3. FRICTION HEADS IN BLACK IRON PIPES FOR A 20 F TEMPERATURE DIFFERENCE OF THE WATER IN THE FLOW AND RETURN LINES

For other temperature drops the pipe capacities may be changed correspondingly. For example, with a temperature drop of 30 F, the capacities shown in this table are to be multiplied by 1.5.

The total radiation load is 98 Mbh, therefore the pump must deliver 10.2 gpm or 4900 lb of water per hour.

Knowing that the rate of flow is 10.2 gpm, the next step is to determine from the characteristics of available pumps, which one will produce a satisfactory velocity in the system. Assume that 4 pumps are available for this load which will produce 10.2 gpm at pressure heads of 2, 5, 10 and 18 ft. At these heads the pumps would produce a velocity high enough to make available a friction head per foot of pipe of 96, 240, 480 and 860 milinches per foot respectively. If 95 milinches per foot were used, the gravity head at 215 F average temperature in the mains would be 26 per cent of the total head and should be considered in sizing the system. At 240 milinches per foot the gravity effect is 10 per cent and as this is lower than the delivery variation from the pipe used, it can be neglected. At 480 and 860 milinches the gravity effect is still a smaller percentage of

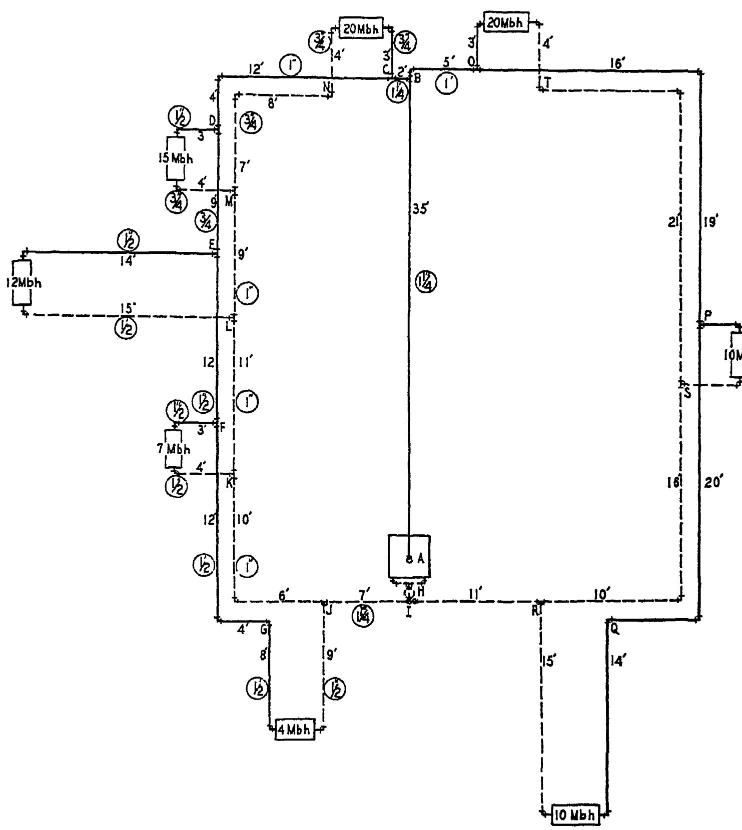


FIG. 4. A FORCED CIRCULATION REVERSED RETURN SYSTEM[a]

[a]Note that the numbers on the radiators indicate thousands of Btu per hour (Mbh) and not square feet.

the total, but at these losses in the average system the cost of pumping will more than offset the advantage gained in pipe sizes. Therefore, pipe size this system at 240 milinches per foot which is equivalent to a total loss of 60,000 milinches for the 250 ft equivalent length of pipe.

The pipe sizes may be selected from Fig. 3 or from Table 1 which has been derived from Fig. 3.

Size the supply main of the longest circuit first. Section AB carries 98 Mbh. From Fig. 3 it will be noted that at 240 milinches per foot, a 1¼-in. pipe carries 98 Mbh. Therefore, use 1¼-in. pipe in Section AB. Section BO carries 40 Mbh. A 1-in. pipe carries 48 Mbh at 240 milinches per foot. Use a 1-in. pipe. Section OP carries 20 Mbh

TABLE 3. IRON AND COPPER ELBOW EQUIVALENTS

Fitting	Iron Pipe	Copper Tubing
Elbow, 90-deg	1.0	1.0
Elbow, 45-deg	0.7	0.7
Elbow, 90-deg long turn	0.5	0.5
Open return band	1.0	1.0
Open gate valve	0.5	0.7
Open globe valve	12.0	17.0
Angle radiator valve	2.0	3.0
Radiator	3.0	4.0
Boiler or heater	3.0	4.0
Tee, per cent flowing through branch:		
100	1.8	1.2
50	4.0	4.0
25	16.0	20.0

and this will require ¾-in. pipe. Section PQ carries 10 Mbh and requires ½ in. pipe. To size the return start from the boiler and proceed backwards. Section IR carries 40 Mbh and from Fig. 3 a 1-in. pipe is required. Section RS carries 30 Mbh which is only slightly over the capacity of a ¾-in. pipe, so use ¾ in. Section ST carries 20 Mbh and requires a ¾-in. pipe. The radiator branches are determined in the same manner. It is evident from the chart that it is impossible to maintain a constant friction loss per foot and therefore as the delivery varies there will be a change in the desired friction loss per foot of pipe.

TABLE 4. PIPING CHECK CHART

Load, *Mbh*			Pipe Length Ft	Elbows	Pipe Size In.	Unit Head Milinches per Ft	Friction Milinches	Total Loss Milinches
Supply Main								
AB	98		37	1	1¼	240	9600	9,600
BC	58		2	4	1¼	90	1080	10,680
CD	38		16	1	1	155	2790	13,470
DE	23		9	0	¾	220	1980	15,450
EF	11		12	0	½	240	2880	18,330
FG	4		16	1	½	50	850	19,180
Return Main								
HI	98		5	5	1¼	240	4320	4,320
IJ	58		11	1	1¼	90	1260	5,580
JK	54		16	1	1	300	5400	10,880
KL	47		11	0	1	230	2530	13,410
LM	35		9	0	1	140	1260	14,670
MN	20		15	1	¾	170	2890	17,560
Radiator Circuits								
CN	20	*Supply*	3	13	¾	170	3910	
		Return	4	2	¾	170	1190	5,100
DM	15	*Supply*	3	19	½	420	9250	
		Return	4	17	¾	96	2880	12,130
EL	12	*Supply*	14	20	½	270	9180	
		Return	15	20	½	270	9450	18,630
FK	7	*Supply*	3	19	½	100	2200	
		Return	4	17	½	100	2100	4,300
GJ	4	*Supply*	8	5	½	50	650	
		Return	9	17	½	50	1300	1,950

It is desirable to check the various circuits so that if the variation from the calculated resistance is too great, it may be compensated by adding additional resistance at the proper point. This may be accomplished by sizing the short circuits by the procedure previously outlined. Prepare a chart such as Table 4 to be used in calculating the resistance of each circuit.

Section AB carries 98 Mbh with a unit head of 240 milinches per foot. In section AB there are 37 ft of pipe and 1¼ in. elbow. At 240 milinches per foot this is equivalent to 9600 milinches total loss in this section. Section BC carries 58 Mbh with a length of 2 ft and 4 elbows. The unit loss in this section is 90 milinches per foot. Loss in this section is then 1080 milinches. Section CD carries 38 Mbh and has 16 ft of pipe and 1 elbow. The unit loss in 1-in. pipe is 155 milinches. The loss in this section is 2790 milinches. The balance of the supply main and the return main are handled in a similar manner.

The radiator circuits are then checked. The 20 Mbh radiator on this circuit has 3 ft of supply pipe and 13 elbow equivalents while the return is composed of 4 ft and 2 elbows. The unit loss in ¾ in. pipe at this delivery is 170 milinches per foot. The total loss in the supply is 3910 milinches. The loss in the return is 1190. Total loss in the radiator circuit is 5100 milinches. Check each radiator circuit in a similar manner.

The total calculated loss for the longest circuit was determined as 60,000 milinches. The maximum loss in the short circuit is 18,630 plus 13,410 plus 15,450 or a total of

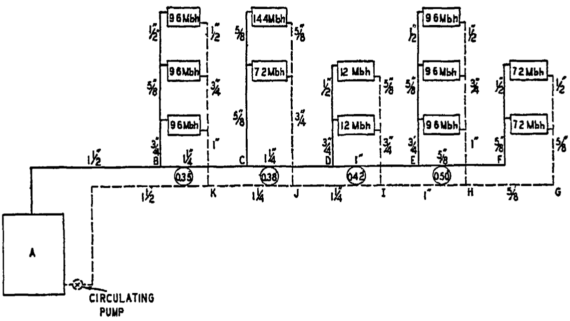


FIG. 5. A FORCED CIRCULATION DIRECT RETURN SYSTEM

47,490 milinches. This difference is caused by the variation in length of the two circuits and may be corrected by using a flow control in the return main to supply the additional resistance or by introducing resistance into each separate circuit to compensate for the difference. A 10 per cent variation will cause no complication as the flow from the various pipes will not exactly follow the curves of Fig. 3 any closer than this value.

Example 2. Design a two-pipe direct return forced circulation system with copper tubing and fittings for the piping layout as detailed in Fig. 5, based on a 20 F temperature drop through the radiation.

The piping circuit from the boiler to the highest radiator on the farthest riser and back to the boiler is 250 ft of pipe. There are about 16 elbow equivalents having an equivalent pipe length of about 50 ft, so that the total equivalent pipe length is 300 ft.

Assume that a circulator is available which will provide a pressure head of 6 ft

Solution. Refer to Table 2, which indicates the total equivalent lengths for pressure heads from 2 to 12 ft. With a circulator having a 6 ft pressure head and a system with a total equivalent length of 300 ft, the piping system will be designed on a basis of 240 milinch.

Checking the piping diagram it will be noted that sections AB and KA, both supply 117.6 Mbh. Referring to the 240 milinch column of Table 2, 1½ in. is shown to be the necessary pipe size. Sections BC and JK carry 88.8 Mbh and require 1¼ in. tubing. Sections CD and IJ supply 67.2 Mbh and require 1¼ in. tubing. Sections DE and HI supply 43.2 Mbh, which requires 1 in. tubing. Sections EF and GH with a load of 14.4 Mbh require ⅝ in. tubing.

The risers are pipe sized in a similar manner. To secure proper distribution of hot

water in the direct return system among the several risers, it is necessary to introduce resistances to balance the circuit.

The first riser is 80 ft nearer the boiler than the fifth riser. In order that the two may be balanced, that is, operated under equal pressure heads, resistance must be added to

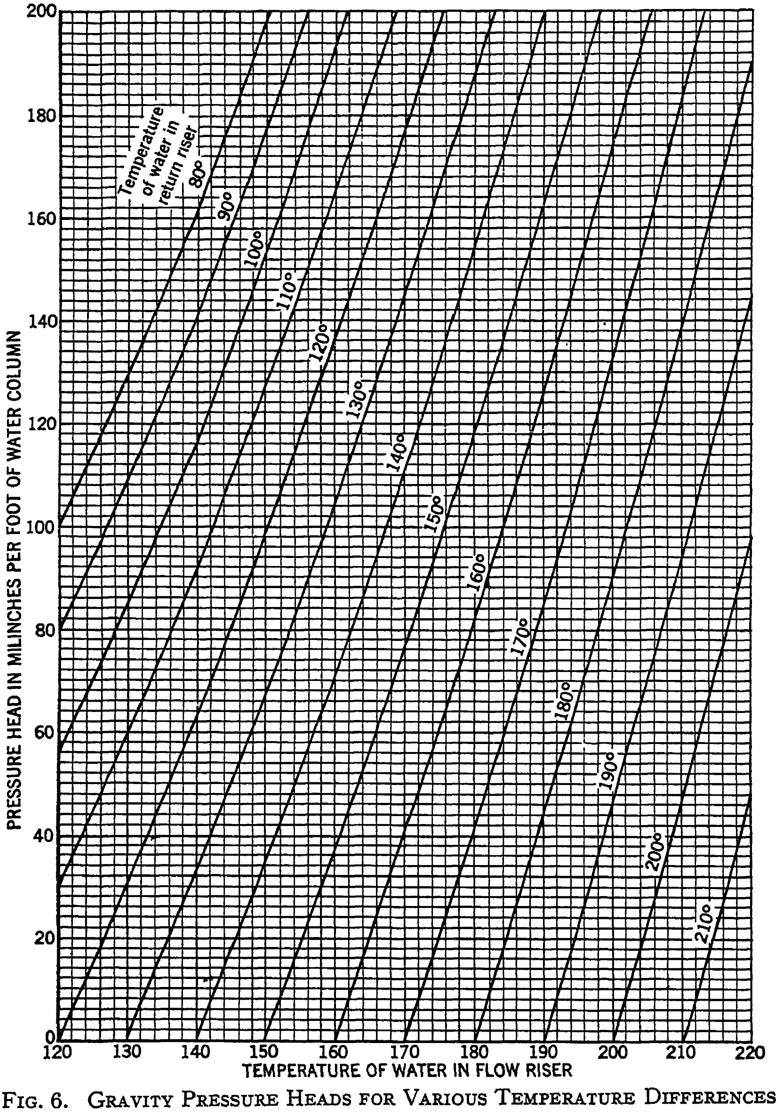


FIG. 6. GRAVITY PRESSURE HEADS FOR VARIOUS TEMPERATURE DIFFERENCES

the first riser equal to the friction head in the 80 ft of supply main B to F plus the 80 ft of return main G to K for a total of 160 ft of pipe.

Having designed the piping system on a 240 milinch basis, the total friction head in the supply and return mains between the first and fifth risers is therefore 160 $\times$ 240 = 38,400 milinches, or 3.2 ft which must be supplied by additional resistance in the first riser. This resistance can be supplied by an adjusting valve or by an orifice of size selected from Table 5.

GRAVITY CIRCULATION PIPE SIZES

In gravity hot water heating systems the difference in temperature (density) between the flow and return produces the circulation of the water. The temperature difference is usually made from 25 to 35 F. Having determined the temperature difference and the temperature of flow, Fig. 6 can be used to obtain the pressure head, and from this point the calculations are the same as for forced hot water. Heat emission rates from 150 to 170 Btu per square foot are commonly used so that flow temperatures range from 180 to 200 F or higher. Assuming a flow temperature of 200 F and 35 F drop and the mains 4 ft above the boiler, a circulating pressure head of 600 milinches results. Assuming first floor radiators 3 ft above the mains and second floor radiators 12 ft above mains, third floor 21 ft and fourth floor 30 ft, the circulating pressure heads are 450, 1800, 3150 and 4500 milinches respectively.

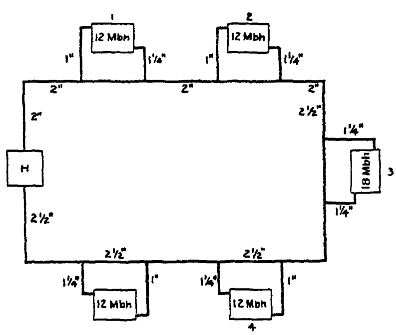


FIG. 7. A ONE-PIPE GRAVITY CIRCULATION SYSTEM

The following examples will illustrate the method to be followed in designing a gravity hot water system:

Example 3. Design a one-pipe gravity circulation system for the layout shown in Fig. 7. Assume that the main circuit consists of 150 ft of pipe, 7 elbows, and one boiler.

Solution. Replace the boiler by 3 elbow equivalents and assume that the size of the main will be about 2 in. According to Table 6, Column 2, a 2-in. elbow is equivalent to 4 ft of pipe, and the total equivalent length of the main will be about 150 plus 40, or 190 ft. Assuming that the center of the boiler will be about 4 ft lower than the horizontal portion of the main and that the temperature drop in the system is to be 35 F, Table 6 may be used to determine the size of the mains. Note from Column 8, for a 200 ft length, that a 2-in. main will supply 48 Mbh and a 2½-in. main, 75.4 Mbh. Since the system to be designed is to supply 66 Mbh, a 2-in. pipe is too small and a 2½-in. pipe too large. The solution is to use some 2 in. and some 2½ in. pipe. Since the 2½ in. is nearer the correct size than the 2 in., select 2-in. pipe for the first 50 or 60 ft from the boiler and 2½ in. for the remaining pipe back to the boiler.

Tables 7 and 8 may be used to design the radiator risers and connections. According to Table 7, for 12 Mbh the flow riser should be ¾ in. and the return riser 1 in., and the riser branches should be 1 in. and 1¼ in., respectively. Note that according to Table 8, both radiator tappings should be 1 in. To simplify the construction, select 1-in. flow risers with 1-in. riser branches and 1-in. radiator tappings. Also select 1¼-in. return risers with 1¼-in. riser branches, and 1¼-in. radiator tappings. Similarly, for 18 Mbh, select 1¼-in. flow and return risers and riser branches, and 1¼-in. radiator tappings.

To develop a rule for determining radiator sizes, assume a system similar to that of Fig. 7, in which the total temperature drop is to be 35 F

Table 5. Friction Heads (in Milinches) of Central Circular Diaphragm Orifices in Unions

Diameter of Orifices (Inches)	Velocity of Water in Pipe in Feet per Minute									
	10	15	20	30	40	50	60	90	120	180
¾-in. Pipe										
0.25	1300	2900	5000	11,300	20,800	32,000	45,000			
0.30	650	1450	2500	5700	10,400	16,000	23,000	57,000		
0.35	330	740	1300	2900	5200	8000	12,000	26,000	47,000	
0.40	170	380	660	1500	2600	4000	6800	13,000	24,000	53,000
0.45		185	330	740	1300	2000	2900	6500	12,000	27,000
0.50			155	350	620	970	1400	3200	5700	13,000
0.55			75	170	300	480	700	1600	2800	6400
1-in Pipe										
0.35	900	2000	3500	7800	14,000	22,000	32,000			
0.40	460	1000	1800	4000	7200	12,000	17,000	37,000	65,000	
0.45	270	570	1000	2300	4100	6400	9300	21,000	37,000	
0.50	160	330	580	1400	2300	3700	5400	12,000	22,000	50,000
0.55		190	330	750	1300	2200	3000	7000	13,000	28,000
0.60			200	440	800	1300	1800	4200	7400	17,000
0.65			120	260	460	720	1100	2400	4300	10,000
1¼-in. Pipe										
0.45	1000	2250	4000	8900	16,000	25,000	36,000			
0.50	660	1450	2600	5800	10,400	16,400	23,000	53,000		
0.55	430	950	1700	3800	6800	10,500	15,000	34,000	60,000	
0.60	280	630	1100	2500	4400	6900	10,000	22,000	40,000	
0.65	190	420	750	1700	3000	4700	6700	15,000	27,000	60,000
0.70		285	510	1150	2000	3100	4500	10,000	18,000	40,000
0.75		190	330	750	1300	2100	3000	6700	12,000	26,000
1½-in. Pipe										
0.55	850	1900	3300	7400	13,000	21,000	30,000			
0.60	600	1300	2300	5400	8600	16,800	21,000	50,000		
0.65	400	850	1500	3600	7200	10,400	14,000	30,000	53,000	
0.70	260	600	1100	2600	4400	7000	10,000	21,000	39,000	
0.75	180	400	760	1800	3000	5000	7000	14,000	28,000	
0.80		300	540	1200	2200	3200	5000	10,200	19,000	45,000
0.85		200	380	860	1600	2300	3000	7800	13,000	30,000
2-in. Pipe										
0.70	890	1850	3500	7400	14,000	22,300	33,000			
0.80	470	975	1800	3900	7400	11,700	17,000	37,000		
0.90	255	560	1000	2200	4200	6500	9500	20,500	38,000	
1.00	160	340	610	1320	2520	4000	5800	12,500	23,000	49,000
1.10		214	375	850	1600	2500	3700	7900	14,000	30.000
1.20			195	460	950	1360	1910	4200	8100	16,800
1.30				275	525	980	1375	3100	4400	8850

Note.—The losses of head for the orifices in the 1½-in. and 2-in. pipe were calculated from those in the smaller pipes, the calculations being based on the assumption that, for any given velocity, the loss of head is a function of the ratio of the diameter of the pipe to that of the orifice. This had been found to be practically true in the tests to determine the losses of head in orifices in ¾-in., 1-in., and 1¼-in. pipe, conducted by the Texas Engineering Experiment Station, and also in the tests to determine the losses of head in orifices in 4-in., 6-in., and 12-in. pipe, conducted by the Engineering Experiment Station of the University of Illinois, (*Bulletin* 109, Table 6, p. 38, Davis and Jordan).

and which is equipped with 7 radiators, all radiators dissipating equal quantities of heat. The mean temperature of the water in the radiators will be reduced 5 F for each successive radiator. If the mean temperature of the water in the first radiator is 200 F, the mean temperature of the water in the seventh radiator will be 170 F, and, according to Table 5, Chapter 13, the heat dissipation of these two radiators will be to each other as 1.62 is to 1.15, or as 140 is to 100, and therefore if the last radiator is to dissipate as much heat as the first, its size must be 40 per cent larger.

Example 4. Design a two-pipe, direct return, gravity circulation system for the layout shown in Fig. 8. Assume that the main circuit from the boiler to the farthest flow riser and from the farthest return riser back to the boiler consists of 160 ft of pipe, 6 elbows, and 1 boiler.

Solution. Replacing the boiler by 3 elbow equivalents and assuming that the largest size of the main will be about 3 in., the total equivalent length of the main will be 160 plus 45, or 205 ft. Assuming that the center of the boiler will be about 4 ft lower than the horizontal portion of the main, and that the temperature drop will be 35 F for the

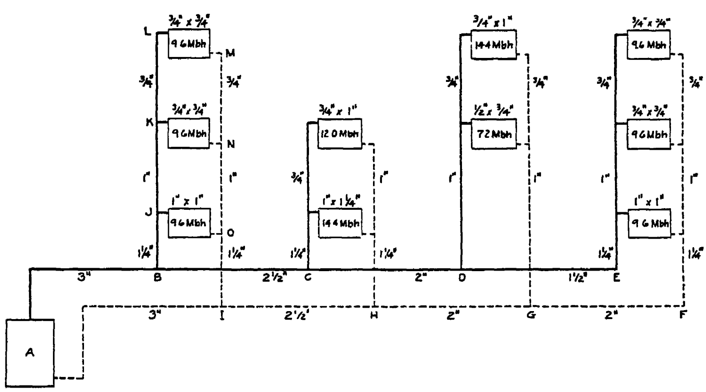


FIG. 8. A TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEM

system, the pressure head caused by the difference in weight between the water in the flow and return risers joining the mains to the boiler will be about 0.6 in. of water.

Table 6 may be used to determine the size of the main as follows: Refer to Column 8 and note that for Sections *AB* and *IA*, which supply 105.6 Mbh, a 3-in. pipe is too large and a 2½-in. pipe is too small; hence, select 2½ in. rather than 3 in. as noted in Fig. 8 for Section *AB* and 3 in. for Section *IA*. For Sections *BC* and *HI*, which supply 76.8 Mbh, a 2½-in. pipe is almost exactly the correct size and is selected for both sections.

Tables 7 and 8 are based on the assumption that the boiler pressure head must be equal to the friction head in the mains, and that the several radiator pressure heads must be equal to the respective radiator and riser friction heads.

To design the radiator risers, use Table 7 and begin with the set nearest the boiler. The first floor risers must supply 28.8 Mbh. According to the table, 1¼-in. flow and return risers will supply 26.0 Mbh; if the return riser is increased to 1½ in., the capacity will be increased to 34.0 Mbh. This is considerably larger than necessary, and 1¼-in. flow and return risers are selected. However, it must be remembered that the riser branches, which are the connections from the flow and return mains to the flow and return risers, are to be one size larger than the risers.

The second floor risers must supply 19.2 Mbh. According to the table, the capacity of 1 in. flow and return risers is 20.0 Mbh, and that size is selected.

The third floor risers must supply 9.6 Mbh. If a ½-in. flow and a ¾-in. return riser

TABLE 6 CAPACITIES OF MAINS IN *Mbh*, FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A TOTAL FRICTION HEAD OF 0.6 IN., A TEMPERATURE DROP OF 35 F. WHEN THE MAINS ARE 4 FT ABOVE THE CENTER OF THE BOILER

1	2	3	4	5	6	7	8	9	10	11
Pipe Size (Inches)	Equivalent Length of Pipe (Feet[a])	Equivalent Total Length of Pipe in Feet in Longest Circuit								
		75	100	125	150	175	200	250	300	350
		Unit Friction Head, in Milinches								
		8.0	6.0	4.8	4.0	3.4	3.0	2.4	2.0	1.7
1½	3.0	*43.0*	*37.5*	*33.0*	*30.0*	*27.0*	*25.0*	*22.2*	*20.2*	*18.7*
2	4.0	*83.0*	*72.0*	*63.0*	*57.0*	*51.0*	*48.0*	*42.0*	*38.0*	*35.0*
2½	4.5	*140.0*	*115.0*	*100.0*	*90.0*	*81.5*	*75.4*	*67.2*	*61.0*	*56.0*
3	5.0	*234.0*	*204.0*	*175.5*	*160.0*	*143.0*	*133.0*	*110.0*	*107.5*	*100.0*
3½	5.5	*347.0*	*300.0*	*260.0*	*236.0*	*214.0*	*200.0*	*177.0*	*160.0*	*146.0*
4	6.0	*490.0*	*422 0*	*370.0*	*334.0*	*297.0*	*278.0*	*248.0*	*223.0*	*205.0*

[a]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

are used, the capacity will be 8.0 Mbh; if both risers are ¾ in., the capacity will be 14.0 Mbh. The ¾-in. pipe is selected for both risers.

To design the radiator connections, use Table 8 and note that for the first floor radiator connections the capacity of a ¾-in. flow and 1-in. return is 9.1 Mbh, and that of a 1-in. flow and a 1-in. return is 12.5 Mbh. The former is more nearly the correct size, but since it is difficult to secure a good flow through first floor radiators, the 1-in. flow and return connection is selected. For the two upper floors, the capacity of a ¾-in. flow and return connection is 10.5 Mbh, and that size is used.

TABLE 7. MAXIMUM CAPACITIES OF RISERS[a] IN *Mbh*, AND **Velocities of Water in Pipes in Inches Per Second** FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A DROP OF 35 F THROUGH EACH RADIATOR

Pipe Size (Inches)		Equivalent Length of Pipe (Feet[c])	1st Floor[b]			2nd Floor	3rd and 4th Floors
Flow	Return		*Mbh*	Vel (In. per Sec[d])		*Mbh*	*Mbh*
				Flow	Return		
½	½	1.0				*5.0*	*6.2*
½	¾					*6.4*	*8.0*
¾	¾	1.5	*9*	2.3	2.3	*10.1*	*14.0*
¾	1		*12*	3.2	2.0	*12.8*	*17.1*
1	1	2.0	*18*	2.5	2.5	*20.0*	*26.0*
1	1¼		*21*	3.0	2.0	*25.2*	*34.0*
1¼	1¼	3.0	*26*	3.0	3.0	*43.0*	*55.0*
1¼	1½		*34*	4.0	2.5		
1½	1½	3.5	*48*	3.0	3.0		

[a]This table is based on pressure heads of 450, 1800, 3150, and 4500, respectively, for the first, second, third, and fourth floor radiators, and on friction heads of 200 milinches for the first floor radiators and connections, and 700 milinches for all other radiators and their connections.

[b]The riser branches, the piping which connects the risers to the mains, are to be one size larger than the risers.

[c]Approximate length of pipes in feet equivalent to one elbow in friction head. This value varies with the velocity.

[d]Velocities apply to the riser branches.

TABLE 8. MAXIMUM CAPACITIES OF RADIATOR CONNECTIONS IN *Mbh*, FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A TEMPERATURE DROP OF 35 F THROUGH EACH RADIATOR

Pipe Size		Equivalent Length of Pipe (Feet[a])	1st Floor	2nd, 3rd, and 4th Floors
Flow	Return		*Mbh*	*Mbh*
½	½	1.0	4.1	5.9
½	¾		5.2	7.5
¾	¾	1.5	7.0	10.5
¾	1		9.1	13.0
1	1	2.0	12.5	17.8
1	1¼		17.5	23.2
1¼	1¼	3.0	23.3	33.2

[a]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

As explained in the design of the forced circulation system of Fig. 5, the two-pipe direct return system of Fig. 8 will not function correctly unless its four sets of risers are balanced among themselves. This necessary balancing is accomplished by adding resistances to all risers, except the one farthest from the boiler, equal to the excess boiler pressure heads available for those risers above the boiler pressure head available for the farthest riser. For example, the first set of risers is 60 ft nearer the boiler than the last set. Since the flow and return mains are designed for a friction head of 3 milinches per foot (see Table 6, Column 8), the boiler pressure head available for the first set of risers is 360 milinches in excess of that available for the fourth set. The velocity in the riser branch is 3 in. per second (see Table 7) and, therefore, according to Table 5, an 0.65-in. orifice in a 1¼-in. union should be used. This will provide a resistance of about 420 milinches. In the same manner it is found that for the second set of risers a resistance of 240 milinches is required and that an 0.70-in. orifice in a 1¼-in. union will provide a resistance of 285

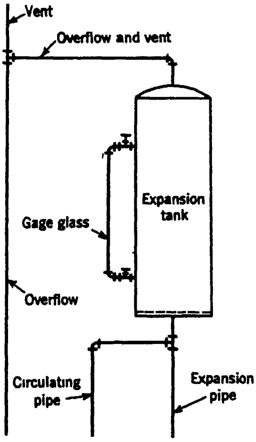


FIG. 9. AN OPEN EXPANSION TANK

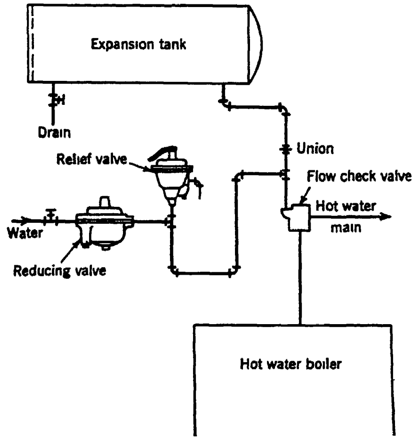


FIG. 10. A CLOSED EXPANSION TANK

milinches. For the third set of risers, a resistance of 120 milinches is required and an 0.60-in. orifice in a 1-in. union will provide sufficient resistance.

EXPANSION TANKS

Expansion tanks may be either of the open or of the closed type. In the open type, (see Fig. 9) the water is subject to atmospheric pressure only, but in the closed tank (see Fig. 10) the system is under pressure and, therefore, a relief valve should be placed on the tank. Water expands about 4 per cent when being heated from 40 F to 200 F, and the expansion tank should have a volume about twice the actual expansion or about 8 per cent of the total volume of water in the entire system including boiler, radiators, pipes, etc. Open expansion tanks should be at least 3 ft above the highest radiator and be protected against freezing. Closed tanks are generally placed in the cellar over the boiler.

A relief valve installed on a closed tank will not operate often provided the tank is of adequate size. It is essential that the relief valve be kept in good condition to eliminate any possible failure when operation is necessary.

INSTALLATION DETAILS

Attention should be paid to the following:

All piping must be so pitched that all air in the system can be vented either through an open expansion tank, radiators or automatic relief valves.

All piping must be arranged so that the entire system can be drained. Sections of piping individually valved shall have corresponding drain valves.

In large buildings, the piping may be zoned according to exposure of building, usage of building, or method of control.

All piping must be installed so that it is free to expand and contract with changes of temperature without producing undue stresses in the pipes or connections. For this purpose it is generally sufficient to allow for a variation in length of 1 in. for 100 ft of pipe.

The pipe system should be designed so that each circuit has its correct friction head for balanced water distribution. This may be done by change of pipe size or change in piping detail.

The connections from the boiler to the mains should be short and direct, to reduce the friction head and allow for expansion. It is frequently possible to avoid an elbow and to reduce the length of the pipe by running the pipe in a diagonal direction, either in a horizontal or in a vertical plane.

The mains and branches should pitch up and away from the heater, generally not less than 1 in. in 10 ft.

The connections from mains to branches and to risers should be such that circulation through the risers will start in the right direction. Hence, in a one-pipe system the flow connection must be nearer the heater than the return connection. In a correctly-designed two-pipe system, the pressure in the flow main is higher than that in the return main, and a slight variation in the distances of the flow and return connections from the heater is not material; but it is generally best to have the two connections about equally distant from the heater.

Generally connections to risers or radiators are taken out of the top of mains either 45 or 90 deg. In some cases it may be advisable to take the flow connection off the top of the main and the return connection from the side.

With forced circulation and high velocities, it is advisable to let the water enter at the top of the radiator and leave at the bottom of the opposite end. With gravity circulation the flow connection may be either at the top or at the bottom of the radiator. With short radiators both flow and return may be at same end, but top and bottom.

Unless used as heating surface, all piping, both flow and return, should be insulated.

Chapter 17

DISTRICT HEATING

Steam Distribution Piping, Selection of Pipe Sizes, Provision for Expansion, Capacity of Returns with Various Grades, Conduits for Piping, Pipe Tunnels, Inside Piping, Steam Requirements, Fluid Meters and Metering, Rates, Utilization, Automatic Temperature Control

THOSE phases of district heating which frequently fall within the province of the heating engineer are outlined here with data and information for solving incidental problems in connection with institutions and factories. Some data are included to cover the piping peculiar to heating systems which are to be supplied with purchased steam. A complete district heating installation should not be attempted without a thorough study of the entire problem by men competent and experienced in that industry.

STEAM DISTRIBUTION PIPING

The methods used in district heating work for the distribution of steam are applicable to any problem involving the supply of steam to a group of buildings. The first step is to establish the route of the pipes, and in this matter the local conditions so fully control the layout that little can be said regarding it.

Having established the route of the pipes, the next step is to calculate the pipe sizes. In district heating work it is common practice to design the piping system on the basis of pressure drop. The initial pressure and the minimum permissible terminal pressure are specified and the pipe sizes are so chosen that the required amount of steam, with suitable allowances for future increases, will be transmitted without exceeding this pressure drop. The steam velocity is therefore almost disregarded and may reach a very high figure. Velocities of 35,000 fpm are not considered high. By the use of this method the pipe sizes are kept to a minimum with consequent savings in investment.

The steam flowing through any section of the piping can be computed from a study of the requirements of the several buildings served. In general a condensation rate of 0.25 lb per hour per square foot of equivalent heating surface is a safe figure. This allows for line condensation which, however, is a small part of the total at times of maximum load. Miscellaneous steam requirements such as laundry, cooking, or process should be individually calculated.

The steam requirements for water heating should be taken into account,

but in most types of buildings this load will be relatively small compared with the heating load and will seldom occur at the time of the heating peak. Unusual features such as large heaters for swimming pools should not be overlooked.

The pressure at which the steam is to be distributed will depend upon (1) boiler pressure, (2) whether exhaust or live steam, (3) pressure requirements of apparatus to be served. If steam has been passed through electrical generating units, the pressure will be considerably lower than if live steam, direct from the boilers, is used.

The advantages of low pressure distribution (2 to 30 lb per square inch) are (1) smaller heat loss from the pipes, (2) less trouble with traps and valves, (3) simpler problems in pressure reduction at the buildings, and (4) general reduction in maintenance costs. With distribution pressures not exceeding 40 lb per square inch there is little danger even if the full distribution pressure should build up in the radiators through the faulty operation of a reducing valve; but with pressures higher than this a second reducing valve or some form of emergency relief is usually desirable to prevent excessive pressures in the radiators.

The advantages of high pressure distribution are (1) smaller pipe sizes and (2) greater adaptability of the steam to various operations other than building heating.

The different kinds of apparatus which frequently must be served require various minimum pressures. Kitchen equipment requires from 5 to 15 lb per square inch, the higher pressures being necessary for apparatus in which water is boiled, such as stock kettles and coffee urns. An increased amount of heating surface, which is easily obtained in some kinds of apparatus, results in quicker and more satisfactory operation at low pressures. For laundry equipment, particularly the mangle, a pressure of 75 lb per square inch is usually demanded although 30 lb per square inch is sufficient if the mangle is equipped with a large number of rolls and if a slow rate of operation is permissible. Pressing machines and hospital sterilizers require about 50 lb per square inch.

PIPE SIZES

The lengths of pipe, steam quantities, and initial and terminal pressures having been chosen, the pipe sizes can readily be calculated by means of the Unwin pressure drop formula. This is one of several formulae which may be used. Unwin's formula, which gives pressure drops slightly larger than actual test results, is as follows:

$$P = \frac{0.0001306\ W^2L\left(1 + \frac{3.6}{D}\right)}{dD^5} \tag{1}$$

where

P = pressure drop, pounds per square inch.
W = weight of steam flowing, pounds per minute.
L = length of pipe, feet.
D = inside diameter of pipe, inches.
d = average density of steam, pounds per cubic foot.

This formula is similar to the Babcock formula given in Chapter 15.

Information on provision for expansion will be found in Chapter 18. Where steam and return piping are installed in the same conduit, the return piping usually follows the same grade as the steam piping. In general, the condensation is pumped back under pressure. Where the condensation returns by gravity, Table 1 gives the sizes of the return piping. It is evident that at points where the grade is great, smaller pipes can be installed.

CONDUITS FOR PIPING

Conduits for steam pipes buried underground should be reasonably waterproof, able to withstand earth loads and to take care of the expansion and contraction of the piping without strain or stress on the couplings, or without affecting the insulation or conduit. Expansion of the piping must be carefully controlled by means of anchors and expansion joints or bends so that the pipes can never come in contact with the conduit. Anchors can be anchor fittings or U-shaped steel straps which partially encircle the pipes and are firmly bolted to a short length of structural or cast steel set in concrete. In general, cast steel is preferable to structural steel.

TABLE 1. CAPACITY OF RETURNS FOR UNDERGROUND DISTRIBUTION SYSTEMS IN POUNDS OF CONDENSATE PER HOUR

Size[a] of Pipe In.	Pitch of Pipe per 100 Ft						
	6"	1'	2'	3'	5'	10'	20'
1	448	998	1890	2240	3490	5490	7490
1¼	1740	2490	3990	4880	6480	9480	13500
1½	2700	4190	5740	7480	9480	14500	20900
2	4980	7380	10700	13900	16900	24900	36900
3	13900	22500	30900	37400	50400	74800	105000
4	30900	44800	64800	79700	105000	154000	229000
5	54800	79800	120000	144800	195000	294000	418000
6	90000	138000	187000	237000	312000	449000	--------
8	190000	277000	404000	508000	660000	938000	--------
10	344000	498000	724000	900000	1190000	--------	--------
12	555000	798000	1148000	1499000	1990000	--------	--------

[a]Size of pipe should be increased if it carries any steam.

In laying out underground conduits the following points should be borne in mind:

1. The depth of the buried conduit should be kept at a minimum. Excavation costs are a large factor in the total cost.

2. An expansion joint, offset, or bend should be placed between each two anchors.

3. If the distance between buildings is 150 ft or less and the steam line contains high-pressure steam, the line may be anchored in the basement of one building and allowed to expand into the basement of the second building. If the steam line contains low-pressure steam (up to 4-lb pressure), this method may be used if buildings are 250 ft or less apart.

4. If the distance between buildings is between 150 ft and 300 ft and the steam line contains high-pressure steam, the lines should be anchored midway between the buildings and allowed to expand into the basements of both buildings. If the steam line contains low-pressure steam this method may be used if buildings are between 250 ft and 600 ft apart. No manhole is required at the anchor, and a blind pit is all that is necessary.

5. For longer lines, manholes must be located according to experience, physical conditions and the expansion value of the type of expansion joint or bend that is used. The

minimum number of manholes will be required when an expansion bend, or an anchor with double expansion joint, is placed in each manhole and the pipes are anchored midway between manholes.

6. A proper hydrostatic test should be made on the assembled line before the insulation and the top of the conduit are applied. The hydrostatic test pressure should be one and one-half times the maximum allowable pressure and it should be held for a period of at least two hours without evidence of leakage. In any case the pressure should be no less than 100 lb per square inch.

There are many types of conduits, some of which are manufactured products and some of which are built in the field. The styles and construction of conduits commonly used may be classified as follows. Some of the more common forms are illustrated in Fig. 1.

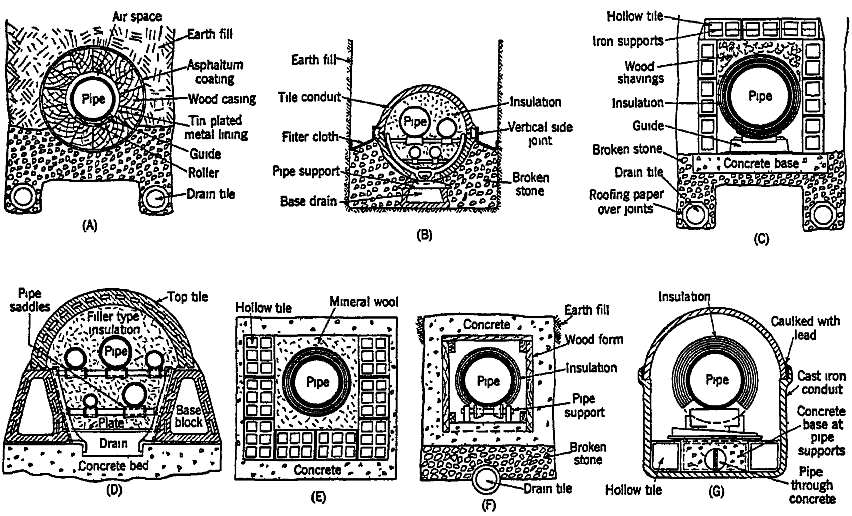


FIG. 1. CONSTRUCTION DETAILS OF CONDUITS COMMONLY USED

Wood Casing: The pipe is enclosed in a cylindrical casing usually having a wall 4 in. thick and built of segments which are bound together by a wire wrapped spirally around the casing. The casing is lined with bright tin and coated with asphaltum. The pipe is supported on rollers carried in a bracket which fits into the casing. The lengths of casing are tightly fitted together with a male and female joint. This form of conduit is illustrated in Fig. 1 at *A*. The casing rests on a bed of crushed stone with tile drains laid below. The tile drains are of 4-in. field tile or vitrified sewer tile, laid with open joints.

Filler Type: The pipes are supported on expansion rollers properly supported from the conduit or independent masonry base. The pipes are protected by a split-tile conduit, and the entire space between the pipes and the tile is filled with an insulating filler. Thus the pipes are nested and the insulation between them and the tile effectively prevents circulation of air. The conduit is placed on a bed of gravel or crushed rock from 4 to 6 in. thick, which is extended upward so as to come about 2 in. above the parting lines of the tile. A tile underdrain is placed beneath the conduit throughout the entire length and is connected to sewers or to some other point of free discharge. At *B* and *D* in Fig. 1 are shown two forms of tile conduit of the filler type.

Circular Tile or Cast-Iron Conduit: The pipes are carried on expansion rollers supported on a frame which rests entirely on the side shoulders of the base drain foundation. The pipes are protected by a sectional tile conduit, scored for splitting, or a cast-iron conduit, both being of the bell and spigot type. The conduit has a longitudinal side joint

for cementing, after the upper half of conduit is in place, so shaped that the cement is keyed in place while locking the top and bottom half of the conduit together with a water-tight vertical side joint. The cast-iron conduit has special side locking clamps in addition to the vertical side joint. The entire space between the conduit and the pipes is filled with a water-proofed asbestos insulation. The conduit is supported on the base drain foundation, each section resting on two sections of the base drain, thus interlocking. The base drain is so shaped that it provides a cradle for the conduit, resting solidly on the trench bottom and providing adequate drainage area immediately under the conduit. The underdrain is connected to sewers or some other point of free discharge. For tile conduit the base drain is vitrified salt glazed tile and for cast-iron conduit it is either extra heavy tile or cast-iron. A free internal drainage area is also provided to carry away any water that may collect on the inside of the conduit from a leaky pipe or joint in the conduit. Broken stone is filled in around the base drain and up to the vertical side joint. The broken stone is covered with an asphalted filter cloth to prevent sand from sifting through the broken stone and clogging the drainage area of the base drain. The tile conduit is made in 2-ft lengths and the cast-iron conduit in 4-ft lengths, cast in separate top and bottom halves. Special reinforcing ribs give the cast-iron conduit ample strength with minimum weight.

Insulated Tile Type: The insulating material, diatomaceous earth, is molded to the inside of the sectional tile conduit. The space between the pipes and the insulating conduit lining may also be filled with insulation. The pipes are carried on expansion rollers supported on a frame which rests on the side shoulders of the base drain foundation. This type of conduit has the same mechanical features as those described under the heading Circular Tile or Cast-Iron Conduit.

Sectional Insulation Type (Tile or Cast-Iron): Each pipe is insulated in the usual way with any desired type of sectional pipe insulation over which is placed a standard waterproof jacket with cemented joints. The pipes are enclosed in a sectional tile or cast-iron conduit as described under the heading Circular Tile or Cast-Iron Conduit.

Sectional Insulation Type (Tile or Concrete Trench): A type of construction frequently used in city streets, where service connections are required at frequent intervals, the pipes are insulated as described in the preceding paragraph, and are enclosed in a box or trench made either entirely of concrete, or with concrete bottom and specially constructed tile sides and tops. The pipes are supported on roller frames secured in the concrete. At *C* and *E*, Fig. 1, are shown two tile conduits using sectional insulation. In these particular designs the space surrounding the pipe is filled partially or wholly with a loose insulating material. The use of loose material in addition to the sectional insulation is, of course, optional and is only justifiable where high pressure steam is used. The conduit shown at *F* is of a similar type and has the advantage of being made entirely of concrete and other common materials.

Sectional Insulation Type (Bituminized Fibre Conduit): Each pipe is individually insulated and encased in a bituminized fibre conduit. The insulating material is 85 per cent carbonate of magnesia sectional pipe covering, applied in the usual manner as on overhead pipes, except that bands are omitted. After every fifth section of magnesia covering there is applied a short, hollow section of very hard asbestos material in the bottom portion of which rests a grooved-iron plate carrying ball-bearings upon which the pipe rides when expanding or contracting. This short expansion section is of the same outside diameter as the adjacent 85 per cent magnesia covering. Over the pipe covering and expansion device there are placed two layers of bituminized fibre conduit with all joints staggered, and the surface of each conduit is finished with liquid cement. Conduits are placed on a bed of crushed rock or gravel, approximately 6 in. deep, and this is extended upward to about the center line of the conduit when trench is backfilled. Underdrains leading to points of free discharge are placed in the gravel or crushed rock beds.

Special Water-Tight Designs: It is occasionally necessary to install pipes in a very wet ground, which calls for special construction. The ordinary tile or concrete conduit is not absolutely water tight even when laid with the utmost care. The conduit shown at *G*, Fig. 1, is of cast-iron with lead-calked joints and is water-tight if properly laid. It is obviously expensive and is justified only in exceptional cases. A reasonably satisfactory construction in wet ground is the concrete or tile conduit with a water-proof jacket enclosing the pipe and its insulation, and with the interior of the conduit carefully drained to a manhole or sump having an automatic pump. It is useless to install external drain tile when the conduit is actually submerged.

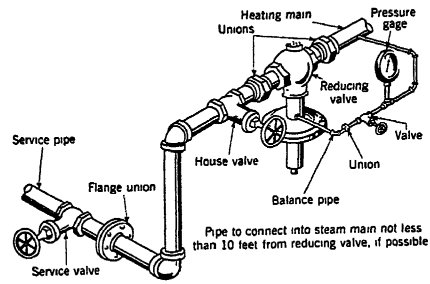


Fig. 2. Connections for Reducing Valves of Size Less thal 4 In.

PIPE TUNNELS

Where steam heating lines are installed in tunnels large enough to provide walking space, the pipes are supported by means of hangers or roller frames on brackets or frame racks at the side or sides of the tunnel. The pipes are insulated with sectional pipe insulation over which is placed a sewed-on, painted canvas jacket or a jacket of asphalt-saturated asbestos water-proofing felt. The tunnel itself is usually built of concrete or brick and water-proofed on the outside with membrane water-proofing.

On account of their relatively high first cost as compared with smaller conduits, walking tunnels are sometimes not installed where provision for the heating lines is the only consideration, but only where they are required to accommodate miscellaneous other services or provide underground passage between buildings.

OVERHEAD DISTRIBUTION

In some industrial and institutional applications, the distribution piping may be installed, entirely or in part, above ground. This method of construction has the advantage of requiring no excavation and being easily maintained.

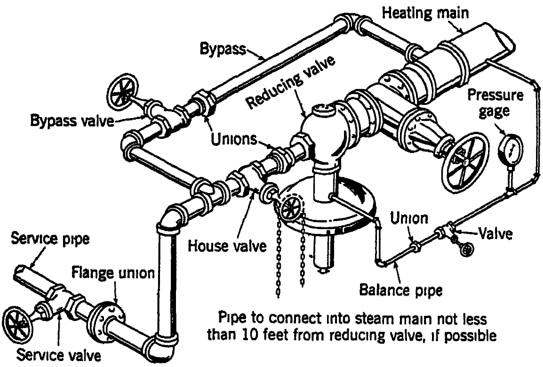


Fig. 3. Connections for Reducing Valves of Size 4 In. and Larger, and for Expanded Valves

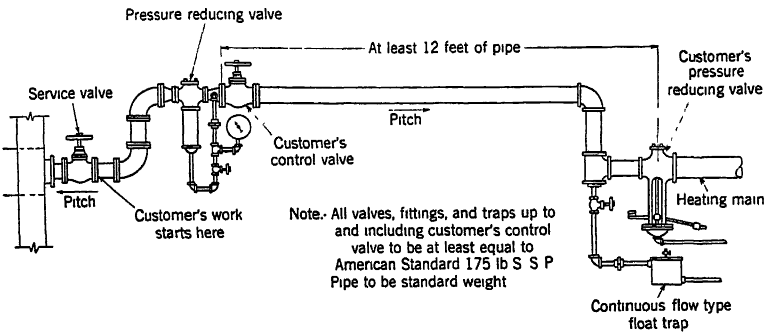


FIG. 4. STEAM SUPPLY CONNECTION WHEN USING TWO REDUCING VALVES

INSIDE PIPING

Figs. 2 and 3 show typical service connections used for low pressure steam service. As shown in Fig. 2, no by-pass is used around the reducing valve on sizes less than 4 in. Fig. 3 illustrates the use of a by-pass around reducing valves 4 in. and larger. This latter construction permits the operation of the line in case of failure in the reducing valve. In the smaller sizes, the reducing valve can be removed, a filler installed, and the house valve used to throttle the flow of steam until repairs are made.

Fig. 4 shows a typical installation used for high pressure steam service. The first reducing valve effects the initial pressure reduction. The second reducing valve reduces the steam pressure to that required.

Most district heating companies enforce certain regulations regarding the consumer's installation, partly to safeguard their own interests but principally to insure satisfactory and economical service to the consumer. There are certain fundamental principles that should be followed in the design of a building heating system which is to be supplied from street mains. Although some of these apply to any building, they have been demonstrated to be especially important when steam is purchased.

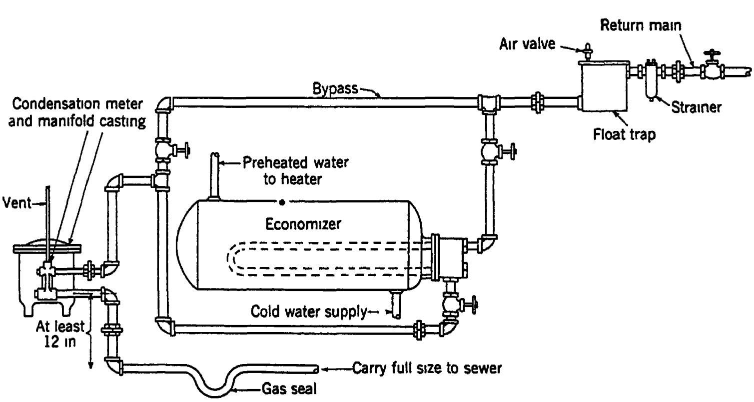


FIG. 5. RETURN PIPING FOR CONDENSATION METER

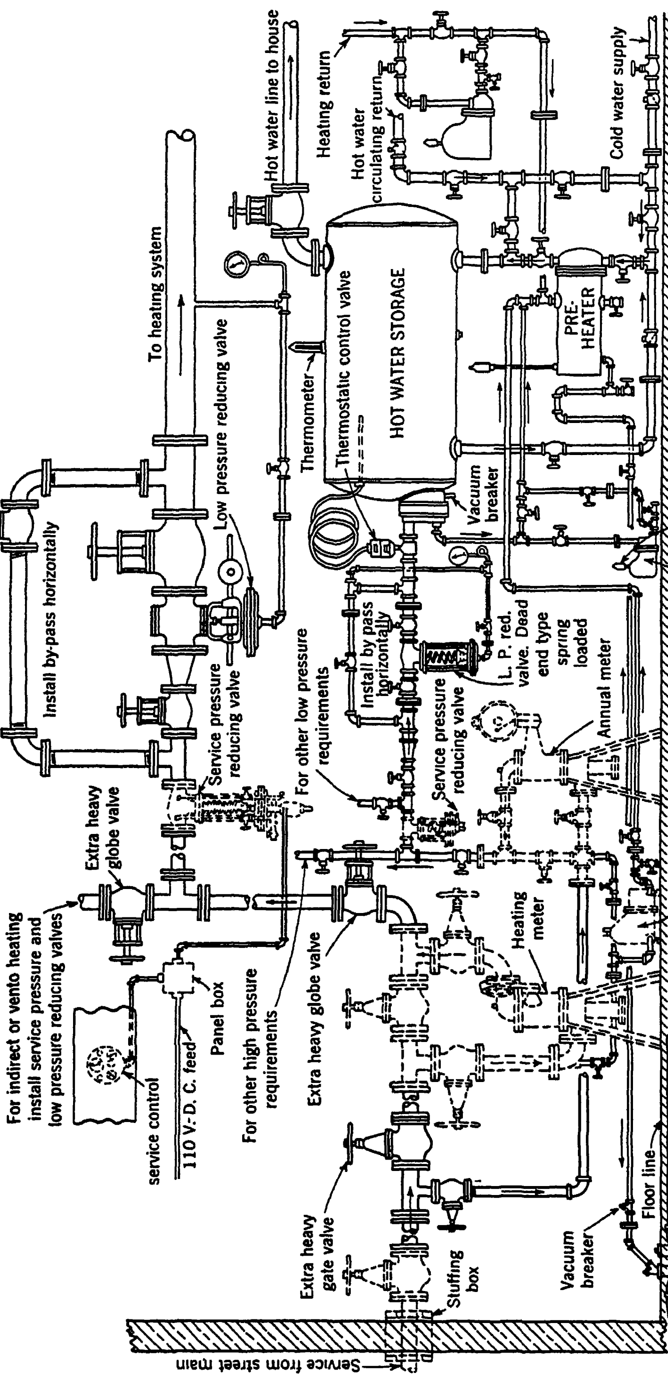


FIG. 6. TYPICAL SERVICE INSTALLATION ALSO SHOWING WATER HEATER AND ECONOMIZER CONNECTIONS

1. Provision should be made for conveniently shutting off the steam supply at night and at other times when heat is not needed.

It has been thoroughly demonstrated that a considerable amount of heat can be saved by shutting off steam at night. Although there is, in some cases, an increased consumption of heat when steam is again turned on in the morning, there is a large net saving which may be explained by the fact that the lower inside temperature maintained during the night obviously results in lower heat loss from the building, and less heat need therefore be supplied.

Steam can be entirely shut off at night in most buildings even in very cold weather without endangering plumbing. It is necessary, however, to have an ample amount of heating surface so that the building can be quickly warmed in the morning. Where the hours of occupancy differ in various parts of the building, it is good practice to install separate supply pipes to the different parts. For example, in an office building with stores or restaurants on the first floor which are open in the evening, a separate main supplying the first floor will permit the steam to be shut off from the remainder of the building in the late afternoon. The division of the building into zones each with a separately controlled heat supply is sometimes desirable, as it permits the heat to be adjusted according to variations in sunshine and wind.

2. Residual heat in the condensate should be salvaged.

This heat may be salvaged by means of a cooling coil, or as is more frequently done, by a water heating economizer (see Fig. 5) which preheats the hot water supply to the building. Fig. 6 shows a typical steam service installation for high pressure steam, complete for steam flow metering, water heating, preheating, automatic heating control, and for using steam for other purposes.

The condensation from the heating system, after leaving the trap, passes through the economizer. The supply to the hot water heater passes through the economizer, absorbing heat from the condensate. If the hot water system in the building is of the recirculating type, the recirculating connection should be tied in *between* the economizer and the water heater proper, not at the economizer inlet, because the recirculated hot water is itself at a high temperature. The number of square feet of heating surface in the economizer should be approximately equal to one per cent of the equivalent square feet of heating surface in the building.

Because of the lack of coincidence between the heating system load and the hot water demand, a greater amount of heat can be extracted from the condensate if storage capacity is provided for the preheated water. Frequently a type of economizer is used in which the coils are submerged in a storage tank.

3. Heat supply should be graduated according to variations in the outside temperature.

This may be done in several ways, as by the use of temperature controls of various types or by orifice systems. Another method which is very simple is the use of an ordinary vacuum return line system in which the pressure in the radiators is varied between a high vacuum and a few pounds pressure, thus producing some control over the heat output of the heating system by varying the temperature of the steam in the radiators. Several proprietary systems are on the market which accomplish this automatically, either with outdoor or indoor controls or a combination of both. One form of control which appears to be well suited for controlling district steam service to a building is the weather compensating control. It regulates the steam supply automatically according to the outdoor temperature, and gives frequent short intervals of intermittent steam supply, and at the same time insures delivery of steam to all the radiators. This type of control can be equipped with time clocks and thermostats to provide a warming-up period in the morning.

Another form of regulation, known as the time-limit control, is sometimes employed for regulating the steam supply from the central station main to the building. Such a control provides an intermittent supply of steam to the radiators either throughout the 24 hours of the day or during the daytime hours only. The setting of a switch may provide no service, continuous service, or periodic service. For the latter, by means of several intermittent settings, steam will be supplied during each period in increments of a certain number of minutes for each successive setting of the switch, steam being shut off during the balance of the period. These settings afford from 15 to 80 per cent of the maximum heating effect required on days of zero temperature. A night switch with a variety of settings may be adjusted so as to maintain throughout the night the intermittent supply called for by the day switch setting, or may be set to interrupt the opera-

tion of the day switch and entirely cut off the supply of steam to the radiation at night during certain hours which are selected by the operating engineer.

The maximum in economical operation and satisfactory heating can only be obtained by the use of some automatic temperature control system.

FLUID METERS

The perfection of fluid meters has contributed more to the advancement of district heating than any other one thing. These meters may be classified as follows:

1. *Positive Meters:* The fluid passes in successive isolated quantities—either weights or volumes. These quantities are separated from the steam and isolated by alternately filling and emptying containers of known capacity.

2. *Differential Meters:* In the differential meter, the quantity of flow is not determined by simple counting, as with the positive meter, but is determined from the action of the steam on the primary element.

Additional sub-divisions of these two general classifications can be made as follows:

- Fluid Meters
  - Positive - Quantity
    - Weighing
      - Weighers
      - Tilting trap
    - Volumetric
      - Rotary
      - Bellows
  - Differential
    - Quantity - Current - Turbine
    - Rate of flow
      - Head (Kinetic)
        - Venturi
        - Flow nozzle
        - Orifice
        - Pitot tube
      - Area (Geometric)
        - Orifice and plug
        - Cylinder and piston
      - Head area (Weir)
        - V-notch
        - Special notch

In selecting a meter for a particular installation, the number of different makes and types of meters suitable for the job is usually limited by one or more of the following considerations:

1. Its use in a new or an old installation.
2. Method to be used in charging for the service.
3. Location of the meter.
4. Large or small quantity to be measured.
5. Temporary or permanent installation.
6. Cleanliness of the fluid to be measured.
7. Temperature of the fluid to be measured.
8. Accuracy expected.
9. Nature of flow: turbulent, pulsating, or steady.
10. Cost.
    (*a*) Purchase price.
    (*b*) Installation cost.
    (*c*) Calibration cost.
    (*d*) Maintenance cost.
11. Servicing facilities of the manufacturer.
12. Pressure at which fluid is to be metered.
13. Type of record desired as to indicating, recording or totalizing.
14. Stocking of repair parts.

15. Use of open jets where steam is to be metered.
16. Metering to be done by one meter or by a combination of meters.
17. Use as a check meter.
18. Its facilities for determining or recording information other than flow.

Condensation Meters

The majority of the meters used by district heating companies in the sale of steam to their customers are condensation meters.

The condensation meter is a popular type for use on small and medium sized installations, where all of the condensate can be brought to a common point for metering purposes. Its simplicity of design, ease in testing, accuracy at all loads, low cost, and adaptability to low pressure distribution has made it standard equipment with many heating companies.

Two types of condensation meters are in general use: the *tilting bucket* meter and the *revolving drum* or *rotor* meter of which there are several makes on the market. Condensation meters should not be operated under

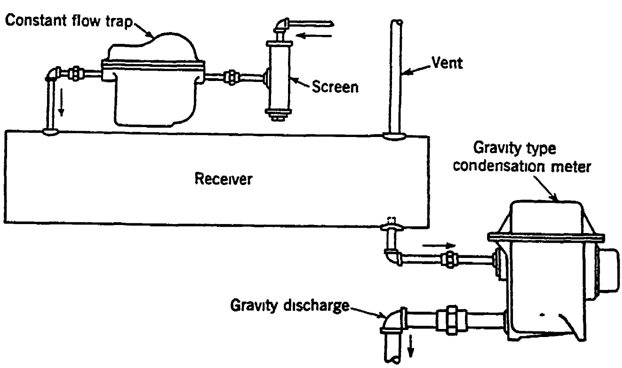


FIG. 7. GRAVITY INSTALLATION FOR CONDENSATION METER USING VENTED RECEIVERS

pressure; they are made for either gravity or vacuum installation. Continuous flow traps are necessary ahead of the meter if a vented receiver is not used. Where bucket traps are used, a vented receiver before the meter is essential. If desirable a receiver may be used with a continuous flow trap, but this is not necessary.

Fig. 7 illustrates a gravity installation using a vented receiver ahead of the meter, while Fig. 8 shows a vacuum installation without a master trap.

Flow Meters

Steam flow meters are available in many types and combinations. The *orifice* and *plug* meter is one in which the steam flow varies directly as the area of the orifice. The vertical lift of the plug, which is proportional to the flow, is transmitted by means of a lever to an indicator and to a pencil arm which records the flow on a strip chart. The total flow over a given period is obtained by measuring the area by using a planimeter on the chart and applying the meter constant.

Fig. 9 shows a typical orifice-type meter connection and indicates typical requirements in the installation of this type of meter.

Flow meters using an orifice, Venturi tube, flow nozzle, or Pitot tube as the primary device are made by a number of manufacturers and can be obtained in either the mechanically or electrically operated type. The electric flow meter makes it possible to locate the instruments at some distance from the primary element.

Flow meters employing the orifice, Venturi tube, flow nozzle or Pitot tube should be so selected as to keep the lower operating range of the load above 20 per cent of the capacity of the meter. This is desirable for accuracy as the differential pressure at light loads is too small to properly actuate the meter. A few general points to be considered in installing a meter of this type are:

1. It is desirable to place the differential medium in a horizontal pipe in preference to a vertical one, where either location is available.

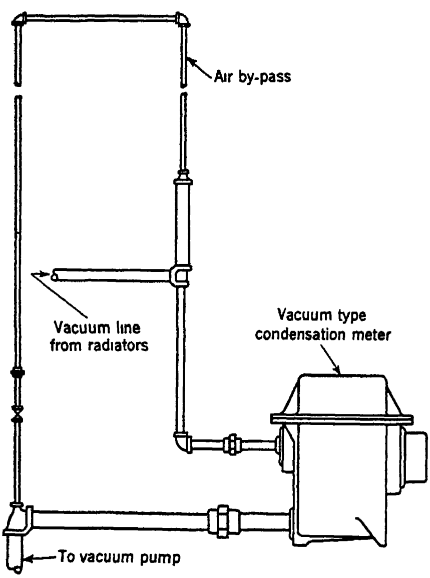


FIG. 8. VACUUM CONDENSATION METER INSTALLATION WITHOUT MASTER TRAP

2. Reservoirs should always be on the same level and installed in accordance with the instructions of the meter company.
3. The meter body should be placed at a lower level than that of the pressure differential medium. Special instructions are furnished where the meter body is above.
4. Meter piping should be kept free from leaks.
5. Sludge should not be permitted to collect in the meter body.
6. The meter body and meter piping should be kept above freezing temperatures.
7. It is best not to connect a meter body to more than one service.
8. Special instructions are furnished for metering a turbulent or pulsating flow.

STEAM REQUIREMENTS

Steam requirements for heating various types of buildings are given in Chapter 11.

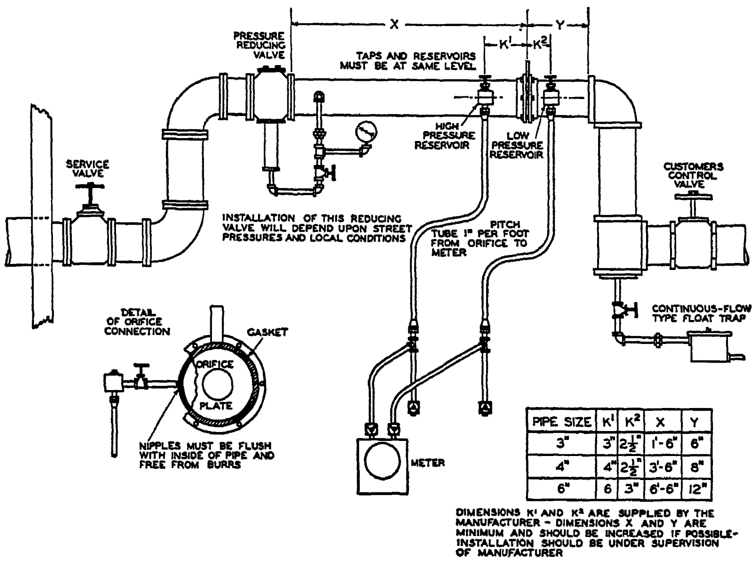
Steam requirements for water heating can be satisfactorily estimated

by using a consumption of 0.0025 lb per day per cubic foot of heated space for office buildings, and 0.0065 lb per day per cubic foot for apartment houses.

Additional data on steam requirements of various types of buildings in a number of cities may be found in the Handbook of the *National District Heating Association.*

RATES

Fundamentally, district heating rates are based upon the same principles as those recognized in the electric light and power industry, the main object being a reasonable return on the investment. However, there are



PIPE SIZE	K^1	K^2	X	Y
3"	3"	2½"	1'-6"	6"
4"	4"	2½"	3'-6"	8"
6"	6	3"	6'-6"	12"

FIG. 9. ORIFICE METER STEAM SUPPLY CONNECTION

other requirements to be met; the rate for each class of service should be based upon the cost to the utility company of the service supplied and upon the value of the service to the consumer, and it must be between these two limits. District heating rates should be designed to produce a sufficient return on the investment regardless of weather conditions, although existing rate schedules do not conform with this principle. Lastly, the rate schedule must be reasonably easy for the intelligent layman to comprehend.

Depreciation should be based on a careful estimate of the life of various elements of the property. Appropriations to reserves should be made, with generosity in good years and with discretion in less favorable years.

Glossary of Terms

Load Factor. The ratio, in per cent, of the average hourly load to the

maximum hourly load. This is usually based on a one year period but may be applied to any specified period.

Demand Factor. The relation between the connected radiator surface or required radiator surface and the demand of the particular installation. It varies from 0.25 to 0.3 lb per hour per square foot of surface.

Diversity Factor. The ratio of the sum of the individual demands of a number of buildings to the actual composite demand of the group.

Types of Rates

A. Flat Rates.
  1. Radiator surface charge. *Obsolescent*

B. Meter Rates.
  1. Straight-line.
  2. Step. *Obsolescent.*
  3. Block.
    (*a*) Class rates.

C. Demand Rates.
  1. Flat demand.
  2. Wright.
  3. Hopkinson.
  4. Doherty (or Three charge)

Straight-Line Meter Rate. The price charged per unit is constant, and the consumer pays in direct proportion to his consumption without regard to the difference in costs of supplying the individual customers.

Block Meter Rate. The pounds of steam consumed by a customer are divided into blocks of thousands of pounds each, and lower rates are charged for each successive block consumed. This type of charge predominates in steam heating rate schedules for it has the advantage of proportioning the bill according to the consumption and the cost of service. It has the disadvantage of not discriminating between customers having a high load factor (relatively low demand) and those having a low load factor (relatively high demand). The utility company must maintain sufficient capacity to serve the high demand customers and the cost of the increased plant investment is divided equally among the users, so the high demand customers are benefitted at the expense of the others.

Demand Rates. These refer to any method of charge based on a measured maximum load during a specified period of time.

The *flat demand rate* is usually expressed in dollars per *M* lb of demand per month or per annum. It is based on the size of a customer's installation, and is seldom used except where a flow meter is not practicable.

The *Wright demand rate* is similar in calculation to the block rate except that it is expressed in terms of hours' use of the maximum demand. It is seldom used but forms the basis for other forms of rates.

The *Hopkinson demand rate* is divided into two elements:

(*a*) A charge based upon the demand, either estimated or measured.
(*b*) A charge based upon the amount of steam consumed.

This rate may be modified by dividing the quantities of steam demanded and consumed into blocks charged for at different rates.

The *Doherty rate* is divided into three elements:

(*a*) A charge based upon demand.
(*b*) A charge based upon steam consumed.
(*c*) A customer charge.

In the *Hopkinson rate*, the last two elements are combined into one element.

Demand rates are comparatively new and are not yet widely used; though they are equitable and competitive they are difficult for the average layman to understand.

They are of benefit to utility companies and to consumers because the investment and operating costs can be divided to suit the particular circumstances into demand, customer, and consumption groups through the use of some modification of the Hopkinson rate. Demand rates are an advantage to the customer in that the use of such a rate reduces the rate per thousand pounds to the long-hour user.

Fuel Price Surcharge. It is usually desirable to establish a rate upon a specified basic cost of fuel to the utility company. Where there are wide variations in the price of fuel, it is also desirable to add a definite charge per *M* lb of steam sold for each increment of increase in the price of fuel. This surcharge automatically compensates for the variations without necessitating frequent changing of the whole rate structure.

UTILIZATION

Considerable savings can be made by the proper and intelligent operation of heating systems. It should be borne in mind that a heating system is designed to heat a building to 70 F inside when the outside temperature is at its lowest point for that particular locality. There is a tendency to overheat the building at any time the outside temperature is above the design temperature unless some method of regulation is used, either automatic or manual.

The general rules for economical operation are as follows:

1. Weatherstrip all windows, and calk all window frames.

2. Provide revolving or vestibule doors on all entrances. Separate shipping and receiving rooms by partitions so that the ever-open large doors will not ventilate the entire building.

3. Keep the radiation near the outside walls, under the windows, if possible.

4. Eliminate all unnecessary ventilation. Ventilating equipment is sized to meet extreme requirements. Do not supply ventilation to a theater or auditorium adequate for an audience of 2000 when there are only 200 present.

5. Determine the hours that heating is required during the day and see that the steam is shut off for the maximum time at night, on Sundays, and holidays.

6. Shut steam off entirely in unoccupied sections of the building, taking care to avoid freezing the water in the plumbing system.

7. Shut off steam during the day whenever possible. During the year steam can be shut off about 55 per cent of the total daytime, and the saving is proportional. An automatic control will do it, but it can be done by hand with amazingly good results.

8. Determine the temperature required for the occupancy of the building. Do not heat a storage garage or a furniture warehouse to the temperature required in a hospital ward.

9. Provide some good means of temperature control.

10. In a hot water heating system keep the temperature of the water down to correspond with existing outdoor temperatures.

11. In a vacuum system maintain a high vacuum. If this is not possible, locate and eliminate all leaks.

12. Install separate lines for those parts of the building that require long-hour or all-night heating. It is much cheaper than heating the entire building all night.

13. See that the entire system responds rapidly when steam is turned on. Locate and eliminate the cause of any sluggish circulation. Balance the radiation, provide adequate air elimination, and correct any trapped run-outs to provide quick system drainage.

14. Keep the system in good repair. Worn, damaged, or defective valves and traps will not function properly.

15. Insulate all steam pipes not used as heating surface.

16. Do not obstruct radiators or prevent the free circulation of air around them; to do so seriously reduces the heating capacity of a radiator.

17. Extract the heat in the condensate for hot water or some other useful purpose.

18. Provide thermometers and recording pressure gages so that the engineer can operate the system with full knowledge of what he is accomplishing.

19. Make all valves and controls convenient and accessible, either direct or through remote control. It is only human nature to delay and avoid doing that which is inconvenient.

20. Keep a daily record consistently, based on weather requirements, and watch it every day.

21. Control the heat supplied to water tanks located on or above the roof. Such tanks require heat to prevent freezing. No heat is required when the temperature in the tank is above 32 F.

22. Investigate every complaint of *no heat* by tenants; find the cause and correct it. Do not overheat an entire building to correct a local condition in one room.

AUTOMATIC TEMPERATURE CONTROL

As stated in Chapter 34, Automatic Control, properly applied to heating, ventilating and air conditioning systems makes possible the maintenance of desired conditions with maximum operating economy.

In addition to the large possibilities for economy, the use of adequate temperature control provides more healthful, comfortable, and efficient working conditions in the buildings because through its use the building is uniformly heated with correct temperatures, and drafts from open windows and overheating are eliminated.

There are many types of temperature control available, each adaptable to a particular type of building, but all require uniform distribution of steam and proper venting.

Before the installation of any type of modern temperature control equipment, it is necessary to see that the heating system is put in good operating condition. In general, the heating system in a building is not given the attention that other mechanical equipment is given because it will continue to function, after a fashion, even though changes in piping, location of radiation, settlement of piping, and the normal wear and tear or other changes have taken place. Through all this depreciation of the system, it becomes more and more costly to operate and parts of the building have to be greatly overheated in order to prevent underheating in a small section of the building. Vents, traps, vacuum pumps, and valves should be given a careful inspection and replaced or repaired if required. The piping should be of adequate size and graded properly. The return piping should have a careful inspection, and any pockets or lifts removed and properly vented. These inspections and repairs are not costly and prevent a much greater outlay in future years. In most cities district heating companies will be willing to make a survey of heating systems and offer recommendations as to operation and changes in piping layout.

The selection of control equipment depends upon the type and size of building and the degree of saving possible.

Chapter 18

PIPE, FITTINGS, WELDING

Pipe Material, Types of Pipe Used, Dimensions of Pipe Commercially Available, Expansion and Flexibility of Pipe, Pipe Threads and Hangers, Types of Fittings, Welding as Applied to Erection of Piping, Valves, Corrosion of Piping

IMPORTANT considerations in the selection and installation of pipe and fittings for heating, ventilating, and air conditioning work are dealt with in this chapter.

PIPE MATERIALS

Use of corrosion-resistant materials for pipe, including special alloy steels and irons, wrought-iron, copper and brass, has increased considerably during the past few years. The recent development of copper, brass, and bronze fittings which can be assembled by soldering or sweating permits the use of thin-wall pipe and thereby has reduced the initial cost of such installation. The following brief discussion indicates the variety of pipe materials and the types of pipe available.

Wrought-Steel Pipe. Because of its low price, the great bulk of wrought pipe used for heating and ventilating work at the present time is of wrought steel. The material used for steel pipe is a mild steel made by the acid-bessemer, the open-hearth, or the electric-furnace process. Ordinary wrought-steel pipe is made either by shaping sheets of metal into cylindrical form and welding the edges together, or by forming or drawing from a solid billet. The former is known as *welded pipe,* the latter as *seamless pipe.*

Many types of welded pipe are available, although the smaller sizes most frequently used in heating and ventilating work are made by the lap-weld, resistance-weld, or butt-weld process. While the lap-weld and resistance-weld processes produce a better weld than the butt type, lap-weld and resistance-weld pipe is seldom manufactured in nominal pipe sizes less than 2 in. Seamless pipe can be obtained in the small sizes at a somewhat higher cost.

Seamless steel pipe is frequently used for high pressure work or where pipe is desired for close coiling, cold bending, or other forming operation. Its advantages are its somewhat greater strength which permits use of a thinner wall and, in the small sizes, its freedom from the occasional tendency of welded pipe to split at the weld when bent.

Wrought-Iron Pipe. Wrought-iron pipe is claimed to be more corrosion-resisting than ordinary steel pipe and therefore its somewhat higher

first cost is said to be justified on the basis of longer life expectancy. Wrought-iron pipe may be identified by the spiral line marked into each length, either knurled into the metal or painted on it in red or other bright color. Otherwise, there is little difference in the appearance of wrought iron and steel pipe, although microscopic examination of polished and etched specimens will readily disclose the difference.

Cast Ferrous Pipe. There are now available several types of cast-ferrous metal pipe made of a good grade of cast-iron with or without additions of nickel, chromium, or other alloy. This pipe is available in sizes from 1½ in. to 6 in., and in standard lengths of 5 or 6 ft with external and internal diameters closely approximating those of extra strong wrought pipe. Cast ferrous pipe may be obtained coupled, beveled for welding, or with ends plain or grooved for the several types of couplings. It is easily cut and threaded as well as welded. The fact that it is readily welded enables the manufacturers to supply the pipe in any lengths practicable for handling.

Alloy Metal Pipe. Steel pipe bearing a small alloy of copper or other alloying element and iron pipe bearing a small alloy of copper and molybdenum have been claimed to possess more resistance to corrosion than plain steel pipe and they are advertised and sold under various trade names.

Copper Pipe and Fittings. Owing to its inherent resistance to corrosion, copper and brass pipe have always been used in heating, ventilating, and water supply installations, but the cost with standard dimensions for threaded connections has been high. The recent introduction of fittings which permit erection by soldering or sweating allows the use of pipe with thinner walls than are possible with threaded connections, thereby reducing the cost of installations.

The initial cost of brass and copper pipe installations generally runs higher than the corresponding job with steel pipe and screwed connections in spite of the use of thin wall pipe, but the corrosive nature of the fluid conveyed or the inaccessibility of some of the piping may warrant use of a more expensive material than plain steel. The advantages of corrosion-resisting pipe and fittings should be weighed against the correspondingly higher initial cost.

COMMERCIAL PIPE DIMENSIONS

The *IPS* dimensions of commercial pipe universally used at the present time conform to the recommendations made by a Committee of the *A.S.M.E.* in 1886. Pipe up to 12 in. in diameter is made in certain definite sizes designated by nominal internal diameter which is somewhat different from the actual internal diameter, depending on the wall thickness required. There are three weights of wrought iron and steel pipe commonly used, known as *standard-weight*, *extra-strong*, and *double extra-strong*. Because of the necessity of maintaining the same external diameter in all three weights for the same nominal size, the added wall thickness is obtained by decreasing the internal diameter. The term *full-weight*, when applied to sizes below 8 in., means that the pipe is up to the nominal weight per foot. When applied to sizes between 8 and 12 in., inclusive, it often indicates that the pipe has the heaviest of several wall

thicknesses listed. In sizes 14 in. and upward, pipe is designated by its outside diameter (O.D.) and the wall thickness is specified.

While the demands for pipe for the heating and ventilating industry are reasonably well served by the *standard-weight* and *extra-strong* pipe, demands for pipe for higher pressures and temperatures in industry resulted in the use of a multiplicity of wall thicknesses for all sizes. Even in heating installations, the erection of piping by welding was deemed to

TABLE 1. DIMENSIONS OF SCHEDULES 30 AND 40 AND STANDARD WEIGHT PIPE[a]

Size	Diameter In.			Weight per Ft, Lb			Circumference, In.		Transverse Area, Sq In.			Length of Pipe, Ft per Sq Ft		Length of Pipe Ft Containing 1 Cu Ft	Weight of Water, Lb per Ft
	External	Internal	Thickness[b], In.	Plain Ends	Threads and Couplings	Threads per In.	External	Internal	External	Internal	Metal	External Surface	Internal Surface		
1/8	0.405	0.269	**0.068**	0.244	0.245	27	1.272	0.845	0.129	0.057	0.072	9.431	14.199	2533.775	0.025
1/4	0.540	0 364	**0.088**	0 424	0.425	18	1.696	1.144	0 229	0.104	0.125	7.073	10.493	1383.789	0.045
3/8	0 675	0.493	**0.091**	0.567	0.568	18	2.121	1.549	0.358	0.191	0.167	5.658	7.748	754.360	0.083
1/2	0.840	0.622	**0.109**	0.850	0.852	14	2.639	1.954	0 554	0.304	0.250	4.547	6.141	473.906	0.132
3/4	1.050	0.824	**0.113**	1.130	1.134	14	3.299	2 589	0.866	0.533	0.333	3.637	4.635	270.034	0.231
1	1.315	1.049	**0.133**	1.678	1.684	11½	4.131	3.296	1.358	0 864	0.494	2.904	3.641	166.618	0.375
1¼	1.660	1.380	**0.140**	2 272	2.281	11½	5.215	4.335	2.164	1.495	0.669	2.301	2.768	96.275	0.65
1½	1.900	1.610	**0.145**	2.717	2.731	11½	5.969	5.058	2.835	2.036	0.799	2.010	2.372	70.733	0.88
2	2.375	2.067	**0.154**	3.652	3.678	11½	7.461	6.494	4.430	3.355	1.075	1.608	1.847	42.913	1.45
2½	2.875	2.469	**0.203**	5.793	5.819	8	9.032	7.757	6.492	4.788	1.704	1.328	1.547	30.077	2.07
3	3.500	3.068	**0.216**	7.575	7.616	8	10.996	9.638	9.621	7.393	2.228	1 091	1.245	19.479	3.20
3½	4.000	3.548	**0.226**	9.109	9 202	8	12.566	11.146	12 566	9.886	2.680	0.954	1.076	14.565	4.29
4	4.500	4.026	**0.237**	10.790	10.889	8	14.137	12.648	15.904	12 730	3.174	0.848	0.948	11.312	5.50
5	5.563	5.047	**0.258**	14.617	14.810	8	17.477	15.856	24.306	20.006	4.300	0.686	0.756	7.198	8.67
6	6.625	6.065	**0.280**	18.974	19.185	8	20.813	19.054	34.472	28.891	5.581	0.576	0.629	4.984	12.51
8[c]	8.625	8.071	0.277	24.696	25.000	8	27.096	25.356	58.426	51.161	7.265	0.443	0.473	2.815	22.18
8	8.625	7.981	**0.322**	28.554	28.809	8	27.096	25.073	58.426	50.027	8.399	0.443	0.478	2.878	21.70
10	10.750	10.192	0.279	31.201	32.000	8	33.772	32.019	90.763	81.585	9.178	0.355	0.374	1.765	35.37
10[c]	10.750	10.136	0.307	34.240	35.000	8	33.772	31.843	90.763	80.691	10.072	0.355	0.376	1.785	34.95
10	10.750	10.020	**0.365**	40.483	41.132	8	33.772	31.479	90.763	78.855	11.908	0.355	0.381	1.826	34.20
12[c]	12.750	12.090	0.330	43.773	45.000	8	40.055	37.982	127.676	114.800	12.876	0.299	0.315	1.254	49.70
12	12.750	12.000	0.375	49.562	50.706	8	40.055	37.699	127.676	113.097	14.579	0.299	0.318	1.273	49.00

[a]Standard-weight wrought-iron pipe has approximately the same wall thicknesses and weights as contained herein for steel pipe. For exact dimensions, see *American Standard for Wrought-Iron and Wrought-Steel Pipe, A.S.A.* B36.10.

[b]Thicknesses shown in bold face type are identical with thicknesses for Schedule 40 pipe of *A S.A.* B36.10.

[c]Same as Schedule 30, *A.S.A.* B36.10.

warrant the use of pipe lighter than standard weight. For these reasons, a *Sectional Committee on Standardization of Wrought-Iron and Wrought-Steel Pipe and Tubing* functioning under the procedure of the *American Standards Association* was appointed to standardize the dimensions and materials of pipe.

The pipe standard recommended by that sectional committee has set up several schedules of pipe including standard-weight and extra-strong thicknesses which are now included in Schedules 40 and 80, respectively. Dimensions and other useful data for standard-weight and extra-strong pipe are given in Tables 1 and 2. Table 3 from *A.S.T.M.* Specifications

A53 and A120 combines the schedule thicknesses of the *American Standard for Wrought-Iron and Wrought-Steel Pipe, A.S.A.* B36.10, and the old series designations.

Standard-weight pipe is generally furnished with threaded ends in random lengths of 16 to 22 ft, although when ordered with plain ends, 5 per cent may be in lengths of 12 to 16 ft. Five per cent of the total number of lengths ordered may be *jointers* which are two pieces coupled together. Extra-strong pipe is generally furnished with plain ends in

TABLE 2. DIMENSIONS OF SCHEDULES 60 AND 80 AND EXTRA-STRONG PIPE[a]

Size	Diameter, In. External	Diameter, In. Internal	Thickness[b], In.	Weight per Ft Plain Ends, Lb	Circumference, In. External	Circumference, In. Internal	Transverse Area, Sq In. External	Transverse Area, Sq In. Internal	Transverse Area, Sq In. Metal	Length of Pipe, Ft per Sq Ft External Surface	Length of Pipe, Ft per Sq Ft Internal Surface	Length of Pipe, Ft Containing 1 Cu Ft	Weight of Water, Lb per Ft
1/8	0.405	0.215	**0.095**	0.314	1.272	0.675	0.129	0.036	0.093	9.431	17.766	3966.39	0.016
1/4	0.540	0.302	**0.119**	0.535	1.696	0.949	0.229	0.072	0.157	7.073	12.648	2010.290	0.031
3/8	0 675	0.423	**0.126**	0.738	2.121	1.329	0.358	0.141	0.217	5.658	9.030	1024.689	0 061
1/2	0.840	0.546	**0.147**	1.087	2.639	1 715	0.554	0.234	0.320	4.547	6.995	615.017	0.102
3/4	1.050	0.742	**0.154**	1.473	3.299	2.331	0.866	0.433	0.433	3.637	5.147	333.016	0.188
1	1.315	0.957	**0.179**	2.171	4.131	3.007	1.358	0.719	0.639	2.904	3.991	200.193	0.312
1 1/4	1.660	1.278	**0.191**	2.996	5.215	4.015	2.164	1.283	0.881	2.301	2.988	112.256	0.56
1 1/2	1.900	1.500	**0.200**	3.631	5.969	4.712	2.835	1.767	1.068	2.010	2.546	81.487	0.77
2	2.375	1.939	**0.218**	5.022	7.461	6.092	4.430	2.953	1.477	1.608	1.969	48.766	1.28
2 1/2	2.875	2.323	**0.276**	7.661	9.032	7.298	6.492	4.238	2.254	1.328	1.644	33.976	1.87
3	3.500	2.900	**0.300**	10.252	10.996	9.111	9.621	6.605	3.016	1.091	1.317	21.801	2.86
3 1/2	4.000	3.364	**0.318**	12 505	12.566	10.568	12.566	8.888	3.678	0.954	1.135	16.202	3 84
4	4.500	3.826	**0.337**	14.983	14.137	12.020	15.904	11.497	4.407	0.848	0.998	12.525	4.98
5	5.563	4.813	**0.375**	20.778	17.477	15.120	24.306	18.194	6.112	0.686	0.793	7.915	7.88
6	6.625	5.761	**0.432**	28.573	20.813	18.099	34.472	26.067	8.405	0.576	0.663	5.524	11.29
8	8.625	7.625	**0.500**	43.388	27.096	23.955	58.426	45.663	12.763	0.443	0.500	3.154	19.78
10[c]	10.750	9.750	0.500	54.735	33.772	30.631	90.763	74.662	16.101	0.355	0.391	1.929	32.35
12	12.750	11.750	0.500	65.415	40.055	36.914	127.676	108.434	19.242	0.299	0.325	1.328	46 92

[a]Extra-strong wrought-iron pipe has approximately the same wall thicknesses and weights as contained herein for steel pipe. For exact dimensions, see *American Standard for Wrought-Iron and Wrought-Steel Pipe, A.S.A.* B36.10.

[b]Thicknesses shown in bold face type are identical with thicknesses for Schedule 80 pipe for *A.S.A* B36.10.

[c]Same as Schedule 60, *A.S.A.* B36.10.

random lengths of 12 to 22 ft, although 5 per cent may be in lengths of 6 to 12 ft.

In addition to *IPS* copper pipe, several varieties of copper tubing are in use with either flared or compression couplings or soldered joints. Dimensions of copper water tubing intended for plumbing, underground water service, fuel-oil lines, gas lines, etc., have been standardized by the U. S. Government and the *American Society for Testing Materials.* There are three standard wall-thickness schedules of copper water tubing classified in accordance with their principal uses as follows:

Type *K*—Designed for underground services and general plumbing service.

Type *L*—Designed for general plumbing purposes.

Type *M*—Designed for use with soldered fittings only.

In general, Type *K* is used where corrosion conditions are severe, and Types *L* and *M* where such conditions may be considered normal as, for instance, in heating work. Types *K* and *L* are available in both hard and soft tempers; Type *M* is available only in hard temper. Where flexibility is essential as in hidden replacement work or where as few joints as possible are desired as in fuel-oil lines, the soft temper is commonly used. New or exposed work generally employs copper pipe of a hard temper. All three classes are extensively used with soldered fittings.

TABLE 3. STANDARD WEIGHTS AND DIMENSIONS OF WELDED AND SEAMLESS STEEL PIPE[a]

Size	Outside Diameter, In.	No. of Threads per In.	Standard-Weight Pipe				Extra-Strong Pipe				Double Extra-Strong Pipe[b]	
			Schedule 30		Schedule 40		Schedule 60		Schedule 80			
			Wall Thickness, In.	Weight per Ft, Lb T & C	Wall Thickness, In.	Weight per Ft, Lb T & C	Wall Thickness, In.	Weight per Ft, Lb Plain Ends	Wall Thickness, In.	Weight per Ft, Lb Plain Ends	Wall Thickness, In.	Weight per Ft, Lb Plain Ends
1/8	0.405	27	...	...	0.068	0.25	...	...	0.095	0.31		...
1/4	0.540	18	...	...	0.088	0.43		...	0.119	0.54	...	...
3/8	0.675	18			0.091	0.57			0.126	0.74		...
1/2	0.840	14			0.109	0.85	...	...	0.147	1.09	0.294	1.71
3/4	1.050	14	...	...	0.113	1.13		...	0.154	1.47	0.308	2.44
1	1.315	11½	...		0.133	1.68	...	...	0.179	2.17	0.358	3.66
1¼	1.660	11½	...		0.140	2.28	...	...	0.191	3.00	0.382	5.21
1½	1.900	11½			0.145	2.73	...		0.200	3.63	0.400	6.41
2	2.375	11½	...		0.154	3.68		...	0.218	5.02	0.436	9.03
2½	2.875	8			0.203	5.82	...		0.276	7.66	0.552	13.70
3	3.500	8			0.216	7.62	...	...	0.300	10.25	0.600	18.58
3½	4.000	8	...	...	0.226	9.20	...		0.318	12.51	0.636	22.85
4	4.500	8		...	0.237	10.89	...		0.337	14.98	0.674	27.54
5	5.563	8			0.258	14.81	...	...	0.375	20.78	0.750	38.55
6	6.625	8			0.280	19.19			0.432	28.57	0.864	53.16
8	8.625	8	0.277	25.00	0.322	28.81			0.500	43.39	0.875	72.42
10[c]	10.750	8	0.307	35.00	0.365	41.13	0.500	54.74	...			...
12[d]	12.750	8	0.330	45.00	0.375	50.71	0.500[d]	65.41	...	...		...

From Standard Specifications for Welded and Seamless Steel Pipe of the *American Society for Testing Materials*, *A.S.T.M.* Designation A120.

[a]Sizes larger than those shown in the table are measured by their outside diameter, such as 14 in. outside diameter, etc. These larger sizes will be furnished with plain ends, unless otherwise specified. The weights will correspond to the manufacturers' published standards although it is possible to calculate the theoretical weights for any given size and wall thickness on the basis of 1 cu in. of steel weighing 0.2833 lb.

[b]The American Standard for Wrought-Iron and Wrought-Steel Pipe *A.S.A.* B36.10-1939 has assigned no schedule number to *Double Extra-Strong* pipe.

[c]A 10 in. Standard Weight pipe is also available with 0.279 in. wall thickness, but this wall is not covered by a Schedule Number.

[d]Owing to a departure from the *Standard-Weight* and *Extra-Strong* wall thicknesses for the 12 in. nominal size, Schedules 40 and 60, Table 2 of the *A.S.A.* B36.10-1939, Standard for Wrought-Iron and Wrought-Steel pipe, the regular *Standard* and *Extra-Strong* wall thicknesses (0.375 in. and 0.500 in.) have been substituted.

Standard dimensions, weights, and diameter and wall-thickness tolerances for these classes of copper tubing are given in Table 4. Copper pipe is also available with dimensions of steel pipe.

Refrigeration lines used in connection with air conditioning equipment also employ copper tubing extensively. For refrigeration use where tubing absolutely free from scale and dirt is required, bright annealed copper tubing that has been deoxidized is used. This tubing is available in a variety of sizes and wall thicknesses.

TABLE 4. STANDARD DIMENSIONS AND WEIGHTS, AND TOLERANCES IN DIAMETER AND WALL THICKNESS FOR COPPER WATER TUBES[a]

(All tolerances in this table are plus and minus except as otherwise indicated)

Standard Water Tube Size, In.	Actual Outside Diameter In.	Average Outside Diameter Tolerance, In ± Annealed	Average Outside Diameter Tolerance, In ± Drawn Temper	Wall Thickness, In. Type K Nominal	Type K Tolerance ±	Type L Nominal	Type L Tolerance ±	Type M Nominal	Type M Tolerance ±	Theoretical Weight, Lb per Ft Type K	Type L	Type M
1/8	0.250	0 002	0.001	0 032	0.003	0.025	0 0025	0.025	0.0025	0 085	0 068	0.068
1/4	0 375	0.002	0.001	0.032	0.004	0.030	0.0035	0.025	0.0025	0 134	0.126	0.107
3/8	0.500	0 0025	0.001	0 049	0.004	0.035	0.0035	0.025	0.0025	0.269	0.198	0 145
1/2	0.625	0.0025	0.001	0.049	0.004	0.040	0.0035	0.028	0.0025	0.344	0.285	0.204
5/8	0 750	0.0025	0.001	0.049	0.004	0.042	0.0035	0.030	0.0025	0.418	0 362	0.263
3/4	0.875	0 003	0.001	0.065	0.0045	0 045	0.004	0.032	0.003	0.641	0.455	0.328
1	1.125	0.0035	0 0015	0.065	0.0045	0 050	0 004	0.035	0.0035	0.839	0.655	0.465
1 1/4	1.375	0 004	0.0015	0.065	0.0045	0.055	0 0045	0.042	0.0035	1.04	0.884	0.682
1 1/2	1.625	0.0045	0.002	0.072	0.005	0.060	0.0045	0.049	0.004	1.36	1.14	0.940
2	2.125	0.005	0.002	0.083	0.007	0.070	0.006	0.058	0.006	2.06	1.75	1.46
2 1/2	2 625	0.005	0.002	0.095	0.007	0.080	0.006	0.065	0 006	2.93	2.48	2.03
3	3 125	0.005	0.002	0.109	0.007	0.090	0.007	0.072	0.006	4.00	3.33	2.68
3 1/2	3 625	0.005	0 002	0.120	0.008	0.100	0.007	0.083	0.007	5.12	4.29	3.58
4	4.125	0.005	0.002	0.134	0.010	0 110	0.009	0.095	0.009	6.51	5.38	4.66
5	5.125	0.005	0.002	0.160	0.010	0.125	0.010	0.109	0.009	9.67	7.61	6.66
6	6.125	0 005	0.002	0.192	0.012	0.140	0.010	0.122	0.010	13.9	10 2	8.92
8	8 125	0 006	+0.002 −0.004	0.271	0.016	0.200	0.014	0.170	0.014	25.9	19 3	16.5
10	10.125	0.008	+0.002 −0.006	0.338	0.018	0.250	0.016	0.212	0.015	40.3	30.1	25.6
12	12.125	0.008	+0.002 −0.006	0.405	0.020	0.280	0.018	0.254	0.016	57.8	40.4	36.7

[a]From Standard Specifications for Copper Water Tube of the *American Society for Testing Materials, A.S.T.M.* Designation B88-39.

NOTE 1:—For copper gas and oil burner tubes, the tolerances shown above for various wall thicknesses (type K) apply irrespective of diameter.

NOTE 2:—For tubes other than round no standard tolerances are established. These tolerances do not apply to condenser and heat exchanger tubes.

EXPANSION AND FLEXIBILITY

The increase in temperature of a pipe from room temperature to an operating steam or water temperature 100 F or more above room temperature results in an increase in length of the pipe for which provision must be made. The amount of linear expansion (or contraction in the case of refrigeration lines) per unit length of material per degree change in temperature is termed the *coefficient of linear expansion* of that material, or commonly, the *coefficient of expansion.* This coefficient varies with the material.

The linear expansion of cast-iron, steel, wrought-iron, and copper pipe, the materials most frequently used in heating and ventilating work, can be determined from Table 5, which was computed from Equation 1.

$$L_t = L_0 \left[1 + a\left(\frac{t-32}{1000}\right) + b\left(\frac{t-32}{1000}\right)^2\right] \quad (1)$$

where

L_t = length at temperature t degrees Fahrenheit, feet.
L_o = length at 32 F, feet.
t = final temperature, degrees Fahrenheit.
a and b are constants as given in the tabulation following.

Metal	a	b
Cast-Iron	0.005441	0.001747
Steel	0.006212	0.001623
Wrought-Iron	0.006503	0.001622
Copper	0.009278	0.001244

The three methods by which the elongation due to thermal expansion may be taken care of are:

1. Expansion joints.
2. Swivel joints.
3. Inherent flexibility of the pipe itself utilized through pipe bends, right-angle turns, or offsets in the line.

Expansion joints of the slip-sleeve, diaphragm, or corrugated types made of copper, rubber, or other gasket material are all used for taking up expansion, but generally only for low pressures or where the inherent flexibility of the pipe cannot readily be used as in underground steam or hot water distribution lines.

Swivel joints are used extensively in low-pressure steam and hot water heating systems and in hot water supply lines. The swivel joints absorb the expansive movement of the pipe by the turning of threaded joints. In many cases the straight pipe in the offset of a swivel joint is sufficiently flexible to take up the expansion without developing enough thrust to produce swiveling in the threaded joint. This is preferable since continued turning in the threaded joint may in time result in a leak, particularly when the pressure is high. The amount of elongation which a swivel joint can take up is controlled by the length of the swing piece employed and by the lateral displacement which is permissible in the long pipe runs.

Probably the most economical method of providing for expansion of piping in a long run is to take advantage of the directional changes which must necessarily occur in the piping and proportion the offsets so that sufficient flexibility is secured. Ninety-degree bends with long, straight tangents in either a horizontal or a vertical plane are an excellent means for securing adequate flexibility with larger sizes of pipe. When flexibility cannot be obtained in this manner, it is necessary to make use of some type of expansion bend. The exact calculation of the size of expansion bends required to take up a given amount of thermal expansion is relatively complicated[1]. The following approximate method, however,

[1]See (1) Piping Handbook, by Walker and Crocker (McGraw-Hill Co.); (2) A Manual for The Design of Piping for Flexibility by the Use of Graphs, by E. A. Wert, S. Smith, E. T. Cope, published by The Detroit Edison Company.

TABLE 5. THERMAL EXPANSION OF PIPE IN INCHES PER 100 FT[a]

(For superheated steam and other fluids refer to temperature column)

Saturated Steam			Elongation in Inches per 100 ft from −20 F up				Saturated Steam		Elongation in Inches per 100 ft from −20 F up			
Vacuum Inches of Hg.	Pressure Pounds per Square Inch Gage	Temperature Degrees Fahrenheit	Cast-Iron Pipe	Steel Pipe	Wrought-Iron Pipe	Copper Pipe	Pressure Pounds per Square Inch Gage	Temperature Degrees Fahrenheit	Cast-Iron Pipe	Steel Pipe	Wrought-Iron Pipe	Copper Pipe
		−20	0	0	0	0	664.3	500	3.847	4.296	4.477	6.110
		0	0.127	0.145	0.152	0.204	795.3	520	4.020	4.487	4.677	6.352
		20	0.255	0.293	0.306	0.442	945.3	540	4.190	4.670	4.866	6.614
		40	0.390	0.430	0.465	0.655	1115.3	560	4.365	4.860	5.057	6.850
29.39	....	60	0.518	0.593	0.620	0.888	1308.3	580	4.541	5.051	5.268	7.123
28.89	....	80	0.649	0.725	0.780	1.100	1525.3	600	4.725	5.247	5.455	7.388
27.99	....	100	0.787	0.898	0.939	1.338	1768.3	620	4.896	5.437	5.660	7.636
26.48	....	120	0.926	1.055	1.110	1.570	2041.3	640	5.082	5.627	5.850	7.893
24.04	....	140	1.051	1.209	1.265	1.794	2346.3	660	5.260	5.831	6.067	8.153
20.27	....	160	1.200	1.368	1.427	2.008	2705	680	5.442	6.020	6.260	8.400
14.63	....	180	1.345	1.528	1.597	2.255	3080	700	5.629	6.229	6.481	8.676
6.45	....	200	1.495	1.691	1.778	2.500		720	5.808	6.425	6.673	8.912
	2.5	220	1.634	1.852	1.936	2.720		740	6.006	6.635	6.899	9.203
	10.3	240	1.780	2.020	2.110	2.960		760	6.200	6.833	7.100	9.460
	20.7	260	1.931	2.183	2.279	3.189		780	6.389	7.046	7.314	9.736
	34.5	280	2.085	2.350	2.465	3.422		800	6.587	7.250	7.508	9.992
	52.3	300	2.233	2.519	2.630	3.665		820	6.779	7.464	7.757	10.272
	74.9	320	2.395	2.690	2.800	3.900		840	6.970	7.662	7.952	10.512
	103.3	340	2.543	2.862	2.988	4.145		860	7.176	7.888	8.195	10.814
	138.3	360	2.700	3.029	3.175	4.380		880	7.375	8.098	8.400	11.175
	180.9	380	2.859	3.211	3.350	4.628		900	7.579	8.313	8.639	11.360
	232.4	400	3.008	3.375	3.521	4.870		920	7.795	8.545	8.867	11.625
	293.7	420	3.182	3.566	3.720	5.118		940	7.989	8.755	9.089	11.911
	366.1	440	3.345	3.740	3.900	5.358		960	8.200	8.975	9.300	12.180
	451.3	460	3.511	3.929	4.096	5.612		980	8.406	9.196	9.547	12.473
	550.3	480	3.683	4.100	4.280	5.855		1000	8.617	9.421	9.776	12.747

[a]From *Piping Handbook*, by Walker and Crocker. This table gives the expansion from −20 F to the temperature in question. To obtain the amount of expansion between any two temperatures take the difference between the figures in the table for those temperatures. For example, if a steel pipe is installed at a temperature of 60 F and is to operate at 300 F, the expansion would be 2.519 − 0 593 = 1.926 in.

has been found to give reasonably good results and is deemed to be sufficiently accurate for most heating work.

Fig. 1 shows several types of expansion bends commonly used for taking up thermal expansion. The amount of pipe, L, required in each of these bends may be computed from Equation 2.

$$L = 6.16 \sqrt{D \Delta} \tag{2}$$

where

L = length of pipe, feet.
D = outside diameter of the pipe used, inches.
Δ = the amount of expansion to be taken up, inches.

This formula, based on the use of mild-steel pipe with wall thicknesses not heavier than extra-strong, assumes a maximum safe value of fiber stress of 16,000 lb per square inch. When square type bends are used, the width of the bend should not exceed about two times the height. It is

further assumed that the corners are made with screwed or flanged elbows or with arcs of circles having radii five to six times the pipe diameter. Use of welding elbows with radii of 1½ times the pipe diameter will decrease the end thrusts somewhat but will raise the fiber stress correspondingly.

All risers must be anchored and safeguarded so that the difference in length when hot from the length when cold shall not disarrange the normal and orderly provisions for drainage of the branches.

Proper anchoring of piping is especially necessary with light-weight radiators, to allow for freedom of expansion in order that no pipe strain will distort the radiators. When expansion strains from the pipes are

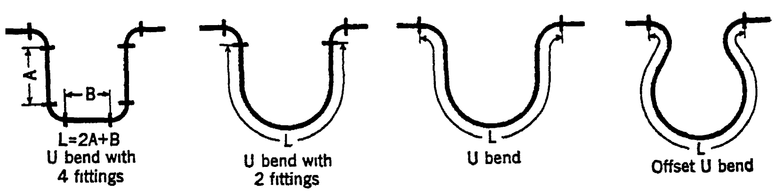


FIG. 1. MEASUREMENT OF L ON VARIOUS PIPE BENDS

permitted to reach these light metal heaters they usually emit sounds of distress which are exceedingly troublesome.

PIPE THREADS

All threaded pipe for heating and ventilating installations uses the American Standard taper pipe thread which is made with a taper of 1 in 16 measured on the diameter of the pipe so as to secure a tight joint. Threads of fittings are tapped to the same taper. The number of threads per inch varies with the different pipe sizes. All threaded pipe should be made up with a thread paste suitable for the service under which the pipe is to be used.

HANGERS AND SUPPORTS

Heating system piping requires careful and substantial support. Where changes in temperature of the line are not large, such simple methods of support may be utilized as hanging the line by means of rods or perforated strip from the building structure, or supporting it by brackets or on piers.

When fluids are conveyed at temperatures of 150 F or above, however, hangers or supporting equipment must be fabricated and assembled to permit free expansion or contraction of the piping. This can be accomplished by the use of long rod hangers, spring hangers, chains, hangers or supports fitted with rollers, machined blocks, elliptical or circular rings of larger diameter than the pipe giving contact only at the bottom, or trolley hangers. In all cases, allowance should be made for rod clearance to permit swinging without setting up severe bending action in the rods.

For pipes of small size, perforated metal strip is often used. For horizontal mains, the rod or strip usually is attached to the joists or steel work of the floor above. For long runs of vertical pipe subject to considerable thermal expansion, either the hangers should be designed to

prevent excessive load on the bottom support when expansion takes place, or the bottom support should be designed to withstand the entire load.

TYPES OF FITTINGS

Fittings for joining the separate lengths of pipe together are made in a variety of forms, and are either screwed or flanged, the former being generally used for the smaller sizes of pipe up to and including 3½ in., and the latter for the larger sizes, 4 in. and above. Screwed fittings of large size as well as flanged fittings of small size are also made and are used for certain classes of work at the proper pressure.

The material used for fittings is generally cast-iron, but in addition to this, malleable-iron, steel and steel alloys are also used, as well as various grades of brass or bronze. The material to be used depends on the character of the service and the pressure.

As in the case of pipe, there are several weights of fittings manufactured. Recognized American Standards for the various weights are as follows:

Cast-iron pipe flanges and flanged fittings for 25 lb (sizes 4 in. and larger), 125 lb, and 250 lb maximum saturated steam pressure.

Malleable iron screwed fittings for 150 lb maximum saturated steam pressure.

Cast-iron screwed fittings for 125 and 250 lb maximum saturated steam pressure.

Steel flanged fittings for 150 and 300 lb maximum steam service pressure.

The allowable cold water working pressures for these standards vary from 43 lb for the 25 lb standard to 500 lb for the 300 lb steel standard.

Screwed fittings include: nipples or short pieces of pipe of varying lengths; couplings, usually of wrought-iron only; elbows for turning angles of either 45 deg or 90 deg; return bends, which may be of either the close or open pattern, and may be cast with either a back or side outlet; tees; crosses; laterals or *Y* branches; and a variety of plugs, bushings, caps, lock-nuts, flanges and reducing fittings. Reducing fittings as well as bushings, both of which are used in changing from one pipe size to another, may have the smaller connection tapped eccentrically to permit free drainage of the water of condensation in steam lines or free escape of air in water lines.

Fittings for copper tubing are available in the soldered, flared, or compression types. Illustrations of each of these types are shown in Fig. 2. Fittings for copper pipe of *IPS* dimensions are available in screwed or soldered types of connection.

American Standard A40.3-1941 has been prepared to cover certain dimensions of soldered-joint fittings for copper water tube including (1) detailed dimensions of the bore, (2) minimum specifications for materials, (3) minimum inside diameter of the fittings, (4) metal thickness for both wrought-metal and cast-brass fittings, and (5) general dimensions for cast-brass fittings including center-to-shoulder dimensions for both straight and reducing cast fittings. Table 6 from *A.S.A.* Standard A40. 3-1941 contains dimensions for soldered joint elbows, tees, crosses, and 45 deg elbows.

The compression type fitting is generally limited to smaller size tubing while the flared and soldered types are used in both large and small sizes.

An American Standard, *A.S.A.* A40.2-1936 has been prepared to standardize dimensions for brass fittings for flared copper water tubes. Flared tube fittings are widely used in refrigerating work where *S.A.E.* dimensions and a 45-deg flare render most fittings interchangeable, although for refrigeration use, thread fits and tolerances on thread gages must be maintained within close limits. Brass fittings with *S.A.E.* dimensions are not interchangeable with the American Standard fittings for water tubes.

Ammonia pipe fittings made of cast-iron were formerly used extensively in handling refrigerants in large installations. Replacement of ammonia by other refrigerants operating at lower pressures has seriously curtailed

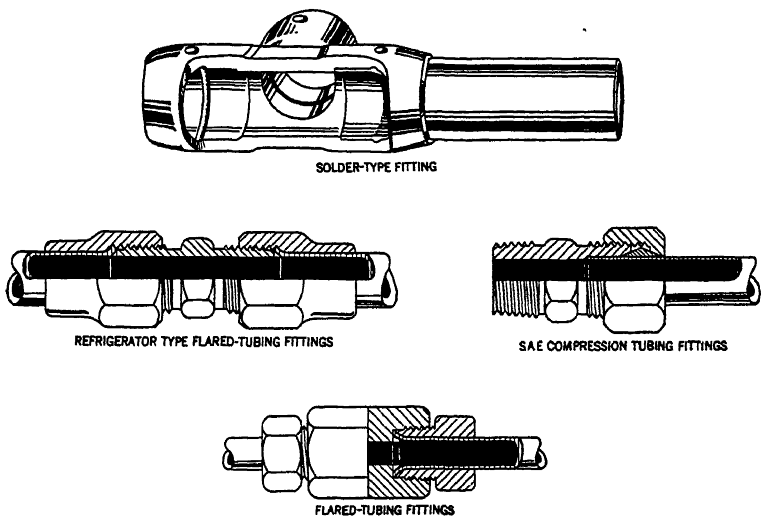


FIG. 2. COPPER OR BRASS TUBING FITTINGS

the market for these fittings. For this reason formulation of an American Standard for these fittings was abandoned by the *A.S.A.* in 1936.

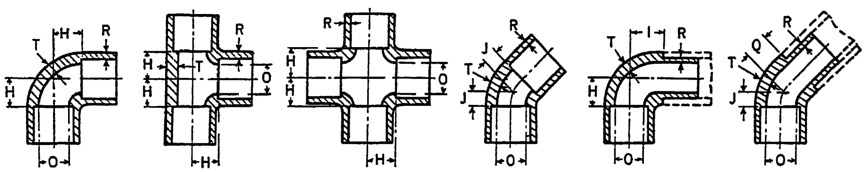
Thread Connections

Threads used for fittings are the same American Standard taper pipe threads as those used for pipe, and unless otherwise ordered, right-hand threads are used. To facilitate drainage, some elbows have the thread tapped at an angle to provide a pitch of the connecting pipe of ¼ in. to the foot. These elbows are known to the trade as pitched elbows and are commercially available. Malleable iron fittings, like brass fittings, are cast with a round instead of a flat band or bead, or with no bead at all. Fittings are designated as male or female, depending on whether the threads are on the outside or inside, respectively.

Flanged fittings are generally used in the best practice for connecting all piping above 4 in. in diameter. While screwed fittings may be used for the larger sizes and are satisfactory under the proper working conditions, it will be found difficult either to make or to break the joints in these large sizes.

A number of different flange facings in common use are plain face, raised face, tongue and groove, and male and female. Cast-iron fittings for 125 lb pressure and below are normally furnished with a plain face, while the 250 lb cast-iron fittings are supplied with a 1/16-in. raised face. The standard facing for steel flanged fittings for 150 and 300 lb is a 1/16-in. raised face although these fittings are obtainable with a variety of facings. The gasket surface of the raised face may be finished smooth

Table 6. American Standard Dimensions of Elbows, Tees, Crosses, and 45 Deg Elbows, Soldered-Joint Fittings, *A.S.A.* A40.3-1941



Nominal Size[a]	Cast Brass[b]							Wrought Metal
	Laying Length, Tee, Ell, and Cross[b]	Laying Length, Ell With External Shoulder	Laying Length, 45 Deg Ell	Laying Length, 45 Deg Ell External Shoulder	Inside Diameter of Fittings,[c] Min.	Metal Thickness[d]		Metal Thickness[e] Min.[f]
	H	I	J	Q	O	T	R	T and R
1/4	1/4	3/8	3/16	1/4	0.31	0.08	0.048	0.030
3/8	5/16	7/16	3/16	5/16	0.43	0.08	0.048	0.035
1/2	7/16	9/16	3/16	5/16	0.54	0.09	0.054	0.040
3/4	9/16	11/16	1/4	3/8	0.78	0.10	0.060	0.045
1	3/4	7/8	5/16	7/16	1.02	0.11	0.066	0.050
1 1/4	7/8	1	7/16	9/16	1.26	0.12	0.072	0.055
1 1/2	1	1 1/8	1/2	5/8	1.50	0.13	0.078	0.060
2	1 1/4	1 3/8	9/16	3/4	1.98	0.15	0.090	0.070
2 1/2	1 1/2	1 5/8	5/8	7/8	2.46	0.17	0.102	0.080
3	1 3/4	1 7/8	3/4	1	2.94	0.19	0.114	0.090
3 1/2	2	2 1/8	7/8	1 1/8	3.42	0.20	0.120	0.100
4	2 1/4	2 3/8	15/16	1 1/4	3.90	0.22	0.132	0.110
5	3 1/8	--------	1 1/16	--------	4.87	0.28	0.168	0.125
6	3 5/8	--------	1 5/8	--------	5.84	0.34	0.204	0.140

All dimensions given in inches.

[a]This size is the nominal bore of the tube.

[b]These dimensions may be used for wrought-metal fittings as well as for cast-brass fittings at manufacturer's option.

[c]This dimension is the same as the inside diameter Class L tubing (American Standard Specifications for Copper Water Tube, *A.S.A.* H23.1-1939 (*A.S.T.M.* B88).

[d]Patterns shall be designed to produce body thicknesses given in the table. Metal thickness at no point shall be less than 90 per cent of the thicknesses given in the table.

[e]This dimension has the same thickness as Type L tubing.

[f]These dimensions are minimum, but in every case the thickness of wrought fittings should be at least as heavy as the tubing with which it is to be used.

Note 1:—Wrought fittings, as well as cast fittings, must be provided with a shoulder or stop at the bottom end of socket.

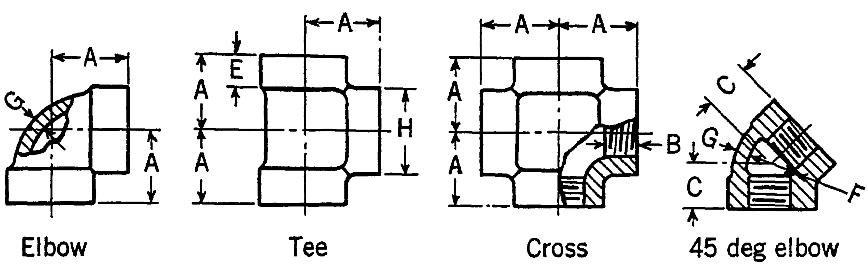
Note 2.—Street fittings with male ends are for use in connection with other fittings illustrated.

or may be machined with concentric or spiral grooves often referred to as serrated face or phonograph finish, respectively.

The dimensions of elbows, tees and crosses for 125 lb cast-iron screwed fittings are given in Table 7, whereas the dimensions for 125 lb cast-iron flanged fittings are given in Tables 8 and 9.

For low temperature service not to exceed about 220 F, a number of paper or vegetable fiber gasket materials will prove satisfactory; for plain raised face flanges, rubber or rubber inserted gaskets are commonly employed. Asbestos composition gaskets are probably the most widely used, particularly where the temperature exceeds 250 F. Jacketed

TABLE 7. AMERICAN STANDARD DIMENSIONS OF ELBOWS, 45-DEG ELBOWS, TEES, AND CROSSES (STRAIGHT SIZES) FOR CLASS 125 CAST-IRON SCREWED FITTINGS, *A.S.A.* B16a-1939



	A	C	B	E	F		G	H
Nominal Pipe Size	Center to End, Elbows, Tees and Crosses	Center to End 45 Deg Elbows	Length of Thread Min	Width of Band, Min	Inside Diameter of Fitting Min.	Inside Diameter of Fitting Max.	Metal Thickness,[a] Min.	Outside Diameter of Band Min
¼	0.81	0.73	0.32	0.38	0.540	0.584	0.110	0.93
⅜	0.95	0.80	0.36	0.44	0.675	0.719	0.120	1.12
½	1.12	0.88	0.43	0.50	0.840	0.897	0.130	1.34
¾	1.31	0.98	0.50	0.56	1.050	1.107	0.155	1.63
1	1.50	1.12	0.58	0.62	1.315	1.385	0.170	1.95
1¼	1.75	1.29	0.67	0.69	1.660	1.730	0.185	2.39
1½	1.94	1.43	0.70	0.75	1.900	1.970	0.200	2.68
2	2.25	1.68	0.75	0.84	2.375	2.445	0.220	3.28
2½	2.70	1.95	0.92	0.94	2.875	2.975	0.240	3.86
3	3.08	2.17	0.98	1.00	3.500	3.600	0.260	4.62
3½	3.42	2.39	1.03	1.06	4.000	4.100	0.280	5.20
4	3.79	2.61	1.08	1.12	4.500	4.600	0.310	5.79
5	4.50	3.05	1.18	1.18	5.563	5.663	0.380	7.05
6	5.13	3.46	1.28	1.28	6.625	6.725	0.430	8.28
8	6.56	4.28	1.47	1.47	8.625	8.725	0.550	10.63
10	8.08[b]	5.16	1.68	1.68	10.750	10.850	0.690	13.12
12	9.50[b]	5.97	1.88	1.88	12.750	12.850	0.800	15.47

All dimensions given in inches.

[a]Patterns shall be designed to produce castings of metal thickness given in the table. Metal thickness at no point shall be less than 90 per cent of the thickness given in the table.

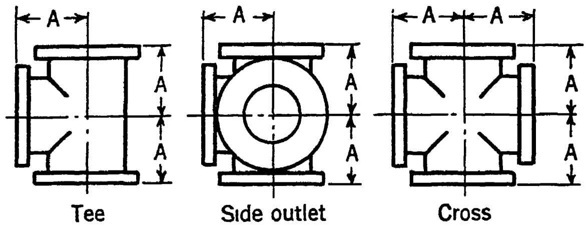
[b]Applies to elbows and tees only.

asbestos and metallic gaskets may be used for any pressure and temperature conditions, but preferably only with a relatively narrow recessed facing.

WELDING

Erection of piping in heating and ventilating installations by means of fusion welding has been commonly accepted in the past few years as a competitive method to the screwed and flanged joint. Since the question

TABLE 8. AMERICAN STANDARD DIMENSIONS OF TEES AND CROSSES[a] (STRAIGHT SIZES) FOR CLASS 125 CAST-IRON FLANGED FITTINGS, *A.S.A.* B16a-1939



Nominal Pipe Size[b-c]	A Center to Face Tees and Crosses[c-d]	AA Face to Face Tees and Crosses[c-d]	Diameter of Flange	Thickness of Flange, Min.	Metal[e] Thickness of Body
1	3 1/2	7	4 1/4	7/16	5/16
1 1/4	3 3/4	7 1/2	4 5/8	1/2	5/16
1 1/2	4	8	5	9/16	5/16
2	4 1/2	9	6	5/8	5/16
2 1/2	5	10	7	11/16	5/16
3	5 1/2	11	7 1/2	3/4	3/8
3 1/2	6	12	8 1/2	13/16	7/16
4	6 1/2	13	9	15/16	1/2
5	7 1/2	15	10	15/16	1/2
6	8	16	11	1	9/16
8	9	18	13 1/2	1 1/8	5/8
10	11	22	16	1 3/16	3/4
12	12	24	19	1 1/4	13/16
14 O.D.	14	28	21	1 3/8	7/8
16 O.D.	15	30	23 1/2	1 7/16	1
18 O.D.	16 1/2	33	25	1 9/16	1 1/16
20 O.D.	18	36	27 1/2	1 11/16	1 1/8
24 O.D.	22	44	32	1 7/8	1 1/4
30 O.D.	25	50	38 3/4	2 1/8	1 7/16
36 O.D.	28	56	46	2 3/8	1 5/8
42 O.D.	31	62	53	2 5/8	1 13/16
48 O.D.	34	68	59 1/2	2 3/4	2

All dimensions given in inches.

[a]Crosses both straight and reducing sizes 18 in. and larger shall be reinforced to compensate for the inherent weakness in the casting design.

[b]Size of all fittings listed indicates nominal inside diameter of port.

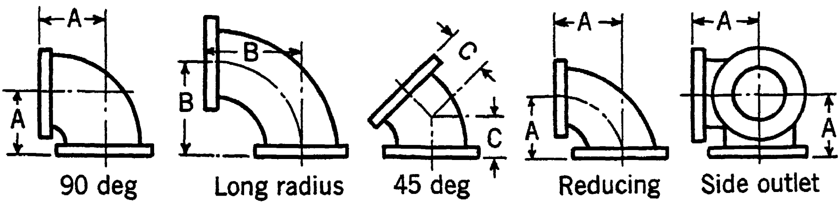
[c]Tees, side outlet tees, and crosses, 16 in. and smaller, reducing on the outlet, have the same dimensions center to face, and face to face as straight size fittings corresponding to the size of the larger opening. Sizes 18 in. and larger, reducing on the outlet, are made in two lengths, depending on the size of the outlet.

[d]Tees and crosses, reducing on run only, carry same dimensions center to face and face to face as a straight size fitting of the larger opening.

[e]Body thickness at no point shall be less than 87 1/2 per cent of the dimensions given in the table.

of economy of welding as against the use of screwed and flanged fittings is dependent on the individual job, the use of welding is generally recommended on the basis of a greatly reduced cost of maintenance and repair, of less weight resulting from the use of a lighter-weight pipe, and of increased economy in pipe insulation, hangers, and supports rather than on the basis of any economy that might be effected in actual erection by welding on low to medium pressure heating jobs.

TABLE 9. AMERICAN STANDARD DIMENSIONS OF ELBOWS FOR CLASS 125 CAST-IRON FLANGED FITTINGS, *A.S.A.* B16a-1939



Nominal Pipe Size [a]	*A* Center to Face Elbow [b-c-d]	*B* Center to Face Long Radius Elbow [b-c-d]	*C* Center to Face 45 Deg Elbow [c]	Diameter of Flange	Thickness of Flange, Min.	Metal [e] Thickness of Body
1	3 1/2	5	1 3/4	4 1/4	7/16	5/16
1 1/4	3 3/4	5 1/2	2	4 5/8	1/2	5/16
1 1/2	4	6	2 1/4	5	9/16	5/16
2	4 1/2	6 1/2	2 1/2	6	5/8	5/16
2 1/2	5	7	3	7	11/16	5/16
3	5 1/2	7 3/4	3	7 1/2	3/4	3/8
3 1/2	6	8 1/2	3 1/2	8 1/2	13/16	7/16
4	6 1/2	9	4	9	15/16	1/2
5	7 1/2	10 1/4	4 1/2	10	15/16	1/2
6	8	11 1/2	5	11	1	9/16
8	9	14	5 1/2	13 1/2	1 1/8	5/8
10	11	16 1/2	6 1/2	16	1 3/16	3/4
12	12	19	7 1/2	19	1 1/4	13/16
14 O.D.	14	21 1/2	7 1/2	21	1 3/8	7/8
16 O.D.	15	24	8	23 1/2	1 7/16	1
18 O.D.	16 1/2	26 1/2	8 1/2	25	1 9/16	1 1/16
20 O.D.	18	29	9 1/2	27 1/2	1 11/16	1 1/8
24 O.D.	22	34	11	32	1 7/8	1 1/4
30 O.D.	25	41 1/2	15	38 3/4	2 1/8	1 7/16
36 O.D.	28	49	18	46	2 3/8	1 5/8
42 O.D.	31	56 1/2	21	53	2 5/8	1 13/16
48 O.D.	34	64	24	59 1/2	2 3/4	2

All dimensions given in inches.

[a] Size of all fittings listed indicates nominal inside diameter of port.

[b] Reducing elbows and side outlet elbows carry same dimensions center to face as straight size elbows corresponding to the size of the larger opening.

[c] Special degree elbows, ranging from 1 to 45 deg, inclusive, have the same center to face dimensions as given for 45-deg elbows and those over 45 deg and up to 90 deg, inclusive, shall have the same center to face dimensions as given for 90-deg elbows. The angle designation of an elbow is its deflection from straight line flow and is the angle between the flange faces.

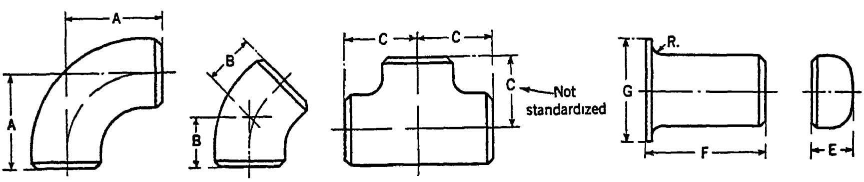
[d] Side outlet elbows shall have all openings on intersecting center-lines.

[e] Body thickness at no point shall be less than 87 1/2 per cent of the dimensions given in the table.

Fusion welding, commonly used in erection of piping, is defined as the process of joining metal parts in the molten, or molten and vapor states, without the application of mechanical pressure or blows. Fusion welding embraces gas welding and electric arc welding, both of which are commonly used to produce acceptable welds.

Welding application requires the same basic knowledge of design as do the other types of assembly, but in addition, requires a generous knowledge of the sciences involved, particularly as to welding qualities of

Table 10. American Standard Dimensions for Butt-Welding Elbows, Tees, Caps, and Lapped-Joint Stub Ends, *A.S.A.* B16.9-1940



Nominal Pipe Size	Outside Diameter	Center-to-End: 90-Deg Elbows *A*	Center-to-End: 45-Deg Elbows *B*	Center-to-End: Of Run Tee *C*[a]	Caps *E*[b-c]	Lapped-Joint Stub Ends: Length *F*[b]	Lapped-Joint Stub Ends: Radius of Fillet *R*	Lapped-Joint Stub Ends: Diam. of Lap *G*[d]
1	1.315	1½	⅞	1½	1½	4	⅛	2
1¼	1.660	1⅞	1	1⅞	1½	4	3/16	2½
1½	1.900	2¼	1⅛	2¼	1½	4	¼	2⅞
2	2.375	3	1⅜	2½	1½	6	5/16	3⅝
2½	2.875	3¾	1¾	3	1½	6	5/16	4⅛
3	3.500	4½	2	3⅜	2	6	⅜	5
3½	4.000	5¼	2¼	3¾	2½	6	⅜	5½
4	4.500	6	2½	4⅛	2½	6	7/16	6 3/16
5	5.563	7½	3⅛	4⅞	3	8	7/16	7 5/16
6	6.625	9	3¾	5⅝	3½	8	½	8½
8	8.625	12	5	7	4	8	½	10⅝
10	10.750	15	6¼	8½	5	10	½	12¾
12	12.750	18	7½	10	6	10	½	15

All dimensions given in inches.

[a]The dimensions of welding tees cover those which have side outlets from one size less than half the size of the run-way opening of the tees to full size.

[b]Dimensions *E* and *F* are applicable only to these fittings in schedules up to and including Schedule 80, *A.S.A.* Standard B36.10-1939.

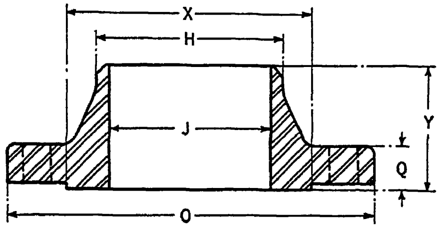
[c]The shape of these caps shall be ellipsoidal and shall conform to the requirements of the *A.S.M.E.* Boiler Construction Code.

[d]This dimension is for standard machined facings in accordance with American Standard for Steel Pipe Flanges and Flanged Fittings (*A.S.A.* B16e-1939). The back face of the lap shall be machined to conform to the surface of the flange on which it seats. Where ring joint facings are to be applied, use dimension *K* as given in *A.S.A.* B16e-1939.

metal, their reaction to extremely high temperatures, and the ability to determine and use only the best quality welding rods. This requirement applies equally to employer and employee with the employer accepting all of the responsibility. Thus the employer should select his welding mechanics with good judgment, provide them with first-class equipment and tools, arrange for their training and use of acceptable workmanship standards, and at regular intervals subject their work to prescribed tests. Industry will not accept the employment of mechanics of undetermined

ability nor on the basis of past experience. Neither does industry accept the statement that a weld is only as good as the workman who makes it. The control Codes now in process of adoption will be the law governing the use of the welding process. These Codes prohibit individual practices contrary to their specified procedure and rules of control, and this is

TABLE 11. AMERICAN STANDARD DIMENSIONS OF STEEL WELDING NECK FLANGES FOR STEAM SERVICE PRESSURE RATING OF 150 LB PER SQ IN. (GAGE) AT A TEMPERATURE OF 500 F, AND 100 LB. PER SQ. IN. (GAGE) AT 750 F, *A.S.A.* B16e-1939



Nominal Pipe Size	Diameter of Flange	Thickness of Flg.[a] Min.	Diameter of Hub	Hub Diam. Beginning of Chamfer[b-c]	Length Thru Hub[a]	Inside Diam. of Pipe Schedule 40[c]	Diam. of Bolt Circle	No. of Bolts	Size of Bolts
	O	*Q*	*X*	*H*	*Y*	*J*			
1/2	3 1/2	7/16	1 3/16	0.84	1 7/8	0.62*	2 3/8	4	1/2
3/4	3 7/8	1/2	1 1/2	1.05	2 1/16	0.82*	2 3/4	4	1/2
1	4 1/4	9/16	1 15/16	1.32	2 3/16	1.05*	3 1/8	4	1/2
1 1/4	4 5/8	5/8	2 5/16	1.66	2 1/4	1.38*	3 1/2	4	1/2
1 1/2	5	11/16	2 9/16	1.90	2 7/16	1.61*	3 7/8	4	1/2
2	6	3/4	3 1/16	2.38	2 1/2	2.07*	4 3/4	4	5/8
2 1/2	7	7/8	3 9/16	2.88	2 3/4	2.47*	5 1/2	4	5/8
3	7 1/2	15/16	4 1/4	3.50	2 3/4	3.07*	6	4	5/8
3 1/2	8 1/2	15/16	4 13/16	4.00	2 13/16	3.55*	7	8	5/8
4	9	15/16	5 5/16	4.50	3	4.03*	7 1/2	8	5/8
5	10	15/16	6 7/16	5.56	3 1/2	5.05*	8 1/2	8	3/4
6	11	1	7 9/16	6.63	3 1/2	6.07*	9 1/2	8	3/4
8	13 1/2	1 1/8	9 11/16	8.63	4	7.98*	11 3/4	8	3/4
10	16	1 3/16	12	10.75	4	10.02*	14 1/4	12	7/8
12	19	1 1/4	14 3/8	12.75	4 1/2	------------	17	12	7/8
14 O.D.	21	1 3/8	15 3/4	14.00	5	To Be	18 3/4	12	1
16 O.D.	23 1/2	1 7/16	18	16.00	5	Specified	21 1/4	16	1
18 O.D.	25	1 9/16	19 7/8	18.00	5 1/2	by	22 3/4	16	1 1/8
20 O.D.	27 1/2	1 11/16	22	20.00	5 11/16	Purchaser	25	20	1 1/8
24 O.D.	32	1 7/8	26 1/8	24.00	6		29 1/2	20	1 1/4

All dimensions given in inches.

[a] A raised face of 1/16 in. is included in *thickness of flange minimum* and in *length through hub.*

[b] The outside surface of the welding end of the hub shall be straight or tapered at not more than 6 deg.

[c] Dimensions *H* and *J* correspond to the outside and inside diameters of pipe as given in *A.S.A.* B36.10-1939, Schedule 40.

*These diameters are identical with the diameters of what was formerly designated as *Standard Weight Pipe* of the corresponding sizes.

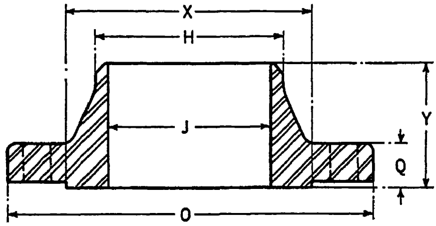
predicated upon the sound requirement that the employer must assume full responsibility for the deposited weld.

It is advisable that this management responsibility be included in all welding specifications and that authoritative standards of workmanship also be specified. The standards of workmanship for this industry are as

set forth in the *Standard Manual on Pipe Welding* of the *Heating, Piping and Air Conditioning Contractors National Association.*

A complete line of manufactured steel welding fittings is now available

TABLE 12. AMERICAN STANDARD DIMENSIONS OF STEEL WELDING NECK FLANGES FOR STEAM SERVICE PRESSURE RATING OF 300 LB PER SQ IN. (GAGE) AT A TEMPERATURE OF 750 F, *A.S.A.* B16e-1939



NOMINAL PIPE SIZE	DIAM. OF FLANGE	THICKNESS OF FLG.[a] MIN.	DIAM. OF HUB	HUB DIAM. BEGINNING OF CHAMFER[b-c-d]	LENGTH THRU HUB[a]	INSIDE DIAM. OF PIPE SCHEDULE 40[c d]	INSIDE DIAM. OF PIPE SCHEDULE 80[c-d]	DIAM. OF BOLT CIRCLE	NO. OF BOLTS	SIZE OF BOLTS
	O	*Q*	*X*	*H*	*Y*	*J*	*J*			
½	3¾	9/16	1½	0.84	2 1/16	0.62*	0.55†	2⅝	4	½
¾	4⅝	⅝	1⅞	1.05	2¼	0.82*	0.74†	3¼	4	⅝
1	4⅞	11/16	2⅛	1.32	2 7/16	1.05*	0.96†	3½	4	⅝
1¼	5¼	¾	2½	1.66	2 9/16	1.38*	1.28†	3⅞	4	⅝
1½	6⅛	13/16	2¾	1.90	2 11/16	1.61*	1.50†	4½	4	¾
2	6½	⅞	3 5/16	2.38	2¾	2.07*	1.94†	5	8	⅝
2½	7½	1	3 15/16	2.88	3	2.47*	2.32†	5⅞	8	¾
3	8¼	1⅛	4⅝	3.50	3⅛	3.07*	2.90†	6⅝	8	¾
3½	9	1 3/16	5¼	4.00	3 3/16	3.55*	3.36†	7¼	8	¾
4	10	1¼	5¾	4.50	3⅜	4.03*	3.83†	7⅞	8	¾
5	11	1⅜	7	5.56	3⅞	5.05*	4.81†	9¼	8	¾
6	12½	1 7/16	8⅛	6.63	3⅞	6.07*	5.76†	10⅝	12	¾
8	15	1⅝	10¼	8.63	4⅜	7.98*	7.63†	13	12	⅞
10	17½	1⅞	12⅝	10.75	4⅝	10.02*	--------	15¼	16	1
12	20½	2	14¾	12.75	5⅛	----------	--------	17¾	16	1⅛
14 O.D.	23	2⅛	16¾	14.00	5⅝	To Be	To Be	20¼	20	1⅛
16 O.D.	25½	2¼	19	16.00	5¾	Speci-	Speci-	22½	20	1¼
18 O.D.	28	2⅜	21	18.00	6¼	fied by	fied by	24¾	24	1¼
20 O.D.	30½	2½	23⅛	20.00	6⅜	Pur-	Pur-	27	24	1¼
24 O.D.	36	2¾	27⅝	24.00	6⅝	chaser	chaser	32	24	1½

All dimensions given in inches.

[a]A raised face of 1/16 in. is included in *thickness of flange minimum* and in *length through hub.*

[b]This outside surface of the welding end of the hub shall be straight or tapered at not more than 6 deg.

[c]Dimensions *H* and *J* correspond to the outside and inside diameters of pipe as given in *A.S.A.* B36.10-1939, Schedules 40 and 80. Purchaser's order must specify which of these two inside diameters is desired.

[d]These flanges are regularly bored to match inside diameter of Schedule 40 pipe, but are bored to Schedule 80 pipe when so ordered.

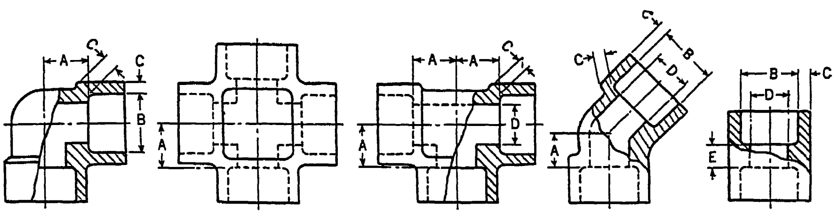
*These diameters are identical with the diameters of what was formerly designated as *Standard-Weight Pipe* of the corresponding sizes.

†These diameters are identical with the diameters of what was formerly designated as *Extra-Strong Pipe* of the corresponding sizes.

and a dimensional standard has been prepared under the procedure of the *American Standards Association* to unify heretofore divergent dimensions for the same type welding fittings as produced by different manufacturers. Standard dimensions for elbows, tees, caps, and lapped-joint stub ends

are given in Table 10. Dimensions for eccentric and concentric reducers, and 180-deg return bends are not shown in Table 10 but are included in the American Standard. Larger sizes also are available in some types of fittings. The welding bevel which is a straight 37½-deg V for wall thicknesses ¾ in. and below, and a U-bevel for thicknesses heavier than ¾ in., conforms to the recommended practice of *A.S.A.* Standard B16e-1939, American Standard for Steel Pipe Flanges and Flanged Fittings. The latter also contains dimensions for steel welding-neck flanges for pressures

TABLE 13. PROPOSED AMERICAN STANDARD DIMENSIONS OF SOCKET-WELDING ELBOWS, TEES, CROSSES, 45-DEG ELBOWS, AND COUPLINGS



Nominal Pipe Size	Minimum Depth of Socket	Center to Bottom of Socket: 90-Deg Ells, Tees, Crosses, Sched. 40 & 80	Center to Bottom of Socket: 90-Deg Ells, Tees, Crosses, Sched. 160	Center to Bottom of Socket: 45-Deg Ells, Sched. 40 & 80	Center to Bottom of Socket: 45-Deg Ells, Sched. 160	Couplings Distance Between Bottoms Sockets	Bore Diameter of Socket, Minimum	Minimum Socket Wall Thickness: Sched. 40	Minimum Socket Wall Thickness: Sched. 80	Minimum Socket Wall Thickness: Sched. 160	Bore Diameter of Fittings: Sched. 40	Bore Diameter of Fittings: Sched. 80	Bore Diameter of Fittings: Sched. 160
		A		A		E	B	Ca			D		
¼	⅜	17/32		5/16	.. ...	¼b	0.555b	0.156	0.156	..	0.364	0.302	...
⅜	⅜	17/32	....	5/16	.	¼b	0.690b	0.156	0.158		0.493	0.423	. .
½	⅜	⅝	¾	7/16	½	⅜	0.855	0.156	0.184	0.234	0.622	0.546	0.466
¾	½	¾	⅞	½	9/16	⅜	1.065	0 156	0.193	0.273	0.824	0.742	0.614
1	½	⅞	1 1/16	9/16	11/16	½	1.330	0.166	0.224	0 313	1.049	0.957	0.815
1¼	½	1 1/16	1¼	11/16	13/16	½	1.675	0.175	0.239	0.313	1.380	1.278	1.160
1½	½	1¼	1½	13/16	1	½	1.915	0.181	0.250	0.351	1.610	1.500	1 338
2	⅝	1½	1⅝	1	1⅛	¾	2.406	0.193	0.273	0.429	2.067	1.939	1.689
2½	⅝	1⅝	2¼	1⅛	1¼	¾	2.906	0.254	0.345	0.469	2.469	2.323	2.125
3	⅝	2¼	2½	1¼	1⅜	¾	3.535	0.270	0.375	0.546	3.068	2.900	2.626

All dimensions are given in inches.

[a]Dimensions *C* is 1¼ times the nominal pipe thickness, minimum, but not less than 5/32 in.

[b]This dimension applies to Schedules 40 and 80 only.

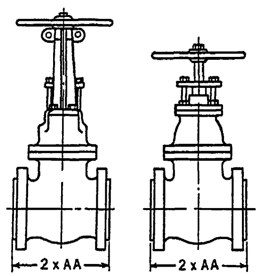
Reducing sizes have same center to bottom of socket dimension as the largest size of reducing fitting

up to 2500 lb per square inch. Tables 11 and 12 give these dimensions for welding-neck flanges suitable for 150 and 300 lb per square inch gage pressure.

Socket welding fittings are also commercially available. These fittings have a machined recess into which the pipe slips. A fillet weld between the pipe and socket edge provides a pressure-tight joint. This type of fitting has gained rapid acceptance due to its ease of installation, low cost, and ability to make a pressure tight joint without weakening the pipe as is the case with threading. Standard dimensions for socket welding fittings are being formulated under the procedure of the *American Standards Association.*

Use of socket welding fittings generally is restricted to nominal pipe size 3 in. and smaller in which range commercial fittings are available. For pipe sizes above 3 in., butt welding fittings of the type shown in Table 10 customarily are used. Proposed American Standard dimensions for socket-welding elbows, tees, crosses, 45-deg elbows, and couplings, which are being formulated under the procedure of the *American Standards Association*, are given in Table 13.

TABLE 14. AMERICAN STANDARD CONTACT SURFACE TO CONTACT SURFACE DIMENSIONS OF CAST-IRON AND STEEL FLANGED WEDGE GATE VALVES, *A.S.A.* B16.10-1939



Nominal Pipe Size	Contact Surface to Contact Surface Dimensions, (2 × AA)				
	Cast-Iron[a]			Steel	
	125	175[bc]	250[b]	150[b]	300[b]
1	....	....	....	....	....
1¼	....	....	....	....	....
1½	....	....	....	....	7½
2	7	7¼	8½	7	8½
2½	7½	8	9½	7½	9½
3	8	9¼	11⅛	8	11⅛
3½	8½	10	11⅞	8½	11⅞
4	9	10½	12	9	12
5	10	11½	15	10	15
6	10½	13	15⅞	10½	15⅞
8	11½	14¼	16½	11½	16½
10	13	16¾	18	13	18
12	14	17½	19¾	14	19¾
14 O.D.	15	....	22½	15	30
16 O.D.	16	....	24	16	33
18 O.D.	17	....	26	17	36
20 O.D.	18	....	28	18	39
24 O.D.	20	....	31	20	45

All dimensions given in inches.

[a]These dimensions are the same for Cast-Iron Double Disc Flanged Gate Valves.

[b]These are pressure designations which refer to the primary service ratings in pounds per square inch of the connecting end flanges.

[c]The connecting end flanges of 175 lb valves are the same as those on 250 lb valves.

NOTE 1:—Where dimensions are not given, the sizes either are not made or there is insufficient demand to warrant the expense of unification.

NOTE 2:—Female and groove joint facings have bottom of groove in same plane as *flange edge*, and center to contact surface dimensions for these facings are reduced by the amount of the raised face.

VALVES

Valves are made with both threaded and flanged ends for screwed and bolted connections just as are pipe fittings.

The material used for valves of small size is generally brass or bronze for low pressures and forged steel for high pressures, while in the larger sizes either cast-iron, cast-steel or some of the steel alloys are employed. Practically all iron or steel valves intended for steam or water work are bronze-mounted or trimmed.

Brass, bronze, and iron valves are generally designed for standard or extra heavy service, the former being used up to 125 lb and the latter up to 250 lb saturated steam working pressure, although most manufacturers also make valves for medium pressure up to 175 lb steam working pressure. The more common types are gate valves or straightway valves, globe valves, angle valves, check valves and automatic valves, such as reducing and back-pressure valves.

Gate valves are the most frequently used of all valves since in their open position the resistance to flow is a minimum, but they should not be used where it is desired to throttle the flow; globe valves should be used for this purpose. These valves may be secured with either a rising or a non-rising stem, although in the smaller sizes the rising stem is more commonly used. The rising stem valve is desirable because the positions of the handle and stem indicate whether the valve is open or closed, although space limitations may prevent its use. The globe valve is less expensive to manufacture than the gate valve, but its peculiar construction offers a high resistance to flow and may prevent complete drainage of the pipe line. These objections are of particular importance in heating work.

An American Standard, *A.S.A.* B16.10-1939, has been prepared giving the face-to-face dimensions of ferrous flanged and welding end valves. The following types are covered: wedge gate, double disc gate, globe and angle, and swing check. One purpose of establishing these dimensions is to insure that gate valves of a given rating and flange dimension of either the wedge or double disc design will be interchangeable in a pipe line. Contact surface to contact surface dimensions of cast-iron and steel flanged wedge gate valves are given in Table 14. End-to-end dimensions for steel butt-welding valves in sizes up to 8 in., inclusive, are the same as those given in Table 14 for steel valves.

Check valves are automatic in operation and permit flow in only one direction, depending for operation on the difference in pressure between the two sides of the valve. The two principal kinds of check valves are the swing check in which a flapper is hinged to swing back and forth, and the lift check in which a dead weight disc moves vertically from its seat.

Valves commonly used for controlling steam or water supply to radiators constitute a special class since they are manufactured to meet heating system requirements. These valves are generally of the angle type and are usually made of brass. Graduations on the heads or lever handles are often supplied to indicate the relative opening of the valve in any position.

Automatic control of steam supply to individual radiators can be effected by use of direct-acting radiator valves having a thermostatic

element at the valve, or near to it. The direct-acting valve is usually an angle-type valve containing a thermostatic element which permits the flow of steam in accordance with room temperature requirements. These valves usually are capable of adjustment to permit variation in room temperature to suit individual taste.

Ordinary steam valves may be used for hot water service by drilling a 1/16-in. hole through the web forming the seat to insure sufficient circulation to prevent freezing when the valve is closed. Valves made particularly for use in hot water heating systems are of less complex design, one type consisting of a simple butterfly valve, and another of a quick opening type in which a part in the valve mechanism matches up with an opening in the valve body.

In one-pipe steam-heating systems, automatic air valves are required at the radiators. Two common types of air valves available are the vacuum type and the straight-pressure type. Vacuum valves permit the expulsion of air from the radiators when the steam pressure rises and, in addition, act as checks to prevent the return of air into the radiator when a vacuum is formed by the condensation of steam after the supply pressure has dropped. Ordinary air valves permit the expulsion of air from the radiator when steam is supplied under pressure, but when the pressure dies down and a vacuum tends to be formed the air is drawn back into the radiator.

A system operating either continuously or intermittently and supplied with vacuum valves will generally hold heat longer and warm up more quickly than one provided with non-vacuum air valves; thus, it will effect considerable economy of fuel because the idle period during which no heat is delivered is shortened. In those cases, where a system is equipped with vacuum air valves and which has been cold for several hours, the system will probably have an internal pressure within the radiator closely approaching atmospheric. At such times, the vacuum valve will not vent the system any more rapidly than the ordinary type. Automatic air valves are provided with a float to close them in case the radiator becomes flooded with water because it does not drain properly.

CORROSION[2]

Corrosion is sometimes encountered in heating work on the outside of buried pipes or the inside of steam heating systems; it is seldom experienced in hot water heating systems unless the water is frequently renewed. Piping buried in the ground is quite successfully protected by coatings of the asphaltic type which are usually applied hot and often reinforced with fabric wrappings. Galvanizing by the hot-dip process and painting with specially prepared mixtures also afford some protection.

Internal corrosion in steam heating systems occurs principally in the

[2]New Light on Heating System Corrosion, by J. H. Walker (*Heating and Ventilating*, May, 1933). A.S.H.V.E. Research Report No. 983—Corrosion Studies in Steam Heating Systems, by R. R. Seeber, F. A. Rohrman and G. E. Smedberg, (A.S.H.V.E. Transactions, Vol. 40, 1934, p. 253). A.S.H.V.E. Research Report No. 1037—Corrosion Studies in Steam Heating Systems, by R. R. Seeber, F. A. Rohrman and G. E. Smedberg, (A.S.H.V.E. Transactions, Vol. 42, 1936, p. 263). A.S.H.V.E. Research Report No. 1071—Corrosion Studies in Steam Heating Systems, by R. R. Seeber and Margaret R. Holley (A.S.H.V.E. Transactions, Vol. 43, 1937, p. 461). Corrosion in Steam Heating Systems, by L. F. Collins and E. L. Henderson, (*Heating, Piping and Air Conditioning*, September, 1939 to May, 1940).

condensate return pipes and is nearly always caused by oxygen or carbon dioxide, or both, in solution in the condensate. Oxygen may enter the heating system with the steam, owing to its presence in the boiler-feed water, or it may enter as air through small leaks, particularly in systems which operate at sub-atmospheric pressures. When a steam heating system is operated intermittently, air rushes in during each shutdown period and oxygen is absorbed by the condensate which clings to the interior surfaces of the pipes and radiators. The rate of corrosion depends upon the amounts of oxygen and carbon dioxide present in solution, upon the operating temperature, and upon the length of time that the pipe surfaces are in contact with gas-laden condensate.

Another possible cause of corrosion is a flow of electric current sometimes resulting from faulty electrical circuits which should be corrected. Electrolytic corrosion also may occur because of the presence of two dissimilar metals, such as brass and iron, but the condensate in practically all steam heating systems is such a weak electrolyte that this cause of corrosion is very infrequent.

If trouble is experienced from corrosion, oxygen should be eliminated from the feed water by proper deaeration with commercial apparatus. The elimination of the oxygen due to air leakage is more difficult because of the multitude of small leaks which exist around valve stems and in pipe joints. In vacuum systems, however, an attempt should be made to minimize such leakage.

Carbon dioxide in varying amounts is contained in steam produced from the majority of water supplies. It is formed from the breaking down of carbonates and bicarbonates which are present in nearly all natural waters. It can be partly removed by chemical treatment and deaeration, but there is no simple method whereby it can be entirely eliminated.

These gases cause corrosion only when in solution in the condensate; when they are mixed with dry steam their corrosive effect is negligible. The amount of gas in solution depends upon the partial pressure of that gas in the atmosphere above the surface of the solution, in accordance with the well known physical law of Henry and Dalton[3]. The correct application of this law, however, requires equilibrium conditions which do not always exist under the flow conditions prevailing in a heating system.

There is a distinction between corrosion in heating systems proper and in the condensate discharge lines from other apparatus using steam at relatively high rates, particularly at the times of the cycle when the steam consumption is at its heaviest. In such equipment the gases tend to accumulate in the steam space and to become dissolved in the condensate in appreciable concentrations, thus greatly increasing the possibilities of corrosion. The condensate will more nearly approach in composition the composition of the steam than will the normal condensate from low rating apparatus such as room radiators, and will, therefore, normally include in solution more contaminants. It is possible that careful venting of such equipment would reduce the amount of contaminants dissolved in the condensate, thus giving less corrosion. There is evidence that the partial pressures of the gases are much lower in heating systems

[3]Some Fundamental Considerations of Corrosion in Steam and Condensate Lines, by R. E. Hall and A. R. Mumford (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 121).

than in high usage equipment, and therefore the corrosion possibilities of the two are not comparable. Hence, corrosion observed in the condensate discharge lines from high usage equipment does not necessarily indicate that equally serious corrosion is taking place in the heating system.

The seriousness of corrosive conditions is best determined by actual measurement rather than by inference from isolated instances of pipe failures. The *National District Heating Association* has perfected a corrosion tester for measuring the inherent corrosiveness of existing conditions. This corrosion tester consists of a frame supporting three coils of wire which are carefully weighed. After the tester has been inserted in the pipe line for a definite length of time, the loss of weight of the coils, referred to an established scale, indicates the relative corrosivensss of the condensate. Accompanying such corrosion measurements, a careful chemical analysis should be made of the condensate, and the findings will serve as a basis for an intelligent study of the problem.

There are some indications that after a condensate containing carbon dioxide has dissolved some iron and thereby raised its *pH* value, its corrosive action is greatly reduced and the solution will remain comparatively inactive until admission of oxygen permits the precipitation of the dissolved iron as ferric oxide. The *pH* value of the condensate may be buffered to a fairly high value by the solution of iron and not correspond to the *pH* value to be expected in the unbuffered solution containing the same amount of carbon dioxide.

Corrosion, if found to exist, can be lessened or overcome by several means. If the steam supply is found to be definitely contaminated, proper chemical treatment of the water, followed by deaeration, is an obvious remedy. The leaks in the piping system, particularly in vacuum systems, should be stopped so far as is practicable.

Although inhibitors of various types have had considerable trial and experimentation and successes have been reported, the information as yet requires considerable study to be made satisfactorily useful. Among these inhibitors are oil, sodium silicate, sodium hydroxide, tannin, and various other organic compounds, some of which release ammonia gas. The best guidance to date in the use of such inhibitors is to compare the results found over a period of years in a similar installation operating under the same conditions.

In view of the fact that corrosion is most frequently found in the return lines from special equipment, which constitute a relatively small part of the total piping in a building, a simple solution of the corrosion problem may be to use non-corroding materials in those certain portions of the piping system, since the higher cost will usually be an unappreciable portion of the total. Brass and copper are undoubtedly less subject to this type of corrosion than the ferrous metals, and considerable attention is now being given to corrosion-resistant linings for ferrous pipe. Cast-iron pipe. sometimes alloyed with other metals, also deserves consideration.

Chapter 19

GRAVITY WARM AIR FURNACE SYSTEMS

Design Procedure, Estimating Heating Requirements, Leader Pipe Sizes, Proportioning Wall Stacks, Register Selections, Recirculating Ducts and Grilles, Furnace Return Connection, Furnace Capacity, Examples, Booster Fans

WARM air heating systems of the gravity type are described in this chapter[1], and those of the mechanical type are described in Chapter 20. In the gravity type, the motive head producing flow depends upon the difference in weight between the heated air leaving the top of the casing and the cooled air entering the bottom of the casing, while in the mechanical type a fan may supply all or part of the motive head. Booster fans are often used in conjunction with gravity-designed systems to increase air circulation.

In general, a warm-air furnace heating plant consists of a fuel-burning furnace or heater, enclosed in a casing of sheet metal or brick, which is placed in the basement of the building. The heated air, taken from the top or sides near the top of the furnace casing, is distributed to the various rooms of the building through sheet metal warm-air pipes. The warm-air pipes in the basement are known as leaders, and the vertical warm-air pipes which are run in the inside partitions of the building are called stacks. The heated air is finally discharged into the rooms through registers which are set in register boxes placed either in the floor or in the side wall, usually at or near the baseboard.

The air supply to the furnace may be taken (1) entirely from inside the building through one or more recirculating ducts, (2) entirely from outside the building, in which case no air is recirculated, or (3) through a combination of the inside and the outside air supply systems.

DESIGN PROCEDURE

The design of a furnace heating system involves the determination of the following items:

1. Heat loss in Btu from each room in the building.
2. Area and diameter in inches of warm-air pipes in basement (known as leaders).
3. Area and dimensions in inches of vertical pipes (known as wall stacks).
4. Free and gross area and dimensions in inches of warm-air registers.
5. Area and dimensions of recirculating or outside air ducts, in inches.
6. Free and gross area and dimensions in inches of recirculating registers.

[1]All figures and much of the engineering data which follow are from University of Illinois, *Engineering Experiment Station Bulletins* Nos. 141, 188, 189 and 246; Warm Air Furnaces and Heating Systems, by A. C. Willard, A. P. Kratz, V. S. Day, and S. Konzo.

7. Size of furnace necessary to supply the warm air required to overcome the heat loss from the building. This *size* should include square inches of leader pipe area which the furnace must supply. It is also desirable to call for a minimum bottom fire-pot diameter in inches, which is the nominal grate diameter.

8. Area and dimensions in inches of chimney and smoke pipe. If an unlined chimney is to be used, that fact should be made clear.

The heat loss calculations should be made in accordance with the procedure outlined in Chapter 6, taking into consideration the transmission losses as well as the infiltration losses.

LEADER PIPE SIZES

In a gravity circulating warm-air furnace system the size of the leader to a given room depends upon the temperature of the warm air entering the room at the register. A reasonable air temperature at the registers must, therefore, be chosen before the system can be designed. The *National Warm Air Heating and Air Conditioning Association* has approved an air temperature of 175 F at the registers as satisfactory for design purposes. At this temperature, the heat-carrying capacity (heat available above 70 F) per square inch of leader pipe per hour for first, second or third floors is shown by Fig. 1 at 175 F to be 105, 170 and 208 Btu, respectively. For average calculations, the values 111, 166 and 200 will simplify the work and may be satisfactorily substituted for these heat-carrying capacities. If H represents the total heat to be supplied any room, the resulting equations are:

$$\text{Leader areas for first floor, square inches} = \frac{H}{111} = \text{approximately } 0.009H \quad (1)$$

$$\text{Leader areas for second floor, square inches} = \frac{H}{166} = \text{approximately } 0.006H \quad (2)$$

$$\text{Leader areas for third floor, square inches} = \frac{H}{200} = \text{approximately } 0.005H \quad (3)$$

In designing for a lower warm-air register temperature, say 160 F, the factors 111, 166 and 200 become 80, 140 and 166 (Fig. 1 at 160 F), and the resulting equations are:

$$\text{Leader areas for first floor, square inches} = \frac{H}{80} = \text{approximately } 0.012H \quad (4)$$

$$\text{Leader areas for second floor, square inches} = \frac{H}{140} = \text{approximately } 0.007H \quad (5)$$

$$\text{Leader areas for third floor, square inches} = \frac{H}{166} = \text{approximately } 0.006H \quad (6)$$

These equations are applicable to straight leaders from 6 to 8 ft in length. Longer leaders must be thoroughly insulated or the vertical stacks must be increased in area as discussed under wall stacks. If some provision is not made for these longer leaders, the air temperature may be much lower than anticipated and the room will not be properly heated.

The values shown in Fig. 1 apply only to the case where the straight, leader pipe is 8 ft in length and is connected to stacks whose cross-sectional area is approximately 75 per cent of that of the leader pipe.

Any deviation from these conditions requires a modification of the constants used in Equations 1, 2, and 3. The temperature drop in leaders of various lengths at three different register temperatures is shown in Fig. 2, and should be used to obtain new register temperatures, lower than 175 F, on which to base selections from the curves of Fig. 1, and thereby new constants for Equations 1, 2 and 3.

Leader sizes should in general be not less than those obtained by Equations 1 to 3 nor should leaders less than 8 in. in diameter be used. In residences requiring a leader pipe area of 650 sq in. or less, it is advisable

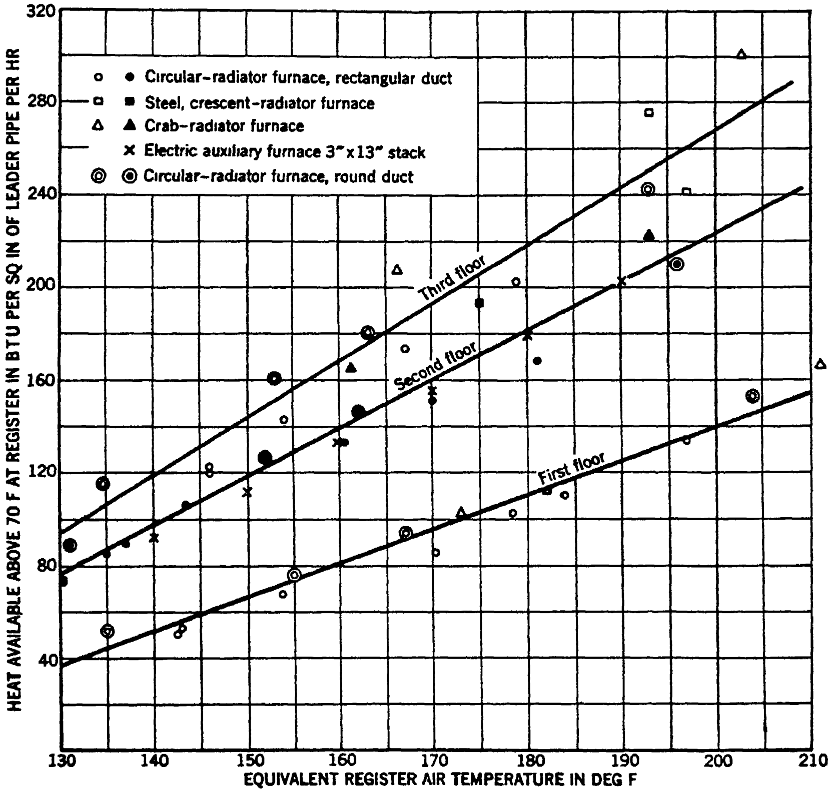


FIG. 1. VALUE OF SQUARE INCH OF LEADER PIPE AREA FOR FIRST, SECOND, AND THIRD FLOORS FOR SIMPLE SYSTEM HAVING LEADERS 8 FT IN LENGTH

to use two or more leader pipes to rooms requiring more than the capacity of a 12 in. round pipe. It is not considered good commercial practice to specify diameters except in whole inches. The tops of all leaders should be at the same elevation as they leave the furnace bonnet, and from this point there should be a uniform up-grade of 1 in. per foot of run in all cases. Leaders over 12 ft in length should be avoided if possible. In cases where such leaders are required, the use of a larger size pipe, than is required by the application of the equations, smooth transition fittings, and duct insulation are recommended.

PROPORTIONING WALL STACKS

The wall stack for an upper floor should be made not less than 70 per cent of the area of the leader. In cases where the leader is short and straight as was the case for Fig. 1, such a practice is probably justified, since the loss (Fig. 3) in capacity occasioned by the smaller stack is not serious for stacks having areas in excess of 70 per cent of the leader area. For leaders over 8 ft in length or for leaders which are not straight, the ratio of stack area to leader area should be greater than 70 per cent in order to offset the greater temperature losses (Fig. 2) in the longer leader. In gravity circulating systems, this ratio of stack to leader area is a very important matter.

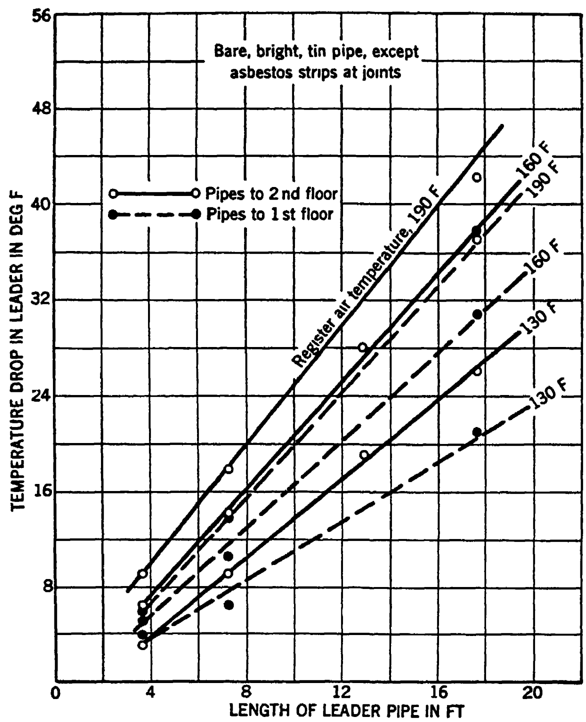


FIG. 2. INFLUENCE OF LEADER PIPE LENGTH ON TEMPERATURE LOSS IN AIR FLOWING THROUGH PIPE

The curves in Figs. 4 and 5 indicate that for rooms having a heat requirement exceeding approximately 9000 Btu per hour, exceedingly high register temperatures are required for stacks whose width is less than 3½ in. For such requirements either multiple stacks, or stacks having larger cross-sectional area (placed in 6 in. studding spaces) will be required.

REGISTER SELECTIONS

The registers used for discharging warm air into the rooms should have a free or net area not less than the area of the leader in the same run of piping. The free area should be at least 70 per cent of the gross area of the register. No upper floor register should be wider horizontally than the wall stack, and it should be placed either in the baseboard or side wall, if this can be done without the use of offsets. First floor registers may be of the baseboard or floor type, with the former location preferred. High

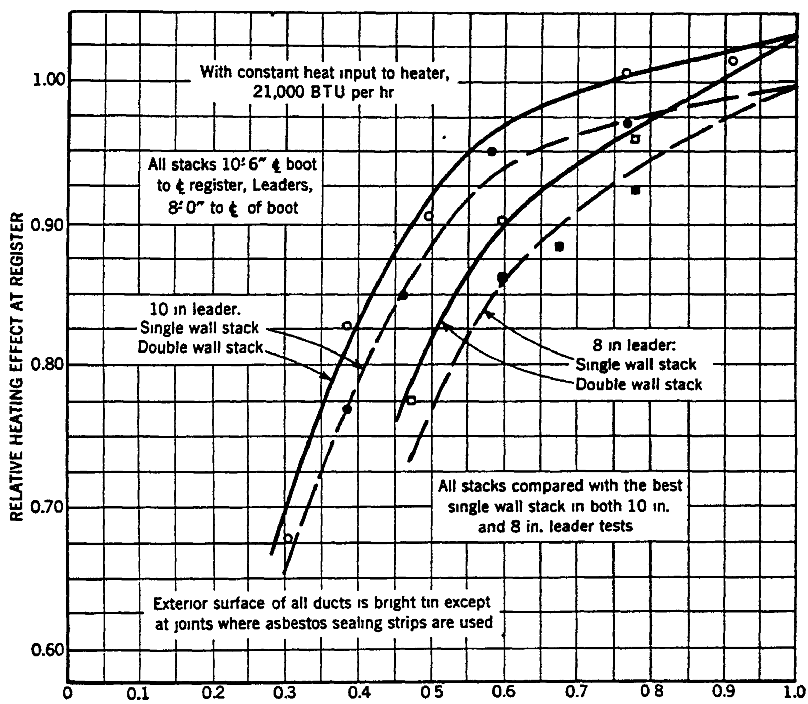


FIG. 3. RELATIVE HEATING EFFECT OF STACKS AT CONSTANT HEAT INPUT TO FURNACE

side wall locations for warm air registers in gravity circulating systems are not recommended on account of the tendency for stratification of the air in the room, resulting in high temperatures at the ceiling.

RECIRCULATING DUCTS AND GRILLES

The ducts through which air is returned to the furnace should be designed to minimize friction and turbulence. They, should be of ample area, equal to or slightly in excess of the total area of warm-air pipes, and

at all points where the air stream must change direction or shape, streamline fittings should be employed. Horizontal ducts should pitch at least ½ in. per foot upward from the furnace.

The recirculating grilles (or registers) should have a free area at least equal to the ducts to which they connect, and their free area should never be less than 50 per cent of their gross area.

The location and number of return grilles will depend on the size, details and exposure of the house. Small compactly built houses may frequently be adequately served by a single return effectively placed in a central hall. More often it is desirable to have two or more returns, provided, however, that in two-story residences one return is placed to effectively receive the cold air returning by way of the stairs.

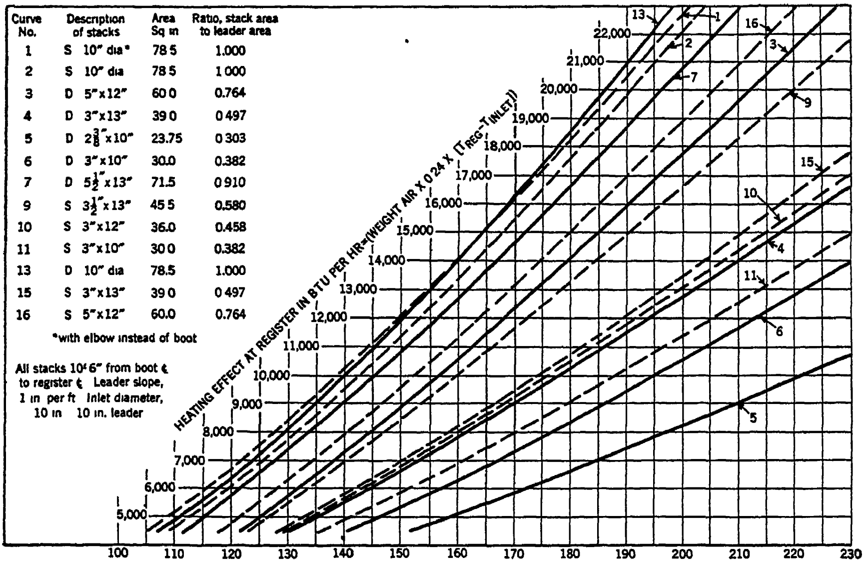


FIG. 4. HEATING EFFECT AT REGISTERS FOR VARIOUS STACKS WITH 10-IN. LEADER

Where a divided system of two or more returns is used, the grilles must be placed to serve the maximum area of cold wall or windows. Thus in rooms having only small windows the grille should be brought as close to the furnace as possible, but if the room has a bay window, French doors, or other large sources of cooling or leakage of cold air, the grille should be placed close by, so as to collect the cool air and prevent drafts. When long ducts of this type are employed they must be made oversize. This precaution is particularly important when long ducts and short ducts are used in the same system. The long ducts must be oversize, if they are to operate satisfactorily in parallel with short ducts.

Return ducts from upstairs rooms may be necessary in apartments or other spaces which are closed off or badly exposed. Metal linings are

advisable in such ducts. It is important that these ducts be free from unnecessary friction and turbulence, and that they be located to prevent preheating of the air before it reaches the furnace.

Furnace Return Connection

Circulation of the air is accelerated if the return connection to the furnace is through a round inclined pipe connected to two 45 deg elbows rather than through a vertical pipe connected to two 90 deg elbows. The top of the return shoe should enter the casing below the level of the grate in the furnace. In order to accomplish this the shoe must be wide as is indicated in Fig. 6, No. 1 arrangement.

Tests of six different systems of cold air returns, Fig. 6, made at the University of Illinois[2], resulted in the following conclusions:

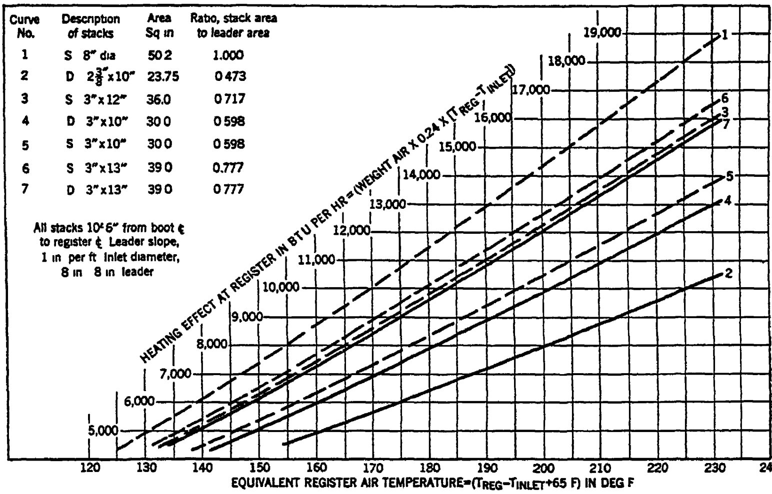


FIG. 5. HEATING EFFECT AT REGISTERS FOR VARIOUS STACKS WITH 8-IN. LEADER

1. In general, somewhat better room temperature conditions may be obtained by returning the air from positions near the cold walls.

2. Friction and turbulence in elaborate return duct systems retard the flow of air, and may seriously reduce furnace efficiency, and lessen the advantages of such a design.

3. The cross-sectional duct area is not the only measure of effectiveness. Friction and turbulence may operate to make the air flow out of all proportion to the various duct areas.

FURNACE CAPACITY

The size of furnace should, of course, be such as will provide the necessary air heating capacity, usually expressed in square inches of leader pipe area, and at the same time provide a grate of the proper area to burn the necessary fuel at a reasonable chimney draft. The total leader pipe area required is obtained by finding the sum of the leader pipe areas as already designated.

[2]Investigation of Warm Air Furnaces and Heating Systems, Part IV, by A. C. Willard, A. P. Kratz, and V. S. Day (University of Illinois, *Engineering Experiment Station Bulletin* No. 189).

The grate area will depend on several factors of which four are very important. First of all, the air temperature at the register for which the plant has been designed must be determined. Usually, this temperature is taken at 175 F. Second in importance is the combustion rate, *which must always correspond with the register air temperature*, as is shown by a set of typical furnace performance curves (Fig. 7) for a cast-iron, circular radiator furnace with a 23 in. diameter grate and 50 in. diameter casing. The third factor is efficiency, which is a function of the combustion rate, and varies with it as shown by the efficiency curve of Fig. 7. The fourth factor is the heat value per pound of fuel burned, which was 12,790 Btu. This is not shown on the curves since it was constant for all combustion rates.

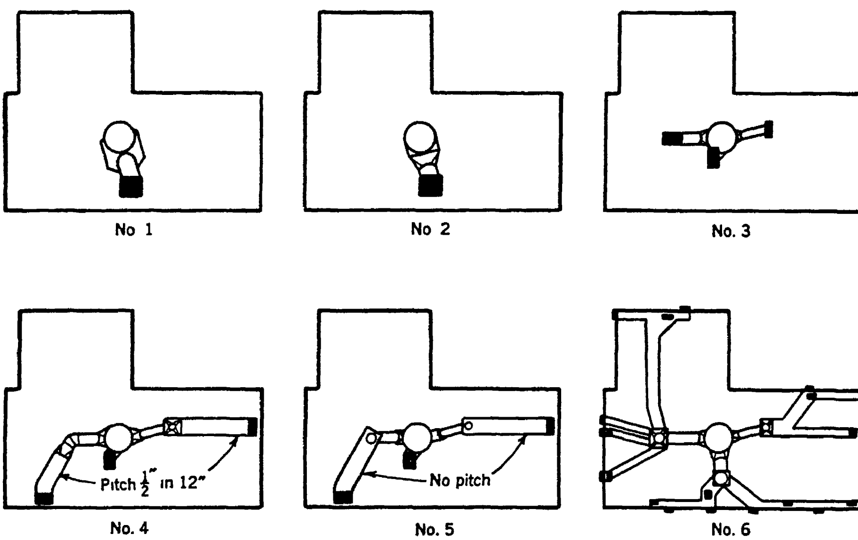


FIG. 6. ARRANGEMENT OF COLD AIR RETURNS FOR SIX INSTALLATIONS

It may be noted from Fig. 7 that for this particular furnace a register temperature of 175 F was accompanied by a combustion rate of approximately 7.5 lb per square foot per hour, a capacity at the bonnet of 152,000 Btu per hour and a furnace efficiency of 58 per cent. Under these conditions the capacity at the bonnet per square foot of grate was equivalent to a value of 52,800 Btu per hour and per square inch of grate was equivalent to 367 Btu per hour. If it is desired to use these curves to select a furnace to deliver air at 175 F register temperature in a house where the total heat loss is H Btu per hour and the loss between the furnace and the registers is 0.25 H Btu per hour, the area of the grate in square inches will be $\frac{1.25\,H}{367} = 0.0034\,H$.

If, on the other hand, it is desired to select a furnace to deliver air at 160 F register temperature, the combustion rate is 5.5 lb and the efficiency

of the furnace is 62 per cent. Under this condition the capacity at the furnace bonnet per square foot of grate is 43,200 Btu per hour and per square inch of grate is 300 Btu per hour, the required area of the grate in square inches in this case will be $\frac{1.25\ H}{300} = 0.0042\ H$. It should be noted that a larger grate area is required if the furnace is to deliver air at a lower register temperature.

The typical performance curves shown in Fig. 7 are not applicable to

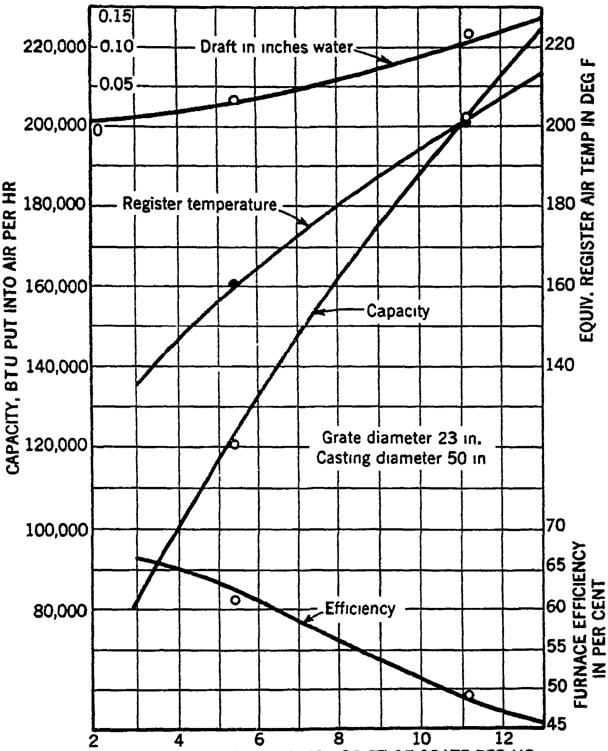


FIG. 7. TYPICAL PERFORMANCE CURVES FOR A WARM-AIR FURNACE AND INSTALLATION IN A THREE-STORY TEN LEADER PLANT, OPERATING ON RECIRCULATED AIR

all furnaces and hence for ordinary design purposes the values recommended in the Standard Code[3] should be used. The equation for a furnace having a ratio of heating surface to grate area of 20 to 1 is equal to:

$$H = \frac{G \times p \times f \times E_1 \times E_2 \times 0.866}{144} \qquad (7)$$

[3]Standard Gravity Code for the Design and Installation of Gravity Warm Air Heating Systems in Residences. This code has been sponsored by the *National Warm Air Heating and Air Conditioning Association*, the *National Association of Sheet Metal Contractors*, and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. It is recommended that the installation of all gravity warm-air heating systems in residences be governed by the provisions of this code, the tenth edition of which may be obtained from the *National Warm Air Heating and Air Conditioning Association*, 145 Public Square, Cleveland, Ohio.

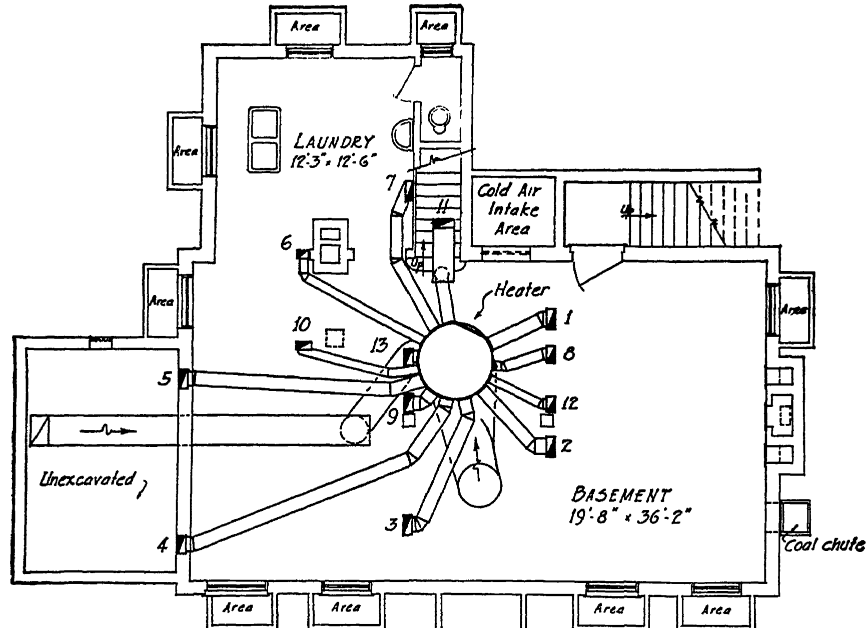


FIG. 8. BASEMENT PLAN, RESEARCH RESIDENCE

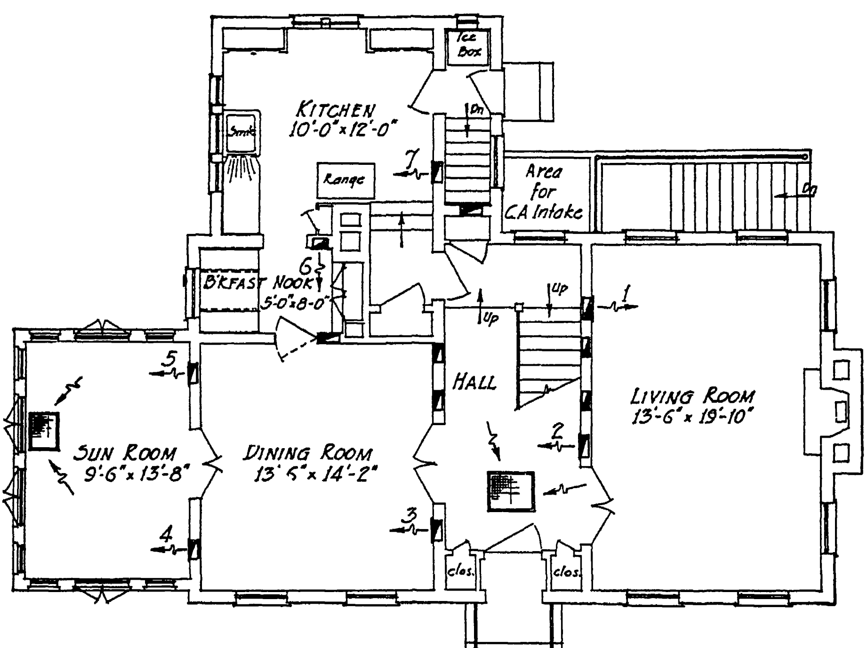


FIG. 9. FIRST FLOOR PLAN, RESEARCH RESIDENCE

where

G = grate area, square inch.
p = combustion rate, pound coal per square foot of grate per hour.
f = heating value of the coal, Btu per pound.
E_1 = efficiency at bonnet, ratio of heat delivered at bonnet to heat developed in furnace.
E_2 = efficiency of duct transmission, ratio of heat delivered at register to heat delivered at bonnet.
0.866 = factor of safety to allow for contingencies under service conditions such as accumulations of soot and ashes, ineffective firing methods, etc.
H = total heat loss from structure.

An addition of 2 per cent of the furnace capacity is proposed for each unit when the ratio of heating surface to grate area exceeds 20. This addition is based on tests[4] conducted at the University of Illinois on seven types of furnaces having varying ratios of heating surface to grate area. This correction does not, however, apply to values of the ratio less than 15 nor greater than 30.

By transposing the terms in Equation 7 and adding the correction term for ratios of heating surface to grate area other than 20 to 1, the following equation is obtained:

$$G = \frac{144 \times H}{p \times f \times E_1 \times E_2 \times 0.866\ [1 + 0.02\ (R\text{-}20)]} \qquad (8)$$

in which R = ratio of heating surface to grate area.

In the case of the Standard Code[5] the numerical values used in Equation 8 were based on those determined from the tests conducted on the different types of furnaces.

$$G = \frac{144 \times H}{7.5 \times 12{,}790 \times 0.55 \times 0.75 \times 0.866\ [1 + 0.02\ (R\text{-}20)]} \qquad (9)$$

$$G = 0.004205 \frac{H}{[1 + 0.02\ (R\text{-}20)]} \qquad (10)$$

As used in these calculations, H = Btu heat loss from the entire house per hour = summation of all room losses $H_1 + H_2$ + etc. + the Btu necessary to heat the outside air, if any, at intake. This outside air loss in Btu per hour will be approximately 1.27 times the cubic feet of air admitted through the intake per hour on a zero day. For systems which recirculate all the air this value will be zero. For systems which have an outside air intake, controlled by damper, this value might well be approximated, since this loss will probably be reduced to a minimum on a zero day. Assume for such cases that the building loss is increased by 25 per cent, and that there is the usual 25 per cent loss between furnace and registers.

TYPICAL DESIGN

The application of the preceding data to an actual example may be of assistance to the designer. Figs. 8, 9, 10 and 11 represent the plans of

[4]University of Illinois, *Engineering Experiment Station Bulletin* No. 246, by A. C. Willard, A. P. Kratz, and S. Konzo, Chapter X, pp. 126-146.

[5]Loc. Cit. Note 3.

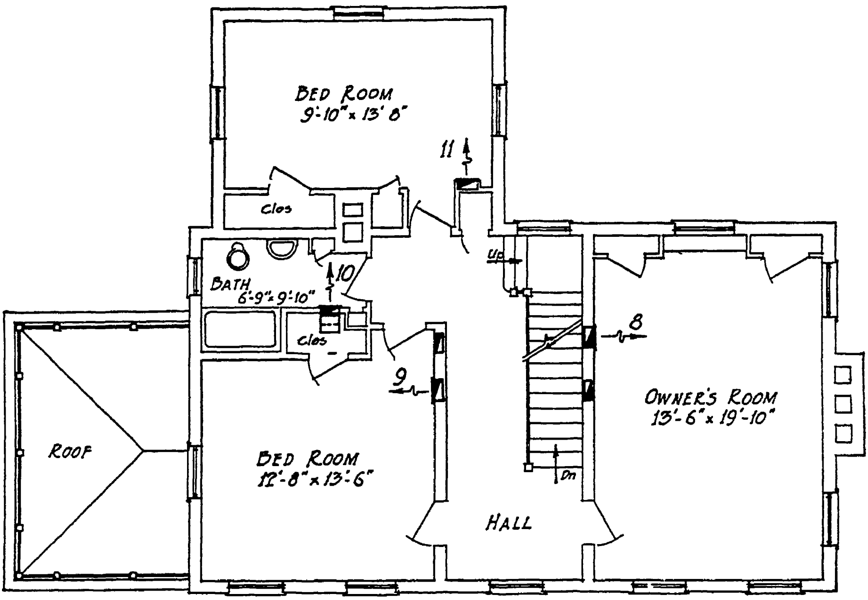


FIG. 10. SECOND FLOOR PLAN, RESEARCH RESIDENCE

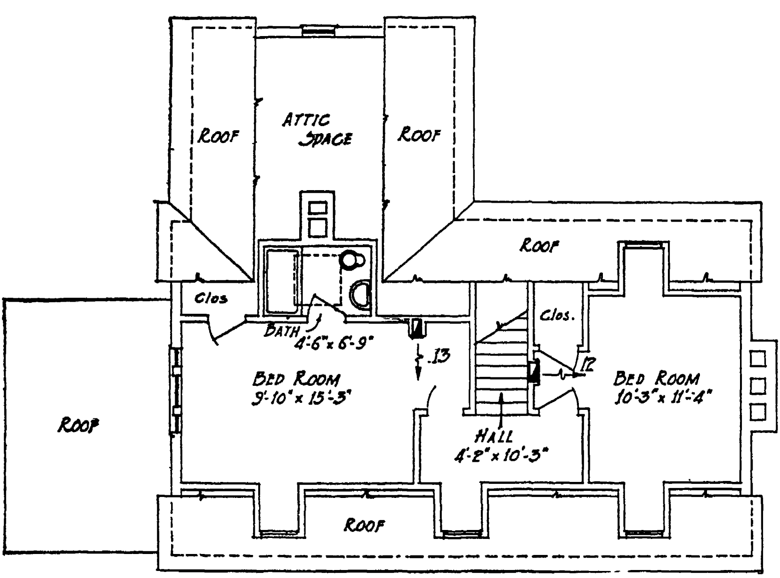


FIG. 11. THIRD FLOOR PLAN, RESEARCH RESIDENCE

the Warm Air Research Residence of the *National Warm Air Heating and Air Conditioning Association* erected at the University of Illinois[6].

Leaders, Stacks and Registers. (Direct Method)

Living Room, 1st floor:

17,250 ÷ 111 = 155 sq in. leader area. See summary Table 1; also see Art. 3 Sec. 1 of the Standard Gravity Code[7].

Leader diameter = 14 in.

Register size = 155 sq in. net area. Gross area = net area ÷ 0.7 = 14 in. × 16 in.

Owner's Room, 2nd floor:

15,030 ÷ 167 = 90 sq in. leader area. See summary Table 1; also see Art. 3 Sec. 2 of the Standard Gravity Code[7].

Leader diameter = 11.4, say 12 in.

Stack area = 0.7 × 90 = 63 sq in. = say 5 in. × 12 in.

Register area = 90 sq in. net area. Gross area = net area ÷ 0.7 = 12 × 12 or 12 in. × 14 in.

In like manner the leaders, stacks and registers are calculated for each room in the house.

Leaders, Stacks and Registers. (Code[7] Method. See Art. 3, Sec. 1, 2, 3)

Living Room (Glass = 90, Net wall = 405, Cubic contents = 2405)

$$\text{Leader} = \left(\frac{90}{12.6} + \frac{405}{57} + \frac{2405}{800} \right) 9 = 155 \text{ sq in.}$$

Register, same as Direct Method.

Owner's Room (Glass = 68, Net wall = 394, Cubic contents = 2275)

$$\text{Leader} = \left(\frac{68}{12.6} + \frac{394}{57} + \frac{2275}{800} \right) 6 = 91 \text{ sq in.}$$

Stack and Register, same as Direct Method.

Assuming all air recirculated, the minimum furnace for the plant will be:

Grate area = 0.0042 × 132,370 = 556 sq in.
Use 27 in. diameter grate. (Equation 10.)

If provision should be made for certain outside air circulation, then increase the building heat loss by, say, 25 per cent and obtain by Equation 10 a 30 in. grate.

Experiments at the University of Illinois[8] have shown that the capacity of a furnace may be increased nearly three times by an adequate fan, with a constant register or delivery temperature maintained, *provided that the rate of fuel consumption can be increased to provide the necessary heat.* In other words, the capacity of a forced circulation system *is limited by the ability of the chimney to produce a sufficient draft, and the ability of the fan to deliver an adequate amount of air.*

[6]Plans used with permission. Bathroom on third floor not heated.
[7]Loc. Cit. Note 3.
[8]University of Illinois, *Engineering Experiment Station Bulletin* No. 120, p. 129.

TABLE 1. SUMMARY OF DATA APPLIED TO WARM AIR RESEARCH RESIDENCE

Rooms	From Chapter 6 Estimating Heat Losses Btu Heat Losses H	Leader Area Sq In.	Stack Area Sq In. $0.7 \times LA$	Leader Diameter Inches	Stack Size Net	Register Size Gross
First Floor		$= 0.009H$				
Living	17250	155	----	14		14×16
Dining	6810	61	----	9		8×12
Breakfast	2300	21	----	8		8×10
Kitchen	9210	83	----	11 or 12		12×14
Sun	25710	230	----	Two 12		Two 12×14
Hall and stair	12570	113	----	12		12×14
Second Floor		$= 0.006H$				
Owner's	15030	90	63	11 or 12	5×12	12×14
S. W. Bed	9800	59	41	9	$3\frac{1}{2} \times 12$	8×12
Bath	2450	15	10	8	3×10	8×10
N. Bed	14800	89	62	11 or 12	5×12	12×14
Third Floor		$= 0.005H$				
E. Bed	8220	41	29	8	3×10	8×10
W. Bed	8220	41	29	8	3×10	8×10

BOOSTER FANS

Booster fans often may be arranged to operate when gas or oil burners are running and to stop automatically when the burners shut down. The booster equipment is most effective in increasing output at low operating temperatures. According to tests, efficiencies may be advanced from 60 per cent for gravity to 70 per cent with boosters at low operating temperatures, but at high operating temperatures gravity and booster efficiencies are almost identical[9].

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[9]University of Illinois, *Engineering Experiment Station Bulletin* No. 141 p. 79, and No. 246.

Chapter 20

MECHANICAL WARM AIR FURNACE SYSTEMS

Furnaces, Fans and Motors, Sound Control, Sprays and Filters, Air Distribution Design, Automatic Controls, Design of Heating System, Selecting the Furnace, Selecting the Fan, Heavy Duty Fan Furnaces, Humidification, Cooling Methods, Cooling System Design

MECHANICAL warm air or fan furnace heating systems[1], which are a special type of central fan systems, are particularly adapted to residences, small office buildings, stores, banks, schools, and churches. Circulation of air is effected by motor-driven fans instead of by the difference in weight between the heated air leaving the top of the casing and the cooled air entering its bottom, as in gravity systems described in Chapter 19. The advantages of mechanical systems, as compared with gravity systems are:

1. The furnace can be installed in a corner of the basement, leaving more basement room available for other purposes.

2. Basement distribution piping can be made smaller and can be so installed as to give full head room in all parts of the average basement, or be completely concealed from view except in the furnace room.

3. Circulation of air is positive, and in a properly designed system can be balanced in such a way as to give a greater uniformity of temperature distribution.

4. Humidity control is more readily attained.

5. The air may be cleaned by sprays or filters, or both.

6. The fan and duct equipment may be utilized for a complete cooling and dehumidifying system for summer, using either ice, mechanical refrigeration, or low temperature water for cooling and dehumidifying, or adsorbers for dehumidifying.

7. The use of the fan increases the volume of air which can be handled, thereby increasing the rate of heat extraction from a given amount of heating surface and insuring sufficient air volume to obtain proper distribution in a large room.

Much of the equipment used in central fan systems is the subject matter of other chapters. It is the purpose of this chapter to discuss the coordinated design and to deal in detail only with problems not covered elsewhere which refer particularly to the whole problem of fan warm air furnace heating and air conditioning.

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[1]See University of Illinois *Engineering Experiment Station Bulletin* No. 266 by A. P. Kratz and S. Konzo for details of tests conducted in Warm Air Research Residence.

Complete specifications for the furnace unit and the installed duct system are shown in The Yardstick for the Evaluation of a Forced Warm Air Heating System, obtainable from the *National Warm Air Heating and Air Conditioning Association*, 145 Public Square, Cleveland, Ohio.

FURNACES

Furnaces for mechanical warm air systems may be made of cast-iron, steel, or alloy. Cast-iron furnaces are usually made in sections and must be assembled and cemented or bolted together on the job. Steel furnaces are made with welded or riveted seams. The proper design of the furnace depends largely on the kind of fuel to be burned. Accordingly, various manufacturers are making special units for coal, oil and gas. Each type of fuel requires a distinct type of furnace for highest efficiency and economy, substantially as follows:

1. Coal Burning:
   a. *Bituminous*—Large combustion space with easily accessible secondary radiator or flue travel.
   b. *Anthracite or coke*—Large fire box capacity and liberal secondary heating surfaces.
2. Oil Burning:
   a. Liberal combustion space.
   b. Long fire travel and extensive heating surface.
3. Gas Burning:
   a. Extensive heating surface.
   b. Close contact between flame and heating surface.

A combustion rate of from 5 to 8 lb of coal per square foot of grate per hour is recommended for residential heaters. A higher combustion rate is permissible with larger furnaces for buildings other than residences, depending upon the ratio of grate surface to heating surface, firing period, and available draft.

Where oil fuel is used, care must be exercised in selecting the proper size and type of burner for the particular size and type of furnace used. It is recommended that the system be designed for blow-through installations, so that the furnace shall be under external pressure in order to minimize the possibility of leakage of the products of combustion into the air circulating system.

In residential furnaces for coal burning, the ratio of heating surface to grate area will average about 20 to 1; in commercial sizes it may run as high as 50 to 1, depending on fuel and draft. Furnaces may be installed singly, each furnace with its own fan, or in batteries of any number of furnaces, using one or more fans.

Furnace Casings

Casings are usually constructed of galvanized iron, 26-gage or heavier, but they may also be constructed of brick. Galvanized iron casings should be lined with sheet iron liners, extending from the grate level to the top of the furnace and spaced from 1 in. to 1½ in. from the outer casing. Casings for commercial or heavy duty furnaces, if built of galvanized iron, should be insulated with fireproof insulating material at least 2 in. thick. It is generally believed that either brick or sheet metal casing should be equipped with baffles to secure impingement of the air to be heated against the heating surfaces. Brick furnace casings should be supplied with access doors for inspection.

For furnace casings sized for gravity flow of air, where a fan is to be

used, some form of baffling must be employed if the desired results are to be expected. Many manufacturers recommend the use of special baffles to restrict the free area within the casing and to force impingement of the air against the heating surfaces. A method for making these baffles for furnaces with top horseshoe radiators and for furnaces with back crescent radiators is illustrated in Fig. 1.

Either square or round casings may be used. Where square casings are

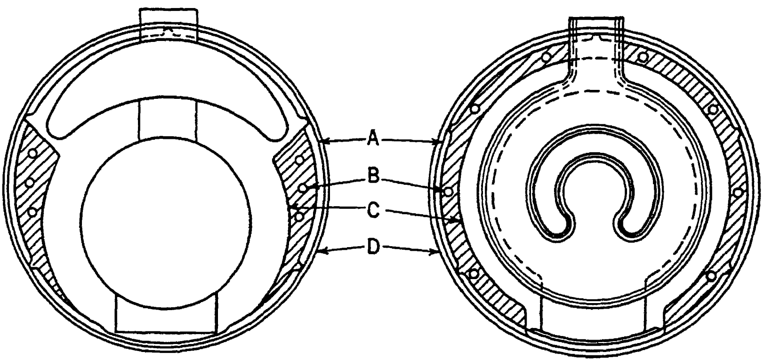

FIG. 1. USUAL METHOD OF BAFFLING ROUND CASINGS FOR FAN FURNACE WORK

A. Liner, 1 in. from casing. *B.* Hole to vent baffle.
C. Baffle, closed top and bottom. *D.* Outer casing

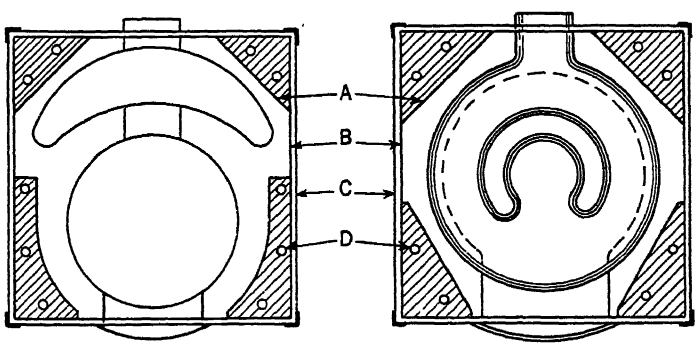

FIG. 2. METHOD OF BAFFLING SQUARE FURNACE CASING FOR FAN FURNACE WORK

A. Baffle, closed top and bottom. *B.* Liner, 1 in. from casing. *C.* Outer casing. *D.* Hole to vent baffle.

used, the corners must be baffled to reduce the net free area and to force impingement of air against the heating surfaces. Fig. 2 shows a satisfactory method of baffling square furnace casings for fan furnace work.

The hood or bonnet of the casing above the furnace should be as high as basement conditions will allow, to form a plenum chamber over the top of the furnace. This tends to equalize the pressure and temperature of the air leaving the bonnet through the various openings. It is generally considered advisable to take off the warm air pipes from the side of the bonnet near the top, as this method of take-off allows the use of a higher bonnet

and thus provides a larger plenum chamber. Fig. 3 illustrates a complete residence fan furnace installation showing location of fan, furnace, filters, plenum chamber and method of take-off of warm air pipe.

FANS AND MOTORS

Centrifugal type fans are most commonly used, and these may be equipped with either backward or forward curved blades. Motors may be mounted on the fan shaft or outside of the fan with belt connection. Multi-speed motors or pulleys are desirable to provide a factor of safety and to allow for increased air circulation. For additional information on fans and motors, see Chapters 30 and 36.

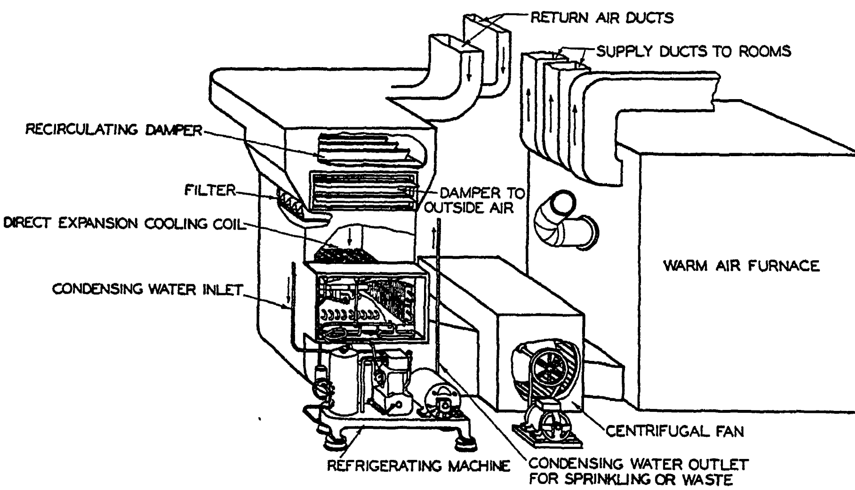

FIG. 3. COMPLETE RESIDENCE FAN FURNACE INSTALLATION FOR WINTER HEATING AND SUMMER COOLING

SOUND CONTROL

Special attention should be given to the problem of noise elimination. The fan housing should not be directly connected with metal, either to the furnace casing or to the return air piping. It is common practice to use canvas strips in making these connections. Motors and their mountings must be carefully selected for quiet operation. Electrical conduit and water piping must not be fastened to, nor make contact with fan housing. The installation of a fan directly under a cold air grille is not recommended on account of the noise objection. (See also Chapter 33.)

FILTERS

There are many satisfactory types of filters on the market. These include dry filters, viscous filters, oil filters and other types, some of which must be cleaned, some of which must be cleaned and recharged with oil, and some of which are inexpensive and may be discarded when they become dirty, and replaced with new ones.

The resistance of a filter must be considered in the design of the system since the resistance rises rapidly as the filter becomes dirty, thus impairing the heating efficiency of the furnace, in fact, endangering the life of the furnace itself. Manufacturers' ratings of filters must be carefully regarded, and ample filter area must be provided. Filters must be replaced or cleaned when dirty. (See also Chapter 29.)

AIR DISTRIBUTION

The conditions of comfort obtained in a room are greatly influenced by the type of register used and the locations of the supply registers and return grilles. In general it has been found that changes in the type, air velocity, and location of the supply register affect the room conditions much more than the changes in the location of the return grilles. Due to the economic considerations involved, it is common practice to locate the supply openings on the inside walls of a residence and the return openings nearest the greatest outside exposure. Many designers prefer, however, to locate the supply registers so that the warm air from the registers *blankets* a cold wall, and mixes with the cold air dropping off from the exposed walls. This may be accomplished by the use of a supply register placed close to an outside wall in such a position that the warm air sweeps the cold wall surface. The ducts leading to supply registers which are located on exposed walls should be adequately insulated to reduce the heat loss from the ducts.

Register and Grille Openings

Supply registers located in the floor are effective, but as they require frequent attention to keep them clean they should be avoided where another effective register location can be found. Tests conducted in the Warm Air Research Residence[2] have indicated that excellent results are obtainable with either high side wall or baseboard registers, providing a reasonable amount of precaution is employed. Baseboard registers should be of a deflecting-diffuser type which throws the air downward toward the floor and diffuses it at the same time. Register air temperatures under 125 F and air velocities over 500 fpm should be avoided as they may cause drafts.

High side wall registers must be of such type that the air is delivered horizontally or in a slightly downward direction, and must be so located as to avoid impingement of air on ceiling or wall. Directional flow diffusing type should be used to insure best results. Register air velocities should be such that the air stream carries to the opposite exposure. Velocities under 500 fpm are not recommended. Register air temperatures under 125 F are not objectionable. In fact, when cooling is desired, better air distribution is obtained with high side wall registers.

Unless registers, regardless of their location, are well proportioned and designed as well as decorated to harmonize with the trim, they may be unsightly. All registers should be equipped with dampers and must be sealed against leakage around the borders or margins.

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[2]Loc. Cit. Note 1.

Velocities through registers may be reduced by the use of registers larger than the connecting pipes. Some suggestions for equalizing velocities over the face area of the register by means of diffusers are illustrated in Fig. 4. Merely to use a larger register may not result in materially reduced velocities unless diffusers are used.

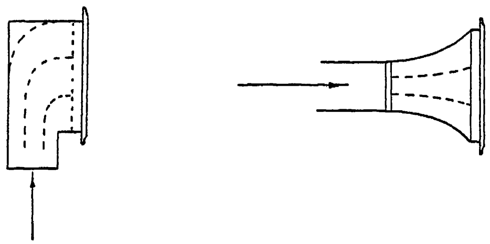

FIG. 4. DIFFUSERS IN TRANSITION FITTINGS TO EQUALIZE VELOCITIES THROUGH REGISTER FACES

Dampers

Suitable dampers are essential to any trunk or individual duct system, as it is virtually impossible to so lay out a system that it will be absolutely in balance without the use of dampers. Special care must be used in the design of any system to avoid turbulence and to minimize resistance. Sharp elbows, angles, and offsets should be avoided. (See chart giving loss of pressure in elbows, Chapter 32.)

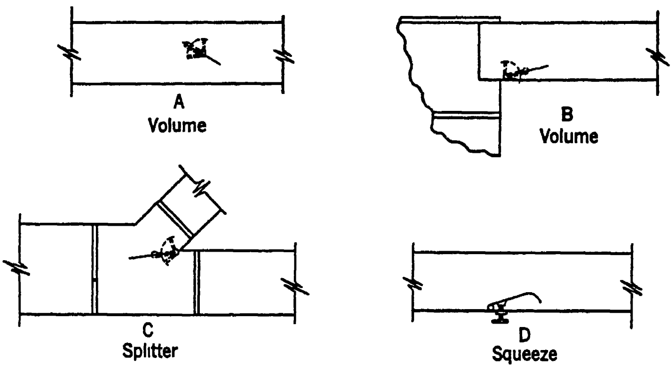

FIG. 5. THREE TYPES OF DAMPERS COMMONLY USED FOR TRUNK AND INDIVIDUAL DUCT SYSTEMS

Three types of dampers are commonly used in trunk and individual duct systems. *Volume dampers* are used to completely cut off or reduce the flow through pipes. (See *A* and *B*, Fig. 5.) *Splitter dampers* are used where a branch is taken off from a main trunk. (See *C*, Fig. 5.) *Squeeze dampers* are used for adjusting the volume of air flow and resistance through a given duct. (See *D*, Fig. 5.) It is essential that a damper be provided for each main or duct branch. A positive locking device should be used with each type of damper.

Ducts

The ducts may be either round or rectangular. The radii of elbows should be not less than one and one-half times the pipe diameter for round pipes, or the equivalent round pipe size in the case of rectangular ducts.

AUTOMATIC CONTROLS

Air stratification, high bonnet temperatures, excessive flue gas temperatures, and heat overrun or lag in a properly designed system can be largely eliminated through proper care in the planning and installation of the control system[3]. The essential requirements of the control are:

1. To keep the fire burning when using solid fuel regardless of the weather.
2. To avoid excessive bonnet temperatures with resultant radiant heat losses into the basement.
3. To avoid the overheating of certain rooms through gravity action during off periods of blower operation.
4. To have a sufficient supply of heat available at all times to avoid lag when the room thermostat calls for heat.
5. To prevent cold air delivery when heat supply is insufficient.
6. To avoid heat loss through the chimney by keeping stack temperatures low.
7. To provide quick response to the thermostat, with protection against overrun.
8. To provide for humidity control.
9. To provide a means of summer control of cooling.
10. To protect against fire hazards.

The following controls are desirable:

1. A *thermostat* located at a point where maximum fluctuation in temperature can be expected, in order to secure frequent operation of fans, drafts, and burners. This location would be near an outside wall but not upon it, in a sun room, or in a room with some unusual exposure. The thermostat, of course, should not be located where it will be affected by direct radiant heat from the sun or from a fireplace, or by direct heat from any warm air duct or register.

2. A thermostatic blower switch located in the bonnet to permit blower operation only between the temperatures of 100 F and 150 F. In certain extreme cases it may be necessary, or weather conditions may make it advisable, to adjust the high limit to a higher temperature than that given. Another location sometimes used for the blower switch is in the main duct near the frame opening from the bonnet.

3. A *protective limit control* located in the bonnet to shut down the system independently of the thermostat if the bonnet temperature exceeds 200 F.

4. On oil and gas burner installations, a control should be included which will shut down the system if the fire goes out or if there is a failure of the ignition system.

5. A *humidistat* to regulate the moisture supplied to the rooms.

6. On automatic stoker installations, a control is usually included which will start the operation regardless of thermostat settings whenever the bonnet temperature indicates that the fire is dying, or a time interval contactor is used that will start the stoker to run a predetermined length of time at predetermined intervals.

METHOD OF DESIGNING FORCED-AIR HEATING SYSTEMS

1. Determine heat loss from each room in Btu per hour. (See Chapter 6.)
2. Locate warm air registers and return registers on plans of house, beginning with the upper story rooms.

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[3]Automatic Controls for Forced-Air Heating Systems, by S. Konzo and A. F. Hubbard (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 37).

3. Sketch in duct layout to connect all registers and grilles with the central unit.

4. Determine equivalent length of duct for each register, allowing 10 diameters of straight pipe as equivalent to each 90 deg elbow having an inner radius not less than the diameter of the round pipe or the depth of the rectangular pipe.

5. Select a value for temperature of the air at the furnace bonnet. It is customary to use some value lying between 150 to 165 F. Use lower value if larger number of air recirculations is desired. It is recommended that the number of air recirculations should be in excess of 5 per hour.

6. Determine approximate value of temperature reduction in each duct caused by heat loss from the ducts. A value of from 0.3 to 0.6 F per foot of duct has been obtained from tests conducted in the Research Residence installation for uninsulated duct lengths up to approximately 60 ft.

7. Subtract this temperature reduction from the assumed bonnet air temperature to obtain an approximate value of the register air temperature for each register.

8. Determine the required air volume for each room from the following equation, or from the values listed in Table 1:

$$Q = \frac{H}{60 \times 0.24 \times d\,(t_r - 65)} \tag{1}$$

where

Q = required air volume, cubic feet per minute.
H = heat loss of room, Btu per hour.
d = density of air at register temperature, pounds per cubic foot.
t_r = register temperature, degrees Fahrenheit.
0.24 = specific heat of air.
65 = return air temperature.

For any given register temperature the solution of this equation simplifies to the following form:

$$Q = H \times \text{Factor} \tag{2}$$

in which the values of the Factor may be obtained from Table 1.

9. Determine register size from the air volume delivered to each room by the following formula:

$$\text{Free area of register, square feet} = \frac{Q}{V} \tag{3}$$

$$\text{Gross area of register, square feet} = \frac{\text{Free Area}}{R} \tag{4}$$

where

Q = required air volume, cubic feet per minute.
V = velocity at register face, feet per minute.
R = ratio of free area to gross area of register.

TABLE 1. FACTORS CORRESPONDING TO REGISTER TEMPERATURE FOR EQUATION 2

Register Temperature	Factor
110	0.0221
120	0.0184
130	0.0158
140	0.0140
150	0.0125
160	0.0114
170	0.0105

Allowable register velocities to be used in Equation 3 are approximately as follows:

Baseboard, non-deflecting type, maximum = 300 fpm.
Baseboard, deflecting toward floor, maximum = 500 fpm.
Baseboard, deflecting and diffusing = up to 800 fpm.
High side wall = not less than 500 fpm.

10. Duct systems for forced-air installations may consist of either trunk systems or individual duct systems.

Trunk Systems. Determine duct sizes and friction losses as outlined in Chapter 32, except that for residence applications the velocities in the main duct and in the various parts of the system should approximate the values recommended in Table 2.

Individual Duct Systems. An individual duct system is one having separate ducts extending from the heating unit to each register. In designing such a system select first the duct having the greatest equivalent length. Select a reasonable velocity using Table 2 as a guide. From friction chart in Chapter 32 determine unit friction loss per 100 ft of run, and from this the total friction loss in the duct selected. If this total friction loss exceeds a reasonable value a lower velocity should be used.

The remaining ducts are proportioned so that the total pressure in each duct is the same as that calculated for the longest duct. The added resistance necessary in the shorter ducts is accomplished by increasing the velocity in these ducts. No duct should be less than 6 in in diameter, nor should the velocity in any duct exceed approximately 1200 fpm. The final adjustment in a duct system may be made by employing dampers.

TABLE 2. RECOMMENDED VELOCITIES THROUGH DUCTS AND REGISTERS

Description	Low Velocity System (fpm)	Medium Velocity System (fpm)	High Velocity System (fpm)
Main ducts	500	750	1000
Branch ducts	450	600	750
Wall stacks	350	500	600
Baseboard registers (max.)	300	350	400
Wall registers above 5 ft (min.)	500	550	600

Instead of proportioning the ducts as outlined in the preceding paragraph it is more usual in practice to proportion all the ducts so that they have the same velocity as that used in the longest duct and to balance the system by employing dampers in the shorter ducts.

Return duct systems are designed making use of the same principles as those used in the design of supply duct systems. In this case the design may be based on the volume of air corresponding to the density of air existing in the return ducts, or in order to provide a factor for air leakage, it may be based on the same volume as used for the supply ducts.

11. Determine frictional resistance in:

a. Supply side of system as outlined in Item 10.
b. Return side of system as outlined in Item 10.
c. Furnace units, casing or hood, which is usually considered as equivalent to 0.03 to 0.10 in. of water.
d. Accessories such as washers or air filters, from manufacturer's data.
e. Inlet and outlet registers and grilles, from manufacturer's data.
f. Other accessory equipment such as cooling coils, from manufacturer's data.

Choose a fan which, according to its manufacturer's rating, is capable of delivering a volume of air, expressed in cubic feet per minute, against a frictional resistance, expressed in inches of water, computed by adding together the items listed in the preceding discussion. In practice it is recommended that liberal allowances should be made so that the

fan will be capable of delivering air against pressures that may not have been foreseen during the design of the duct system.

12. Select a furnace capable of delivering heat at the register outlets equal to the total heat loss of the structure to be heated.

The following formula may be used for coal burning furnaces:

$$G = \frac{H}{f \times p \times E_1 \times E_2 \, [1 + 0.02\,(R - 20)]} \quad (5)$$

where

G = required grate area, square feet.
H = total heat loss from building, Btu per hour.
f = calorific value of coal, Btu per pound.
p = combustion rate in pounds of fuel per square foot of grate per hour.
E_1 = furnace efficiency based on heat available at bonnet.
E_2 = efficiency of transmission based on ratio of heat delivered at register to heat available at bonnet.
R = ratio of heating surface to grate area.

In practice it is customary to use the following constants:

f = 12,000 (for specific values, see Table 5, Chapter 8).
p = 7.5 lb.
E_1 = 0.65 lower efficiency must be used with highly volatile solid fuel.
E_2 = 0.85.

The foregoing procedure for determining the size of the furnace to be used applies to continuously heated buildings.

13. Although intermittently heated buildings usually have their heat losses computed according to the standard rules for determining such losses, these rules do not take into account the heat which will be absorbed by the cold material of the building after the air is raised in temperature. This heat absorption must be added to the normal heat loss of the building to determine the load which the heating plant must carry through the warming-up process. It is customary to increase the normal heat loss figure by from 50 to 150 per cent depending upon the heat capacity of the construction material, the higher percentage applying to materials of high heat capacity such as concrete and brick. Fan furnace systems are well adapted for heating intermittently heated buildings as these systems do not require the warming of intermediate piping, radiators, or convectors, the generation of steam, or the heating of hot water.

14. Follow the same methods for an oil furnace as for coal where a conversion unit is to be used, making sure that the ratio of heating surface to grate area exceeds 20 to 1. If it does not, a size larger furnace should be selected. Use the manufacturer's Btu ratings of furnaces designed for exclusive use with oil, and select a burner with liberal excess capacity.

15. The selection of the proper size gas furnace for a constantly heated building can be easily made by using the following *American Gas Association* formula:

$$R = \frac{H}{0.9} \quad (6)$$

where

H = total heat loss from building, Btu per hour.
R = official *A.G.A.* output rating of the furnace, Btu per hour.

In the case of converted warm air furnaces a slightly different procedure is necessary, as the Btu input to the conversion burner must be selected rather than the furnace output. The proper sizing may be done by means of the following formula:

$$I = 1.59\,H \quad (7)$$

where

I = Btu per hour input.

The factor 1.59 is the multiplier necessary to care for a 10 per cent heat loss in the distributing ducts and an efficiency of 70 per cent in the conversion burner.

16. Specify location and type of all dampers in both supply air and return air sides of system. Specify controls including location of all thermostats. Arrange for proper control of humidifying equipment.

HEAVY DUTY FAN FURNACES

Fan furnaces for large commercial and industrial buildings are available in sizes ranging from 400,000 to 3,000,000 Btu per hour per unit. Heavy duty heaters may be arranged in combinations of one or more units in a battery. One typical arrangement is shown in Fig. 6.

Most manufacturers of heavy duty furnaces rate their furnaces in Btu per hour and also in the number of square feet of heating surface. Conservative practice indicates that at no time in the heating-up period should the furnace surface be required to emit more than an average of 3500 Btu per square foot. A higher rate of heat emission tends to increase the heat loss up the chimney, and raise fuel consumption, to shorten the life of the furnace, and to overheat the air. The ratio of heating surface to grate area on furnaces for this type of work should never be less than 30 to 1 and as indicated previously may run as high as 50 to 1.

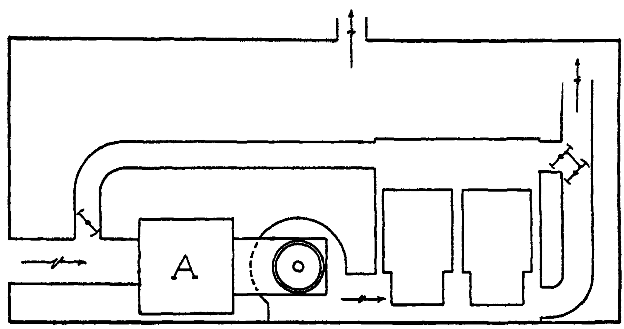

Fig. 6. Heater Arranged for Use of Air Washer or Filter (A) with Heated Air to Mix with Outside Air for Tempering, showing Mixing Damper from Warm Air and Tempered Air and Exhaust to Atmosphere

Control of temperature is secured through (1) controlling the quantity of heated air entering the room, (2) using mixing dampers, or (3) regulating the fuel supply.

The design of heavy duty fan furnace heating systems is in many respects similar to that of the central fan heating systems described in Chapter 21. Ducts are designed by the method outlined in Chapter 32.

HUMIDIFICATION

Mechanical warm air systems offer a means of proportioning and distributing moisture-bearing air; consequently, during the winter months

humidifiers may be employed to deliver water vapor to the fan-driven air stream in proper amounts to produce a more humid atmosphere, with increased comfort for people and increased life for household furnishings. Temperatures and relative humidities should be governed within the limits of the generally accepted standards. See Chapters 2 and 27 for more detailed information on this point.

In earlier types of furnaces, water evaporating pans were usually placed in the cool portions of the air stream, but modern types usually locate them in air which has been heated by contact with the heating surfaces. To change water into vapor capable of being carried in an air stream as part of the mixture, about 1000 Btu per pound are required. Without the addition of this heat, termed the latent heat of evaporation, water injected into the air will be carried along in the form of tiny globules until it falls out of the stream or is deposited upon some surface. Furthermore, when dry air is in contact with water for a sufficient length of time without the presence of a sizable body of water or a source other than air from which this latent heat of evaporation can be taken, such heat is supplied from the air. There is, therefore, a trend in present practice toward heating the water in addition to heating the air. Equipment for doing this may make use of sprays, or it may take the form of water circulating coils placed within the combustion chamber and connected by pipes to the humidifier pans where a constant water level is maintained by some separate float device. (See Chapter 27.)

Sprays for residence systems may be provided in separate housings to be installed on the inlet or outlet side of the fan, or they may be integral with the fan construction. They operate at water pressures of from 10 to 30 lb and use two or more spray nozzles for washing and humidification. The sprays should be adjusted to completely cover the air passages.

Sprays are usually controlled by solenoid valves wired in parallel with the fan motor. The water supply may, in turn, be controlled by a humidity-controlling device located in one of the living rooms, so that the washer will operate at all times when the fan is in operation, unless the relative humidity should rise beyond a desirable percentage. Sprays used in connection with commercial or heavy duty plants should be a regulation type of commercial spray.

Residence Requirements

The principles underlying humidity requirements and limitations for residences are summarized in *University of Illinois Bulletin* No. 230[4], as follows:

1. Optimum comfort is the most tangible criterion for determining the air conditions within a residence.

2. An effective temperature of 65 deg[5] represents the optimum comfort for the majority of people. Under the conditions in the average residence a dry-bulb temperature of 69.5 F with relative humidity of 40 per cent is the most practical for the attainment of 65 deg effective temperature.

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[4]See Humidification for Residences, by A. P. Kratz (University of Illinois, *Bulletin* No. 230).

[5]66 deg is the optimum winter effective temperature recommended by the A.S.H.V.E. Committee on Ventilation Standards.

3. Evaporation requirements to maintain a relative humidity of 40 per cent in zero weather depend on the amount of air inleakage to the average residence, and vary from practically nothing to 24 gal of water per 24 hours.

4. Relative humidity of 40 per cent indoors cannot be maintained in rigorous climates without excessive condensation on the windows unless tight-fitting storm sash or the equivalent is installed.

5. The problems of humidity requirements and limitations cannot be separated from condensations of good building construction, and the latter should receive serious attention in the installation of humidifying apparatus.

The following conclusions were drawn from the experimental results reported in the aforementioned bulletin:

1. None of the types of gravity warm air furnace water pans tested proved adequate to evaporate sufficient water to maintain 40 per cent relative humidity in the Research Residence except only in moderately cold weather.

2. The water pans used in the radiator shields tested did not prove adequate to maintain 40 per cent relative humidity in a residence similar to the Research Residence when the outdoor temperature approximated zero degrees Fahrenheit.

COOLING METHODS

A slight cooling effect may be obtained under certain conditions by the use of basement air. A more positive cooling effect may be obtained through air washers where the temperature of the water is sufficiently low (55 F or lower), and where a sufficient volume of water can be provided. Unless the temperature of the leaving water is below the dew-point temperature of the indoor air at the time the washer is started, both the relative and absolute humidities will be somewhat increased.

Coils of copper finned tubing through which cold water is pumped are available for cooling. They require less space than air washers and have the advantage that no moisture is added to the air when the temperature of the water rises above the dew-point. Ample coil surface is necessary with this type of cooling.

It is thoroughly feasible to use ice or mechanical refrigeration in connection with the fan and duct system for the heating installation, and to cool the building by this method, provided the building is reasonably well constructed and insulated. Windows and doors should be tight, and awnings should be supplied on the sunny side of the building. (See also Chapters 21 and 24.)

Results at Research Residence

The following conclusions may be drawn from the studies thus far completed in the Research Residence, subject to the limitations of the conditions under which the tests were run[6]:

1. An uninsulated building of ordinary residential type may require the equivalent of three tons of ice in 24 hours on days when the maximum outdoor temperature reaches 100 F if an effective temperature of approximately 72 deg is maintained indoors.

2. The use of awnings at all windows in east, south, and west exposures may result in savings of from 20 to 30 per cent in the required cooling load.

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[6]A.S.H.V.E. Research Report No. 947—Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo (A.S.H.V.E. Transactions, Vol. 39, 1933, p. 95). A.S.H.V.E. Research Report No. 979—Study of Summer Cooling in the Research Residence for the Summer of 1933, by A. P. Kratz and S. Konzo (A.S.H.V.E. Transactions, Vol. 40, 1934, p. 167).

3. The cooling load per degree difference in temperature is not constant but increases as the outdoor temperature increases.

4. The heat lag of the building complicates the estimation of the cooling load under any specified conditions and makes such estimates, based on the usual methods of computation, of doubtful value.

5. The seasonal cooling requirements are extremely variable from year to year, and the ratio between the degree-hours of any two seasons occurring within a 10-year period may be as high as 7.5 to 1. Hence an average value of the degree-hours cooling per season is comparatively meaningless.

6. The duct system in a forced-air heating installation can be successfully converted to a system for conveying cool air for the purpose of cooling the structure. No condensation of moisture was observed when the duct temperatures were not less than 65 F.

7. Cooling by means of water at a temperature of 60 F is not satisfactory unless an indoor temperature of less than 80 F is maintained.

8. In the selection of cooling coils, the frictional resistance of the coil to flow of air must be given careful consideration.

9. Cooling the structure by introducing large quantities of air from outdoors at night tended to reduce the amount of cooling required on the following day and was a practical means of providing more comfortable conditions in those homes where cooling systems were not available.

METHOD OF DESIGNING COOLING SYSTEM

The general procedure for the design of a cooling system in a forced-air installation is as follows:

1. Calculate heat gain for each room or space to be conditioned. (See Chapters 4 and 7.) Allowance for addition of outside air must be included in this calculation.

2. Select a temperature of air leaving supply inlets. In Research Residence tests[7] a value of from 65 to 70 F was found satisfactory.

3. Determine indoor conditions to be maintained. In Research Residence 80 F dry-bulb and 45 per cent relative humidity were found satisfactory.

4. Determine the quantity of air to be introduced into each room. (See Chapter 21.)

5. Estimate heat loss in duct system between cooling unit and supply registers.

6. Calculate the heat to be removed by the cooling unit, in the form of sensible heat and latent heat.

7. Determine size of ducts in duct system and size of registers, as explained in this chapter under the heading of Method of Designing Forced-Air Heating Systems.

8. Determine pressure loss in duct system and select fan as also explained in the same section.

9. Select cooling unit from manufacturer's data. Specify temperature and pressure of available cooling water, voltage and characteristics of electrical supply, and method of control of apparatus.

10. Select cooling coils from manufacturer's data to take care of latent heat load and to give required drop in air temperature with the weight of air flowing. (See Chapter 26.)

11. If system is to be used for both winter heating and summer cooling, duct sizes must be checked to insure that velocities and friction losses are reasonable for both conditions of operation. Adjustable dampers will be necessary to make changes in air distribution for the two seasons. Provision must also be made for changing fan speeds for summer and winter operation.

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[7]Loc. Cit. Note 6.

Chapter 21

CENTRAL SYSTEMS FOR COMFORT AIR CONDITIONING

Types of Systems for Ventilating, Heating, Air Conditioning, Factors Involved in Use and Design of Systems, Design Procedure

THE purpose of this chapter is to present a discussion of types of central systems usually encountered, together with a discussion of the factors involved in use and design and an outline of design procedure.

Insofar as this chapter is concerned, a central system is defined as a field assembled apparatus, comprising such elements of equipment as are necessary to fulfill the purpose for which it is designed, and serving one or more conditioned spaces. It may be argued with justification that a factory produced unit, including all the essential items of equipment can be employed as a central system. Unitary equipment is discussed in Chapter 23. Further, this chapter is confined to comfort air conditioning systems as such, and ventilating systems, warm air heating systems, together with central systems of a special nature, are excluded from the discussion.

This chapter assumes a knowledge of all the component parts of a system and the reader is referred specifically to other chapters covering design conditions and physiological principles, cooling and heating load; spray equipment, heat transfer surface coils, cooling dehumidification and dehydration, fans, air cleaning devices, refrigeration, air distribution and air duct design, automatic controls and instruments. In addition, the engineer should refer to the Code of Minimum Requirements for Comfort Air Conditioning[1] prepared by the joint committee of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *American Society of Refrigerating Engineers*, and to national, state or local codes that may apply.

CLASSIFICATION OF SYSTEMS

The generally accepted method of classifying systems is with regard to their function. A given type of system may be changed by the omission of certain of its functions or by the inclusion of others. As an example, a winter air conditioning system, by the omission of humidifying sprays, air cleaning devices, etc., will become a simple warm air heating system,

[1]Code of Minimum Requirements for Comfort Air Conditioning (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 27). Reprints of this code are available at $.10 a copy.

USE OF CENTRAL SYSTEMS

Several factors must be considered in deciding on whether or not to use a central system and in deciding on the type of central system and modifications required. These factors are:

1. Comparative effectiveness.
2. Characteristics and requirements of the load.
3. Space requirements.
4. Initial cost.
5. Operating costs and maintenance.

The comparative effectiveness of the type of system and modification plays a major part in the choice and is to a large extent affected by the other factors. One great advantage provided by the central system lies

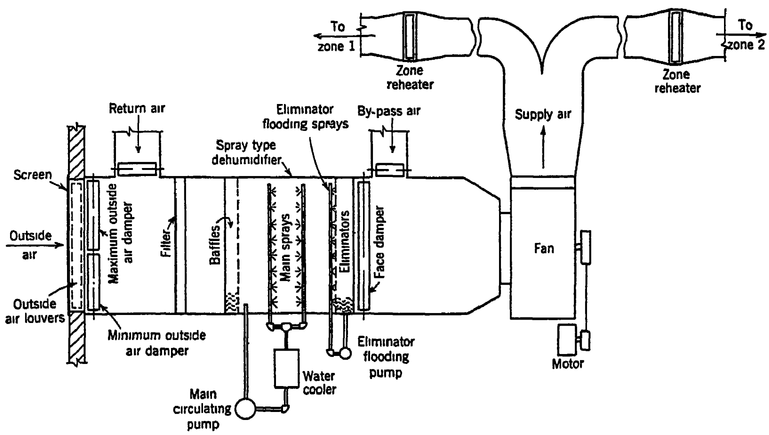

FIG. 4. CENTRAL SYSTEM WITH ZONING BY REHEATING

in its ability to diffuse odors and smoke which may occur in parts of the system, so that the outside air is determined by the average instead of the sum of the peak requirements. However, caution must be used where odors are apt to be objectionable even if greatly diluted. In such cases a positive exhaust to the outdoors or a separate treatment for the particular locality is recommended. The *averaging ability* both as to odors and thermal effects of the system is one item to be considered in studying the comparative effectiveness, particularly where adequate zoning is to be provided. Full advantage of diversity and non-simultaneous peak load requirements can be taken in determining the dehumidified air quantity where adequate zoning is provided.

The characteristics and requirements of the load frequently are the deciding factors. Wide, non-simultaneous variations in load between spaces or parts of the same space, indicate the necessity of zoning. Isolated spaces having a short time occupancy or brief load duration may be handled by units advantageously at times. The occurrence of simultaneous heating and cooling requirements in spaces having the same exposure or

on the same zone presents a problem to be studied. Ability to maintain conditions during the intermediate seasons without the use of refrigeration must be considered at all times, particularly in those applications where a high internal load exists.

The matter of space requirements may rule out one type of system or another. The avoidance of the use of rentable and usable space for equipment and ductwork, in office buildings, stores, etc., is most important since the loss of revenue is directly chargeable to the operating cost of the system. Consideration of the spaces available with respect to the type of system and method of distribution used and their effect on overall cost is essential.

Initial cost is given first consideration all too frequently. While elaborately designed installations are seldom justifiable, proper consideration must be given to operating costs and maintenance, performance required, and the life of the system. The increase in cost incurred by

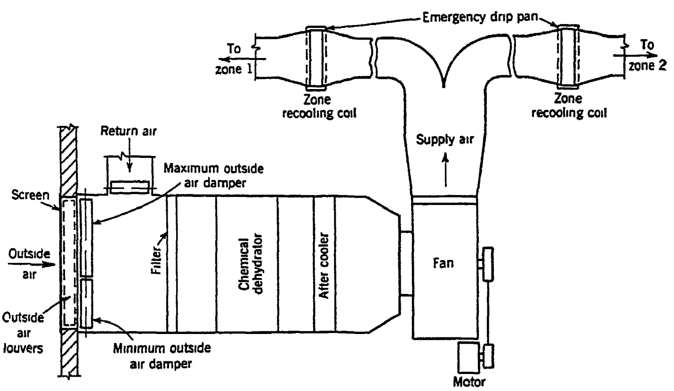

FIG. 5. SUMMER CENTRAL SYSTEM WITH ZONING BY RECOOLING

suitable zoning can be offset partially by reduced quantities of dehumidified air with smaller air handling apparatus, and partially by reduced refrigeration requirements. In the end it may prove less expensive than an unsatisfactory single zone system. A small increase in initial cost to provide better access to equipment, better airflow and distribution, and better zoning, usually will pay for itself.

Operating costs often receive too little attention. Proper relationship between equipment selected and the load to be carried must be obtained. Cooling systems in general operate at maximum capacity less than 20 per cent of the time and heating systems operate under design conditions but a few days in the year. Good partial load performance is essential for low operation costs. Proper zoning, good arrangement and selection of equipment and type of system all tend to reduce operation costs. Central systems having the bulk of equipment in a central location properly arranged, and having only the equipment required for zoning distributed through the conditioned space or spaces, will have relatively low maintenance costs. Care must be used to provide ease of access to important equipment and those items requiring servicing.

DESIGN OF SYSTEMS

Various factors are to be considered in the design of a central system, some of which have been touched upon in the foregoing as a modification or economic factor, while others are a part of the normal design procedure. These, in the order of usual occurrence are discussed herewith. For practical purposes these factors cover *Year 'Round Air Conditioning Systems.* Those directly applicable to summer only or winter only systems are to be viewed accordingly.

Design Conditions

Physiological principles and design conditions are covered in Chapter 2. A detailed study of these is beyond the scope of this chapter other than certain recommendations with regard to their application.

Where extreme or unusual outdoor conditions prevail for long periods of time, such as in the tropics, at high altitudes, in regions of extremely

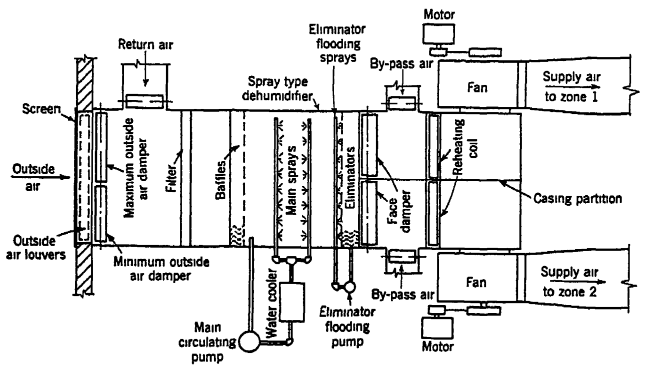

FIG. 6. CENTRAL SYSTEM USING MULTIPLE FANS WITH BY-PASSES AND REHEATING FOR ZONING

low humidities, etc., due allowance must be made for the fact that the people have become acclimatized, to a certain degree at least, to these conditions and the inside conditions should be selected accordingly.

A change in the inside conditions from a set standard sometimes is warranted from an economic standpoint. It is possible at times to make substantial savings in initial and operating costs by maintaining a lower temperature and higher humidity in the conditioned space while maintaining the same effective temperature.

In winter, the matter of condensation on windows, walls, etc., is of extreme importance. Humidities low enough to avoid this should be maintained, and when it is necessary to carry the higher humidities, double glass should be used or suitable means of handling the condensation should be provided.

During intermediate seasons the use of refrigeration as a source of cooling may be undesirable from an operating cost standpoint depending on the local energy rate structure and demand charges. As a result the

use of outdoor air, either directly or indirectly, as a source of cooling will be required and the system may have to operate on an unusual basis with regard to the temperatures and humidities that can be maintained. This situation should be carefully investigated. As a rule, provision should be made for introducing all outdoor air into the system during intermediate seasons as described later.

With regard to outside conditions it must be noted that where systems are to operate at certain hours of the day only, the outside conditions that occur during these times can be used providing that the cumulative effect or heat gain lag is taken into consideration in the estimate.

Outdoor Air

Standards affecting the quantity of outdoor air have been established in Chapter 2. These standards relate the minimum amount of outside

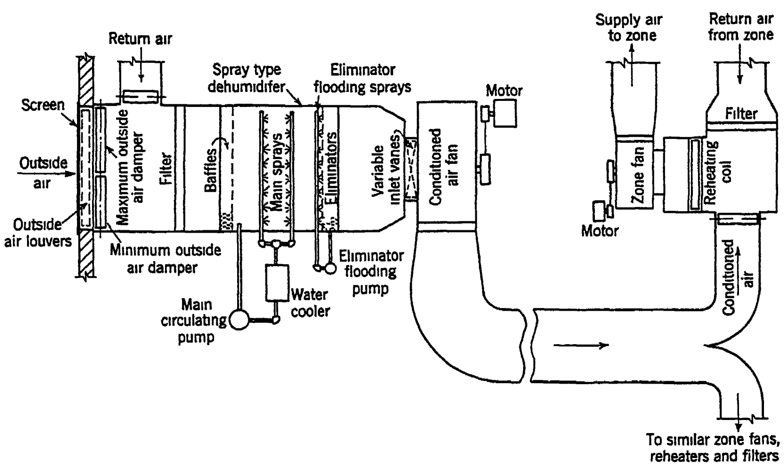

FIG. 7. CENTRAL SYSTEM WITH CENTRAL FAN AND CONDITIONER AND WITH INDIVIDUAL ZONE FANS

air to be introduced into the conditioned space for both the number of occupants and the type of occupancy, *i.e.*, people smoking, etc. This, of course, is the common-sense approach to the problem, but there are some cases, such as those spaces having a very low occupancy with regard to the cubical content where mustiness may develop unless a sufficient air change is provided. Where such a condition exists in a few spaces out of several which are being handled by a central system, the *averaging* ability of the central system may cope with the situation satisfactorily. This is due to the fact that the return air from the particular space is mixed thoroughly with the return air from all the other spaces, and all the outside air supplied to all the other spaces.

It should be noted that the minimum quantity of outdoor air is affected by infiltration and leakage. Infiltration will reduce the quantity to be introduced by the system while leakage may have to be offset by an increase in the quantity of outdoor air.

Most summer and year 'round air conditioning systems should be so designed and arranged that a quantity of outside air at least equal to the quantity of cooled or dehumidified air can be taken in when desired. In the intermediate seasons and in the cooling season it is more economical to take all outside air into the dehumidifier when the outdoor air wet-bulb temperature is lower than the inside wet-bulb temperature to be maintained. Also, where using a spray type dehumidifier or air washer, whenever the outdoor wet-bulb temperature is below the apparatus dew-point temperature required, the system can be operated on an evaporative cooling basis, dispensing with the need for refrigeration though cooling may be required. Automatic controls are available for accomplishing this and on reasonably large systems are justifiable from an economic standpoint.

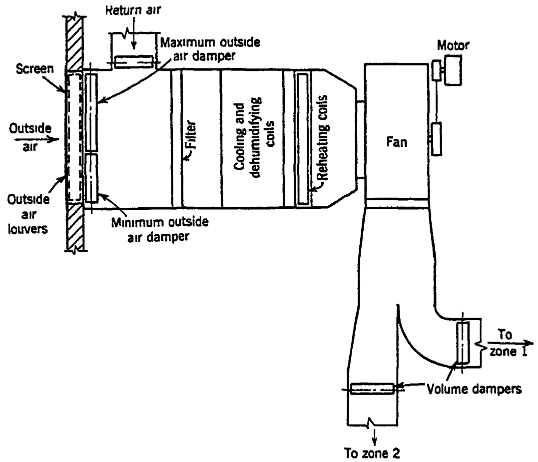

FIG. 8. CENTRAL SYSTEM WITH ZONING BY VOLUME CONTROL

Cooling Load

The method of determining the cooling load for a conditioned space or spaces is outlined in Chapter 7. As pointed out therein, many of the items of heat gain are variable and do not reach their maximum values simultaneously. Proper consideration of these *peaks* and the avoidance of pyramiding these peaks in the cooling load calculations is stressed. Maximum solar heat gain on an east exposure is seldom coincident with the maximum outdoor wet-bulb.

A large difference in the incidence of the peaks between various spaces or parts of the same space indicates the necessity for zoning. In a building having an east and west exposure where solar heat gain forms a fair share of the cooling load, the times of their individual peaks are apt to be hours apart, and the peak load of one plus the off peak load of the other will be substantially less than their combined peak loads. Proper zoning will permit taking full advantage of this condition or similar conditions of non-simultaneous peaks and will result in a lower total load, reflecting itself in savings in equipment.

A factor, similar in effect and closely related to the non-simultaneous occurrence of peak loads is *diversity*. Typical of this is the case of a large department store where the air handling equipment serving a certain space must be sufficient to handle the load created by the throngs of people attending sales in that space. Under such a condition the number of people in other spaces is usually normal or below. While this means that the air handling equipment for certain departments must be large enough to cope with the situation, the refrigeration equipment must be only large enough to handle the average maximum. If a system employing zone recirculating fans and a single central fan and dehumidifier were used, the saving would be reflected in the capacity of the central fan and dehumidifier. Another example of this diversity is found in an office building having restaurants and stores of certain types on the first floor

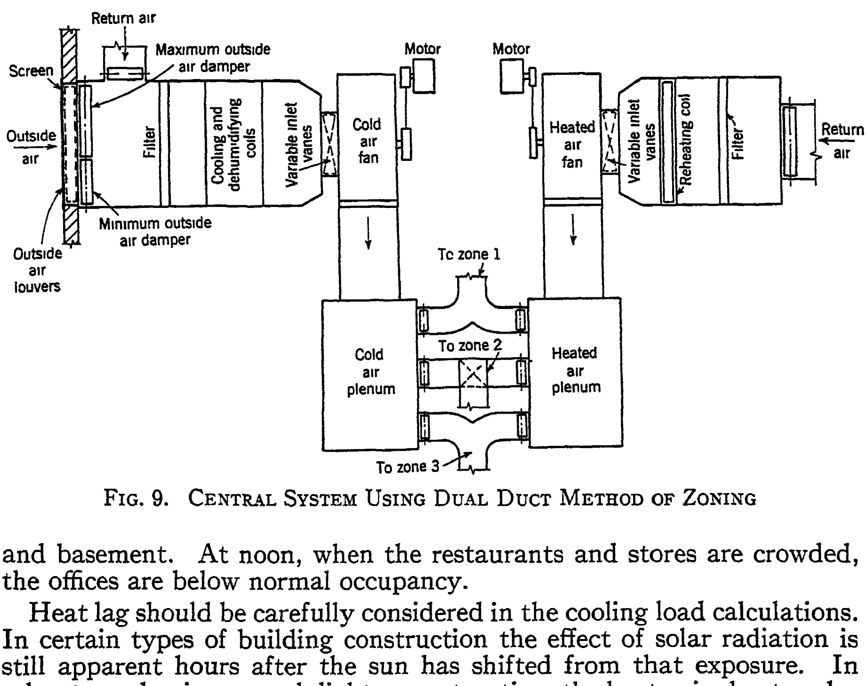

FIG. 9. CENTRAL SYSTEM USING DUAL DUCT METHOD OF ZONING

and basement. At noon, when the restaurants and stores are crowded, the offices are below normal occupancy.

Heat lag should be carefully considered in the cooling load calculations. In certain types of building construction the effect of solar radiation is still apparent hours after the sun has shifted from that exposure. In other types having a much lighter construction, the heat gain due to solar radiation decreases markedly with the passing of the sun. Some walls, having been warmed by the sun, may radiate heat long after the incidence of the sun, requiring lower inside temperatures to offset the radiant energy.

Storage effect is usually present in some degree. Often it can be utilized to great advantage and it has more than once provided an unknown safety factor. If a space is kept below the design inside temperature for a period of time, the interior walls, floors, furniture and fixtures begin to assume the temperature of the space. If the period of time is sufficient, the entire mass may reach the room temperature, rather than just the skin or surface of the item. If the space has been precooled below the design maximum

temperature for a period of time prior to the advent of the peak load, when the heat gain begins to increase to peak conditions, some of the increase is used in raising the temperature of the furniture, fixtures, etc., to the design conditions and the cooling load can be reduced accordingly. However, unless very accurate data with regard to the mass, surface, specific heat, etc., of the items within the space are available, due caution must be used in discounting the cooling load for this storage effect. In the absence of reliable data it is often a matter of experience rather than calculation.

Where air conditioning supply and return ducts pass through unconditioned spaces there will be a transfer of heat from these spaces to the air in the ducts, even though these ducts are well insulated. An allowance

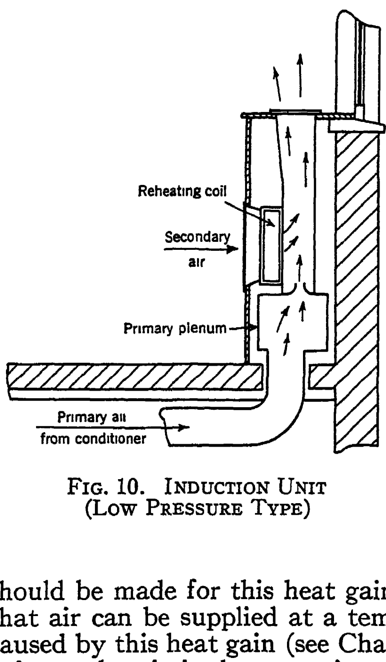

FIG. 10. INDUCTION UNIT (LOW PRESSURE TYPE)

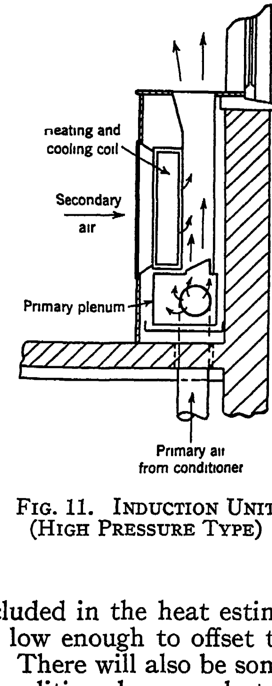

FIG. 11. INDUCTION UNIT (HIGH PRESSURE TYPE)

should be made for this heat gain and included in the heat estimate so that air can be supplied at a temperature low enough to offset the rise caused by this heat gain (see Chapter 43). There will also be some heat gain to the air in ducts passing through conditioned spaces, but since a cooling effect is produced in the space through which the duct passes, this is not a loss and usually can be compensated for by adjustment of air quantities between the various spaces.

Heating Load

Methods of calculating the heating load are shown in Chapter 6. Many of the factors outlined previously under Cooling Load, such as zoning, non-simultaneous peaks, and diversity, apply in the reverse manner due to the heating requirement instead of the cooling requirement. However, these factors enter into the heating load picture from a stand-

point of control of inside conditions, overall performance and economy of operation more than from a capacity of equipment standpoint.

Where heating is concerned it is not only necessary to heat a building or space to its design conditions when there is but the merest fraction of normal occupancy, practically no lights, internal heat, or solar radiation, but it is also necessary to provide capacity to heat the building quickly after a shut-down such as when a sudden cold snap follows relatively warm weather, or after a week-end or holiday. However, in normal operation during week-ends and holidays, buildings are usually kept at a *holding* temperature to prevent the freezing of services and conserve fuel. In many cases it requires less fuel to keep a building or space at a temperature of 50 to 65 F for some time than to shut the system down and then bring the temperature up again.

Apparatus Dew-point

The term *apparatus dew-point* is commonly applied to the temperature of the air leaving the dehumidifier, or in the case of winter air conditioning systems, the humidifier. To a certain extent this is a misnomer since only in an apparatus having a saturating efficiency of 100 per cent is the dry-bulb temperature of the air leaving the dehumidifier equal to the dew-point temperature. This is seldom encountered in actual practice, and some commercial dehumidifiers may have a spread of 3 to 5 F between the dew-point and dry-bulb temperatures. Note that in winter, saturation of the air is usually undesirable for practical reasons. The determination of the apparatus dew-point required for a system is one of the most important steps in the design of the system since the minimum air quantity that can be used on a given system is dependent on this.

For summer cooling, in order to maintain a given temperature and humidity within a space, *both latent and sensible heat gains must be absorbed by the air supplied to the space in the exact ratio in which these occur.* This ratio may be expressed in a number of ways such as latent heat to sensible heat, latent heat to total heat, or sensible heat to total heat. The latter perhaps is the most common since it is a natural result of summarizing the heat gain estimate. This ratio is known as the *sensible heat factor* and is expressed in Equation 1 as:

$$\text{S.H.F.} = \frac{H_{sr}}{H_{sr} + H_{sl}} = \frac{H_{sr}}{H_{rt}} \qquad (1)$$

where

H_{sr} = room sensible heat, Btu per hour.
H_{sl} = room latent heat, Btu per hour.
H_{rt} = room total heat, Btu per hour.

If the sensible heat of the room or space is to be absorbed as it occurs within the space in order to maintain a given temperature, then the relationship between the quantity of air supplied, temperature of air supplied, room temperature and room sensible heat gain must be:

$$Q = \frac{H_{sr}}{(t_r - t_e)\, d \times c_p \times 60}$$

where

t_r = room dry-bulb temperature, degrees Fahrenheit.
t_e = supply air temperature (or apparatus dew-point temperature), degrees Fahrenheit.
c_p = specific heat of air (constant pressure).
d = density of air, lb per cubic foot.
H_{sr} = room sensible heat gain, Btu per hour.
Q = quantity of supply air, cubic feet per minute.

For standard air this Equation may be expressed:

$$Q = \frac{H_{sr}}{(t_r - t_e)\,1.08} \tag{2}$$

Similarly, if the room latent heat is to be absorbed by the supply air as it occurs within the space, then the relation between the supply air quantity, room air moisture content, supply air moisture content, and room latent heat content is as follows:

$$Q = \frac{h_{lr} \times 7000}{h_v \times d\,(g_r - g_e) \times 60}$$

where

h_{lr} = room latent heat, Btu per hour.
h_v = heat of vaporization, Btu per pound.
d = density of air, pounds per cubic foot.
g_r = room moisture content, grains per pound of dry air.
g_e = supply air moisture content, grains per pound of dry air.

For standard air this equation becomes:

$$Q = \frac{h_{lr}}{0.67\,(g_r - g_e)} \tag{3}$$

If a dehumidifier is used that will produce a saturated condition of leaving air, then Equations 2 and 3 can be solved for H_{sr} and h_{lr} and these substituted in Equation 1, resulting in Equation 4.

$$\text{S.H.F.} = \frac{0.241\,(t_r - t_e)}{(z_r - z_e)} \tag{4}$$

where

z_r = heat content of room air, Btu per pound.
z_e = heat content of supply air, Btu per pound.
t_r = room dry-bulb temperature, degrees Fahrenheit.
t_e = apparatus dew-point temperature, degrees Fahrenheit.

From Equation 4, it will be noted that only one temperature of supply air can be used to maintain the design room conditions if the supply air is saturated. This equation involving two unknowns requires a cut and try solution or graphical solution in order to obtain the apparatus dew-point. Several types of charts have been prepared to present a simple graphical solution. One of the forms of this chart is shown in Fig. 12. In addition to this, a psychrometric chart in which the dry-bulb temperature (sensible heat) and moisture content (latent heat) are represented by the abscissae and ordinates may be used since the slope of the

hypotenuse connecting the base of a triangle as represented by the temperature change, $t_r - t_e$ of the air supplied and the leg of the triangle as represented by the change in moisture content, $g_r - g_e$ of the supply air is a measure of the sensible heat factor. Such a chart with the sensible heat factor lines is shown in Fig. 13. The determination of the apparatus dew-point with such a chart is simple. A line, parallel to the proper sensible heat factor line, is drawn from the room or space condition through the saturation line as indicated and the point of intersection is the apparatus dew-point.

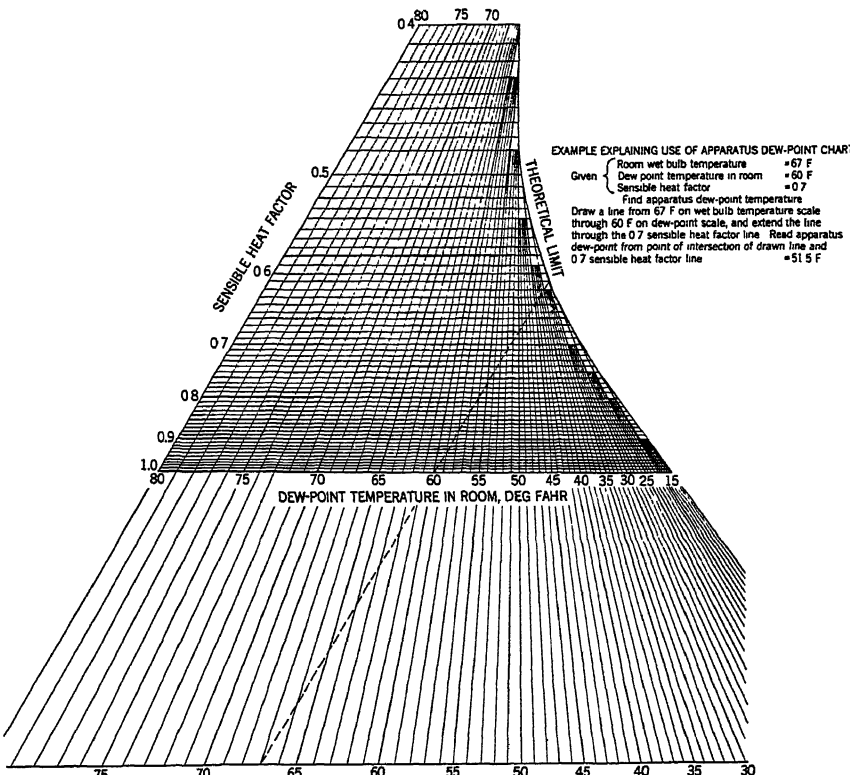

FIG. 12. APPARATUS DEW-POINT CHART

All of the foregoing is based on a dehumidifier that will produce a saturated condition of leaving air. As pointed out, this situation is infrequent due to the fact that in the majority of dehumidifiers some percentage of the air handled passes through the dehumidifier without coming in contact with the cooling surface or cooling medium. In the case where room air is recirculated through the dehumidifier or conditioner, the quantity of air that passes through without being contacted has no more effect on the heat balance than taking air from the room and reintroducing it to the room without treatment. On the other hand outdoor air that

is passed through the dehumidifier without being contacted by the surface or medium presents a definite sensible and latent cooling load which must be absorbed by the treated air. In accordance with these premises, when room air is bypassed or passed through the cooling surface or medium without being treated, then Equations 2 and 3 can be solved independently without a cut and try or graphical solution, if either the quantity of supply air has been established, or the temperature difference between the room and supply air temperature has been determined.

If the load created by the uncontacted outside air which passes through the dehumidifier is added to sensible and latent heat gain of the space in their proper proportions and the sensible heat factor corrected accordingly, then Equations 2, 3 and 4 will still apply if they are modified as described later.

Assuming that 20 per cent of the air handled by the dehumidifier passes through without contacting the cooling surface or medium, if this air is treated exactly as infiltration and the room sensible and latent heat gains increased accordingly, then Equations 2 and 3 may be modified to give:

$$Q = \frac{h_{sr}}{0.80\ (t_r - t_e)\ 1.08} \tag{2a}$$

$$Q = \frac{h_{sl}}{0.80\ (g_r - g_e)\ 0.67} \tag{3a}$$

Equation 4 will remain as before if the untreated air load is apportioned as indicated previously. It should be noted that when evaluating the load added by untreated outdoor air that the temperature difference and moisture content difference between room and outdoor air are to be used and not the difference between outdoor air and apparatus dew-point since the rise from apparatus dew-point to room conditions is charged against the dehumidifier as the cooling and dehumidifying load.

Where winter air conditioning is concerned, the determination of supply air temperature and supply air moisture content does not follow the exact procedure as outlined for summer cooling due to the reversal of requirements and other factors. Because of the nature of the loads and their magnitude and the equipment involved, the sensible and latent losses may be handled as if they were entirely separate in their occurrence since the air quantity required by the sensible heat losses will exceed that required by the moisture losses when related to reasonable and practical supply air temperatures and moisture content. Room sensible heat losses, supply air quantity, and supply air temperature will have the relationship (based on standard air as for Equation 2):

$$Q = \frac{h_{sr}}{(t_e - t_r)\ 1.08}$$

where

h_s = room sensible heat loss, Btu per hour.
t_e = supply air temperature, degrees Fahrenheit.
t_r = room temperature, degrees Fahrenheit.

Similarly, room latent heat losses or moisture losses and supply air moisture content have the following relationship (based on standard air and saturated conditions of supply air, as for Equation 3):

$$Q = \frac{h_{lr}}{0.67\,(g_e - g_r)} \text{ or } \frac{m_r}{(g_e - g_r)\,4.5}$$

where

h_{lr} = room latent loss, Btu per hour.
m_r = room moisture loss, grains per hour.
g_e = moisture content of entering air, grains per hour.
g_r = moisture content of room air, grains per pound.

As indicated for winter conditions the determination of the apparatus dew-point may resolve into a solution of Equation 3 following the solution of Equation 2.

Since reheaters are normally rated on average leaving air temperatures with the uncontacted air temperature taken into account Equation 2a usually will not apply.

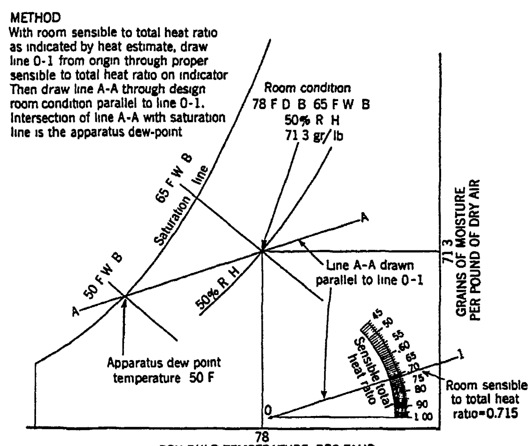

FIG. 13. USE OF PSYCHROMETRIC CHART TO DETERMINE APPARATUS DEW-POINT

If an air washer or humidifying sprays do not saturate, Equation 3a will be modified as (assuming 20 per cent of the air is not contacted by the sprays):

$$Q = \frac{h_r}{0.80\,(g_e - g_r)\,0.67} = \frac{m_r}{0.80\,(g_e - g_r)\,4.5}$$

In winter, room humidities in excess of 30 per cent are seldom required and a low saturating efficiency may be desirable or even necessary where the full summer circulation air through the conditioner is maintained. With a spray type dehumidifier the main sprays may be shut off and an eliminator flooding pump provided which may give sufficient saturation. In other cases, such as where cooling coils are sprayed, the spray water may be throttled.

If the saturating efficiency of the sprays is too low the spray water may be heated. The amount of heat put into the spray water by open or

closed water heaters will be equal to that required to bring the dew-point temperature of the air entering the sprays up to that required before entering the reheater. It is possible where clean steam is available, to introduce the steam directly into the air stream to produce the desired dew-point temperature of supply air. However, the steam must be exceptionally clean or objectionable odors will result. This precaution should be observed also where open water heaters or ejector water heaters are used.

Insofar as determining the apparatus dew-point and leaving air conditions for cooling coils, and spray dehumidifiers is concerned, particularly where saturation is not obtained, there are several methods other than those previously outlined which are more or less satisfactory. This discussion represents one of the more simple and accurate methods. Other methods are shown in Chapter 24.

Present day practice, for spray type dehumidifiers of good design, assumes that the air leaves the dehumidifier at 1 to 2 F higher than the temperature of the spray water leaving the dehumidifier. A spray dehumidifier having sufficient length of spray chamber and density of spray together with a proper arrangement of nozzles may closely approach absolute saturation.

Air Quantity and Effective Temperature Difference

The difference between the room air temperature and the supply air temperature at the outlet to the room is known as the *effective temperature difference*. In the theoretical case of a dehumidifier having a 100 per cent saturating efficiency and where this air is delivered directly to the room without temperature increases due to heat gain, then the effective temperature difference is the difference between room temperature and apparatus dew-point temperature. If duct heat gains are considered a part of the room load, this still holds true. The apparatus dew-point, as outlined previously, is fixed by the latent and sensible loads of the space, but in many cases, it is desirable to deliver more air to the spaces than is determined by the solution of Equations 2 and 3, with 4 or by the charts.

It has been indicated that where a percentage of air is passed through the dehumidifier without being treated that the relationship was modified in direct proportion, and that if room air passed through untreated no effect on the heat balance resulted. Similarly, if room air is passed around the dehumidifier and mixed with the treated air the heat balance is not adversely affected. Therefore, if the quantity of air passed through the dehumidifier is determined by the usual methods, room air can be passed around the dehumidifier and mixed with the dehumidified air, increasing the supply air quantity and temperature and decreasing the effective temperature difference. Thus if a solution of Equations 2 and 3 in conjunction with Equation 4 indicates that 10,000 cfm at 30 F below room temperature will be required to hold conditions, that quantity can be passed through the dehumidifier and cooled to 30 F below the room, then mixed with 10,000 cfm of room air resulting in a supply air quantity of 20,000 cfm and an effective temperature difference of 15 F instead of 30 F. Supply air outlets and grilles that have a high induction ratio (that is, a large amount of room air is mixed with the air leaving the outlet within

a short distance of the outlet through the induction effect of the air stream) are available as well as induction units. A proper selection of outlets or units may make it possible to introduce air at low temperatures and high velocities without causing objectionable drafts or cold spots, but care must be used to see that too little air motion is not a result. Lower effective temperature differences may be required for this reason. While the use of a high effective temperature difference results in a saving in initial cost of fans and ducts and in the operating cost of fans, this difference should be carefully considered. If the sensible heat load of a space is subjected to substantial variations lower effective temperature differences should be considered, since systems employing a low effective temperature difference will be less exacting in control requirements.

Considering a space having a sensible heat load such that with 10,000 cfm of air supplied at a 30 F effective temperature difference is required to maintain a room temperature of 80 F; and assuming that the load is suddenly reduced 50 per cent with the air supplied at the same temperature, the resultant room temperature would become 65 F. On the other hand, if 20,000 cfm were supplied at an effective temperature difference of 15 F and the load suddenly reduced 50 per cent the resultant room temperature would be 73.5 F. This, while a rather extreme example, indicates the less exacting demand on the controls brought about by the use of the lower effective temperature difference. Of even greater importance is the case where two or more spaces are controlled from an average condition such as by a thermostat located in the return air stream of all the spaces. From the previous example, it can be seen that a large variation in the load in one of the spaces will not reflect itself in such a large change in room temperature. Thus in the long run the larger effective temperature difference may not be the most economical.

The analysis in the foregoing applies largely to summer air conditioning. The same analysis will apply to some extent in winter. The mathematical relationship is revised due to a heating requirement rather than cooling. Present practice indicates a high temperature difference in winter in comparison to that for summer. It is due to the fact that the heat losses in winter in Btu per hour far exceed the summer heat gains in Btu per hour, and particularly to the fact that in winter, reheating the air to produce desired room conditions is not reflected as a load on the system as it is in the summer, but merely accomplishes the necessary work.

In line with the latter, reduction of air quantity by slowing down the fans for the winter season and increasing the temperature difference often is feasible, creating a saving in fan horsepower at no expense to the final heat balance, providing the air distribution is not seriously affected.

Extremes should be avoided in all cases. For summer air conditioning low supply air temperatures will result in larger heat gains to the air passing through the ducts, poor control, etc. Too high a supply air temperature may result in excessive initial and operating costs. Suggested limits for the effective temperature are from 12 to 25 F, the actual selection being based on the requirements of the particular case. For winter air conditioning too high supply air temperatures result in excessive heat losses from the ducts and stratification within the room unless thorough mixture is insured, while too low supply air temperature may cause drafts,

high operating costs, etc. Suggested limits are from 15 to 35 F. Basically there is no set rule and each case should be judged according to its particular requirements.

By-Pass

The by-pass, in its accepted form, consists of an arrangement of ducts and apparatus connections with the necessary dampers which will permit air to pass around the dehumidifier or conditioner without being treated. It has two functions which may be employed separately or simultaneously.

The first of these is to provide a means of temperature control at a substantially constant total air quantity. If in summer, the load within the conditioned space is reduced and the temperature begins to fall, this drop in temperature can be offset by passing some of the air around the conditioner instead of through it, while the total quantity of air in circulation remains unchanged. When used for this purpose, it is termed an *adjustable* or *automatic* by-pass. The second function is to maintain a lower effective temperature difference between the air supplied to the room and the room temperature than could be obtained if air at the apparatus dew-point were supplied, and when so used is called a *fixed by-pass*. As discussed previously, if return air from the conditioned space is passed around the conditioner and mixed with the conditioned air, the effect on the heat balance is the same as if the air were removed from the space and immediately reintroduced. This is not strictly true, due to the fact that when ducts pass through unconditioned spaces, there is a heat gain by this air, and an additional gain is imposed by the heat of compression of the circulating fan in moving the air against the resistance of the system. However, the heat gain, where the by-pass is used to lower the effective temperature difference, usually favors its use due to the fact that the increased volume and the resultant higher temperature of the mixture of conditioned air and room air may show a lower net duct heat gain with a smaller temperature increase per unit of volume of supply air. The advantages thus obtained may offset the increased fan power.

The adjustable or automatic by-pass can be made to serve the purpose of the fixed by-pass by providing a stop on the by-pass damper so that it can not close completely. In some cases this stop is unnecessary since commercial dampers will permit an air quantity of 10 to 20 per cent of the dehumidifier capacity, depending on the resistance of the conditioner, to leak through even when fully closed.

Where a reduction in room sensible heat is not accompanied by a reduction in room latent heat the by-pass is to be used with care. When the dew-point temperature of the air leaving the dehumidifier is controlled at a fixed value the reduced quantity passed through the conditioner may be sufficient to handle the sensible heat load but insufficient to handle the latent heat load, resulting in humidities that are too high. If the dew-point is not controlled, as when cold water is supplied at a constant temperature or where direct expansion cooling coils are used, the reduced loading on the conditioner brought about by the reduced air quantity, will result in a lower dew-point temperature which usually is entirely adequate. This condition should be investigated in each case.

The by-passing of outdoor air is to be avoided in general. While the sensible heat requirements of the space may be such that the by-passing of high temperature outdoor air will aid in controlling room temperature, high moisture content air introduced to the space may raise the humidity to an objectionable amount. If return ducts from which the by-pass air is taken run through unconditioned spaces and there is an inward leakage of outdoor or moist air, the effect, in a lesser degree, is that of by-passing outdoor air. Therefore the location of such return ducts and the points from which such air is taken is of importance. Exceptions to this are where the moisture content of the outdoor air is lower than that of the room, and where the sensible heat to latent ratio increases at partial loads.

The foregoing applies largely to summer air conditioning. In winter the by-pass usually is kept closed and room temperature control obtained by means of regulating the amount of heat supplied to the air. In some instances when the by-pass is located after the reheaters, the operation of the by-pass damper may be reversed and the by-pass still used as a means of control with the reheaters full on or operated in sequence with the by-pass. In other cases the reheater may be located in the by-pass as described further in this chapter.

The principle of the by-pass may be applied to items of equipment other than conditioners, such as humidifiers, dehydrators, heaters, etc., in much the same manner. Where a heater has a capacity such that the temperature rise of the air is higher than desired, a smaller heater may be used and a portion of the air by-passed around the heater. Throttling of the sprays in a humidifier is, in effect, a by-pass since it increases the portion of air passing through without being contacted.

Reheating

Reheating the supply air is necessary in winter where this air is used to offset heat losses. Tempering of the supply air (merely reheating to a lesser degree) is required where other means of heating, such as direct radiation or panel heating is used to carry the main heating load. Supply air at the required apparatus dew-point would add to the load to be carried by the direct radiation and in addition may create a movement of cold air that while desirable in summer may be undesirable in winter. Modulation of the amount of steam supplied to the coils can be used to control temperature, or air can be by-passed around these heating or tempering coils.

Since reheating or tempering coils are required for winter and year 'round air conditioning, their use as a means of summer as well as winter temperature control is indicated. Where heat is available in summer, as the room sensible heat falls off the low temperature of the supply air can be raised by means of these coils to maintain the desired room temperature while still providing adequate air at the proper dew-point.

Reheating presents an excellent method of accurate temperature control since the quantity of air passed through the conditioner is not changed as in the case of the by-pass, and since the distribution and circulation is not affected as when the volume of supply is reduced. However, reheating in summer has one disadvantage. It has the effect of maintaining a constant internal sensible heat load on the system. This means that when an

effective temperature difference of 15 F is being maintained at the maximum room sensible heat load, this temperature difference must be reduced by means of heating to 7.5 F at half of the sensible heat load. The room sensible heat usually is about 35 to 45 per cent of the total cooling load and thus the penalty imposed on the refrigeration cycle is not extremely large. When the outdoor wet-bulb temperature is less than the desired room wet-bulb temperature, all outdoor air should be passed through the conditioner and under these circumstances reheating does not impose a load on the refrigeration cycle since the heated air is not returned to the conditioner. Further, where the volume of outdoor air introduced to the system, which for all practical purposes is wasted after its work has been done, is sufficiently large in relation to the amount of reheat used (that is, if the heat required does not exceed that necessary to raise the outdoor air only from dew-point to room temperature) the use of reheat does not impose a load on the refrigeration cycle. This, of course, assumes that there is no economic penalty involved in the use of heat itself.

One of the best applications of reheating for summer purposes is in combination with other means of temperature control for the purpose of levelling off accentuated demands for temperature control. As an example, the by-pass can be applied to a certain extent, or the volume of supply air throttled to a limited degree, or both of these used in sequence, and reheating can be used as the final step in temperature control.

In the foregoing it has been assumed that summer reheating is derived from an extraneous source of heat such as steam, electric heaters or hot water. The economics of reheating as outlined may be improved by the use of sources of heat that are available within the equipment used.

Where certain types of refrigeration cycles are used, auxiliary refrigerant condensing coils can be placed in the air stream. At partial loads these can be used as additional refrigerant condensing surface and improve the performance of the refrigeration plant. This is generally known as hot gas reheating. Less effective is the use of coils in the air stream through which liquid refrigerant from the condenser is passed before being delivered to the evaporator, sub-cooling the liquid refrigerant and improving the performance of the refrigeration cycle. The use of refrigeration condenser water as a source of reheating provides definite economies. If the condenser water is passed through coils in the air stream before being used for refrigerant condensing purposes, the lowering of the condenser water temperature accomplished by reheating the air will result in savings in the refrigeration plant power consumption and increased refrigeration capacity. The latter two methods usually are at some disadvantage in that the amount of reheat available decreases as the need for reheating increases, particularly where evaporative condensers or cooling towers are used.

Zoning

Zoning consists of an arrangement of equipment or a division of equipment into sections that will permit individual control of the temperature and humidity of those spaces or groups of spaces that do not have simultaneous variations in sensible or latent heat load. The equipment is so arranged or divided that air can be supplied to spaces or groups of spaces

in accordance with the individual load requirements of that space or group of spaces.

Solar heat gain is one of the major causes of the zoning requirement since its effect and amount vary with the season, time of day and exposure. Other sources of heat gain, subject to variations, such as large changes in the number of occupants in one space with a constant occupancy in another indicate the necessity of zoning. Some of the various methods of zoning are:

1. Separate equipment.
2. Reheating or recooling. (See Figs. 4 and 5.)
3. Multiple fans with individual by-pass. (See Figs. 6 and 7.)
4. Volume control. (See Fig. 8.)
5. Dual duct system. (See Fig. 9.)
6. Combinations of the above methods.

Zoning by separate equipment represents the extreme in zoning. Individual conditioners, fans, heaters, controls, distributing duct work, etc., are provided for each zone and are separate from those of other zones. Each assembly of equipment is arranged to operate at full or partial load according to the requirements of that zone. In extreme cases an individual refrigeration plant may be provided for each zone. In general, this method of zoning is uneconomical since each piece of equipment is large enough to handle the full load requirements of that zone and no advantage can be taken of the fact that while one zone is at its full load others may be operating at considerably less than load capacity. This applies both to the initial cost of a system and the operating cost. There are, however, many cases where this method of zoning when used to a limited degree or combined with other methods is most desirable.

Zoning by reheat is one of the more simple methods of approaching the problem of differing variations in load. Reheating has been discussed in the foregoing. Heating coils located in the distributing duct work or apparatus connections which supply only those spaces having substantially the same load variations will, by adding heat when the cooling load falls off (or the reverse in the case of a heating load), maintain the desired temperature conditions. It has been shown previously that where the amount of reheating required is not excessive and that heat for this purpose is available from an economic standpoint this method of zoning may be entirely practical and even highly desirable.

Zoning by recooling is literally the reverse of zoning by reheating. It is not used often but there are many cases where it is desirable. Its most practical application is limited to a sensible heat removal function where the latent heat requirements are handled by another source such as a dehumidifier or dehydrator, and where the recooling equipment (usually cold water cooling coils) can be utilized for reheating (usually with hot water) in the heating season. In using this method of zoning, when the cooling load of a given space is reduced, the cooling effect produced by the recooler is also reduced.

The use of *multiple fans with individual by-passes* presents a simple and sometimes inexpensive method of zoning. Two or more fans may be arranged so that each draws its treated air from the same conditioner.

The connections between the conditioner and fans is divided or partitioned in such a manner that a by-pass connection can be made to each fan. See Fig. 6. In this way the amount of by-pass air can be regulated according to the requirements of the zone served by that fan. In this application a face damper must be used for each segment of the conditioner, arranged to close as the individual by-passes open or an unbalanced system may result. A much better adaptation of this principle, though slightly higher in initial cost, is obtained by using a single conditioner or dehumidifier, having a central conditioned air fan delivering the treated air to the necessary zone fans where the conditioned air is mixed with return air according to the requirements of the zone. See Fig. 7. With this method of zoning, the zone fans are provided with casings to which both return air and conditioned air are delivered, their proportions being regulated by dampers working in opposite directions. If conditioned air is delivered to the casing at slight positive pressure the return air damper may be omitted. Reheating can be effectively combined with either of these for year 'round use or for winter use only. The reheater is usually located in the air stream to the zone fan. Where a central conditioned air fan is used to deliver conditioned air to a number of zones a static pressure regulator controlling a volume damper or inlet vanes on this fan should be installed to prevent unbalancing the system when one or more zone fan conditioned air dampers are throttling. This is one of the better methods of zoning and is particularly effective when used in combination with reheating for winter or year 'round conditioning as previously outlined.

Volume control is the least expensive and most frequently used method of zoning. It is usually obtained by placing a throttling damper in the supply duct feeding a particular zone and operating the damper to restrict a flow of air as the heating or cooling load is reduced. The damper may be operated manually or automatically. Volume control, however, has two serious disadvantages. The first of these is that any large reduction in the quantity of air supplied may impair the ventilation. The second is that a large reduction of the air supply may entirely upset the distribution from the room outlets causing dead pockets, stratification, lack of air motion, or the reverse—undesirable drafts. Where the degree of volume control is large or the system extensive and where volume control is applied to one portion of a system and not to another, the use of a static pressure regulator controlling a fan discharge damper or fan inlet vanes is indicated. Its best application is in combination with some other method of zoning such as reheat or by-pass where it is used as one step in a control sequence, and the reduction in volume limited to a proper amount.

The *dual duct* method of zoning, sometimes referred to as the schoolhouse system, can be successfully applied to comfort air conditioning though certain precautions must be observed. Essentially this method employs a source of warm air and a source of cold air both of which are delivered to a common point where either or a mixture of both are delivered to a particular zone according to the requirements of that zone. See Fig. 9. Several variations of this method are possible, some of which do not employ dual ducts as denoted by the name but which utilize the principle. An example of this is found in a blow-through system where the fan is located on the entering side of the conditioner and the con-

ditioned air passes from the conditioner into a plenum from which distributing ducts for the various zones are taken. A by-pass connection around the conditioner from the fan to the zone duct is made and dampers provided so that conditioned or untreated air, or a mixture of both is passed into the zone supply duct. In this method of zoning and in most of the variations of this method, the matter of by-passing untreated or outdoor air presents itself. This has been discussed earlier in this chapter. If a return air fan is used and the by-pass connection made from the return fan to the zone supply duct, then the by-passing of outdoor air does not need to be considered. Complication of ducts and connections should be avoided since it may result in difficult sheet metal work with accompanying leakage of air and waste of cooling or heating effect.

Combinations of these several methods of zoning usually provide the most effective zoning. It is then possible to use each method to its greatest advantage without incurring operative or economic penalties which may be inflicted by the exclusive use of any one. Thus, where wide variations in load occur, volume control can be used to reduce the air quantity a limited amount, then as the load continues to fall off reheat or the by-pass or both can be used. By-pass and reheat can be used in series very effectively. Many combinations are possible and each case should be considered with regard to its particular requirements when deciding on the method of zoning.

Induction Units—Low pressure type

Induction units are essentially induction type convectors. These units utilize a jet of conditioned air (or primary air) to induce into the unit a flow of room or secondary air which mixes with the primary air. The mixture is discharged into the room through a grille at the top of the unit. Heating coils are located in the secondary air stream for use in heating. Control is obtained by either manually or automatically throttling the jet, and in addition, heat may be supplied to the secondary coils in summer as well as winter to provide control by reheating. The use of these induction units presents several advantages. Since the secondary air stream is thoroughly mixed with the high velocity low temperature air stream before leaving the discharge of the unit, the resultant temperature of the mixture is satisfactory even though the primary air is introduced at a temperature too low for ordinary methods of distribution. A unit is usually provided under each window in place of the customary direct radiation, and combines the air distribution system with the heating system. With a conventional system it may be necessary to provide supplementary heating in the form of direct radiation. These induction units may be selected so that their heating coils will have sufficient capacity under gravity conditions (that is, with the fan system off and no primary air entering the unit) to maintain the building or spaces at a reasonable temperature. The use of low temperature dehumidified air which has not been reheated or mixed with room air before delivery to the room results in a reduction in fan capacity and smaller sized duct work. In some cases the use of the by-pass may be desirable in order to keep the primary air volume up and provide additional control. This system can provide a degree of zoning that is usually impossible with conventional systems since each unit can be put under manual or automatic volume

control and reheat control. The selection of units should be made with regard to noise level when related to the noise level of the spaces. The inductive capacity of the unit increases with the jet velocity but too high jet velocities result in a high noise level.

Induction Units—High pressure type

A recent development of the induction unit as previously outlined is the high pressure type of unit. This unit employs nozzles which produce a high velocity jet quietly. The term, high pressure type of unit, is to some extent inaccurate since the pressure at the nozzles, while several times that of the low pressure unit, is still less than the total resistance pressure of a conventional central system. The high velocity jet of primary air induces a flow of secondary room air through coils located in the secondary air stream. The coil in the secondary air stream is supplied with chilled water in summer and hot water in winter and thus handles a large portion of the room sensible heat gain in summer and of the room sensible heat loss in winter. The primary air is supplied at a sufficiently low dew-point to take care of room latent heat gain in summer. In winter, it is supplied at a sufficiently high dew-point to take care of room latent heat losses. Control of temperature is obtained by throttling the water quantity supplied to the secondary coils. The quantity of primary air is greatly reduced due to the fact that a portion of the room sensible heat load is carried by the secondary air stream coil. Since the primary quantity is small, very high velocities can be carried in the supply ducts without requiring fan power in excess of that required for a conventional system. This means that the supply ducts or pipes can be very small and can be run in chases, or furred in at columns with the water pipes. The primary air is treated in the usual manner to provide air at the required dew-point and either a surface or spray dehumidifier or a dehydrator may be used. The primary air quantity is sufficient for ventilation purposes, and frequently consists entirely of outdoor air. The water piping for the units can be so arranged and valved that hot water can be supplied to one zone that may require heating while cold water may be supplied to a zone that requires cooling.

This system is usually limited in application to hotels, apartments, office buildings and other multi-room installations having a large perimeter with relation to the floor area. The units are usually installed beneath the windows, replacing direct radiation or convectors. Where the spaces to be conditioned extend a large distance from the outer wall into the interior of the building, a separate system or zone for the conditioning of the interior portions may be required.

Evaporative Cooling

In climates where on design maximum days, the outdoor wet-bulb depression is relatively high it may be possible to dispense with refrigeration or other cooling sources by use of the evaporative cooling effect. A well designed air washer using recirculating sprays will reduce the entering dry-bulb temperature to within a degree or two of the entering wet-bulb condition. Thus, it may be possible that with air entering at 100 F dry-bulb and 60F wet-bulb temperature a leaving condition of

62 F dry-bulb, nearly saturated can be obtained. Under some conditions of latent and sensible heat load the results may be entirely satisfactory.

Under those conditions when the outdoor wet-bulb temperature is not quite low enough to permit the use of straight evaporative cooling it is possible to use precooling coils with refrigeration, well water or a cooling tower as the basic source of cooling to lower the wet-bulb temperature (by sensible heat removal) of the air before it enters the air washer. Where internal heat loads are high, this may be more economical than using return air. Under other conditions where the required supply air dew-point is too low to permit straight evaporative cooling and the sensible heat load not too great, intentional partial saturation may be employed. That is, the low dew-point of the outdoor air is utilized by permitting some of it to pass through the humidifying sprays untreated or pass around the humidifier. All of these remarks with regard to evaporative cooling are based, as indicated, on the assumption that the supply air will consist entirely of outside air. Provision should be made for the return of air from the conditioned spaces for control purposes as well as for winter use in all cases.

Precooling

Where sufficiently cold water from wells or streams is available a saving in the refrigeration cycle may be obtained by the use of precooling. Cooling coils are placed ahead of the dehumidifier or conditioner and the cold water from a well or stream circulated through the coils. The resultant cooling of the air decreases the load to be carried by the dehumidifier and refrigeration plant. In normal practice the water after passing through the precooling coils is delivered to the refrigeration plant for condensing purposes. The economic advantages of this scheme are apparent and it is frequently used.

Sensible Cooling with Dry Cooling Coils

Under certain atmospheric conditions where a large wet-bulb depression exists and the dew-point of the outdoor air is sufficiently low at all times, proper inside conditions may be obtained by removing sensible heat only from the outdoor air and supplying it to the spaces. Under this condition of a high wet-bulb depression a cooling coil may be located in the air stream and this coil supplied with water from a cooling tower of one type or another. When humidity control is desired sprays to saturate or partially saturate the air may be used after the dry cooling coil. Saturation or partial saturation after the dry air cooler will further reduce the dry-bulb temperature of the supply air and reduce the supply air quantity required. This system has very definite application in hot dry climates and in general is most economical.

Run-Around System

An interesting method of control is found in the use of combined reheating and precooling usually termed the *run-around system.* Coils are placed in the air stream before and after the conditioner and water or brine is circulated around the conditioner from one coil to the other. The water passing through the reheating coil is cooled by the air leaving the dehumidifier and the air is heated by the water. The cooled water is then

circulated through the precooling coil where the entering air is cooled by the water and the heated water sent back to the reheating coil. The run-around has the advantage of permitting a higher supply air dew-point temperature than would be possible otherwise. This is due to the fact that continual reheating is available which is not a large penalty on the refrigeration plant since it provides precooling at the same time. This reheating at peak load creates an artificial sensible heat gain which increases the ratio of room sensible heat to room total heat and for a given room temperature results in a higher apparatus dew-point. See Equations 2, 3 and 4. Thus, while the volume of supply air is increased, the low side temperature level of the refrigeration plant is raised and this may effect savings in initial and operating costs. This system has the disadvantage of providing a decreasing amount of heat for reheating as the demand for reheating increases.

RELATION TO BUILDING TYPE

Few buildings or spaces are physically identical and those that are similar in this respect may have marked differences in internal loading, zoning requirements, and economic limitations. Consequently it is virtually impossible to establish fixed rules governing the type of system to be used. Each case must be considered on its own merits with due regard to all engineering and economic factors. However, some generalizations are possible.

In small single spaces the use of an elaborate system is undesirable from an initial cost standpoint. Zoning may be often eliminated. This also applies to large single spaces except that in very large spaces the necessity of providing adequate zoning is encountered more frequently. If the spaces are extremely large, physical and economic limitations such as the size of equipment, size and length of ducts, may require the division of the space into sections. Whether these sections are to be made according to zones or whether each section is to be zoned will depend on the particular case.

Where groups of spaces or small buildings are encountered, simple systems still prove the most economical. A single central station with zoning by means of volume control and reheat combined may be entirely satisfactory. If the perimeter of the building is large with regard to the area, induction units of the low or high pressure type may be employed, particularly if it is a multi-room application. Occasionally the dual duct system may be considered, but is infrequently used due to its complications.

Low buildings with large floor areas, such as large department stores and large general offices, may have to be divided into sections for treatment. In the case of large department stores it may be possible to provide a single conditioner with a fan delivering the conditioned air to recirculating fans which supply the various departments or spaces. This application is limited by practicability of running the large conditioned air ducts to the various recirculating fans. In some cases the use of separate systems for each section will be indicated with the added necessity of dividing into horizontal as well as vertical sections. If the latter is required, each vertical section may be handled by a separate system consisting of a single conditioner and fan delivering conditioned air to recircu-

lating fans supplying the horizontal sections. Zoning is obtained by proper allocation of the recirculating fans or other conventional methods used in conjunction with the recirculating fans. For general offices in particular, or types of buildings having a large perimeter, the use of one of the induction type units for perimeter treatment, combined with a conventional system for the treatment of the interior portions, offers possibilities.

High buildings having large floor areas may be successfully handled in many ways. Horizontal or vertical sectionalizing or both may be required, as determined by economic factors and physical limitations. Where vertical sections only are required the use of a single conditioner and fan for each section delivering conditioned air to zone fans at various floors may be used. These zone fans can be so arranged that one recirculating fan can handle similar zones on several floors, thus reducing the number of fans required and providing a degree of vertical zoning. Where vertical sectionalizing is not indicated, the building may be divided into horizontal groups, each group handled by a central system and adequately zoned. In some extremely large buildings apparatus rooms for the systems may be located in the basement and in the attic and on intermediate floors.

In high buildings having small floor areas the treatment required may be the same as for that of a vertical section of one having a large floor area. A single conditioner and fan can be used to deliver conditioned air to recirculating fans located at various floors.

In all cases of high buildings the necessity for horizontal sectionalizing is indicated by the economical size of air supply and return risers and by the extent to which they encroach upon usable space. In all buildings the necessity for vertical sectionalizing is indicated by the economical size of horizontal supply and return ducts and the space requirements of these ducts.

Usually the most simple systems are best adapted to theatres, auditoriums, and similar applications. Zoning is seldom required other than in connection with auxiliary spaces served by the same system. A conventional central system with by-pass control possibly augmented by reheat usually will suffice. The auxiliary spaces may be supplied with air from the same system controlled by volume reduction and reheat. Balconies and large lobbies frequently justify the use of separate zoning fans.

The foregoing are merely generalizations and suggestions. It is the responsibility of the engineer to explore thoroughly the possibilities of all types of systems and employ that best suited to the purpose from a standpoint of maintained economy, maintenance, life, operation and physical applicability.

EQUIPMENT SELECTION

Other chapters cover in detail most of the items of equipment used in a central system. Each item must be selected not only on its own merits, but in relation to all the other items that go to make up the complete central system. Each item should be considered from the standpoint of both initial cost and operating costs. Consideration must be given to performance at partial loads since most systems operate at full load but

a small percentage of time. Many items of equipment have been well standardized and are manufactured in certain definite sizes. The fullest advantage of this should be taken. One item that may be oversized of necessity may permit the use of a smaller piece elsewhere.

Fans operate at full capacity continually in many systems and therefore should be selected for good efficiencies. In winter where higher temperature differentials are used it is possible to use lower air quantitities and a two-speed motor may be provided for the fan, resulting in a power saving. In such cases the air distribution under the reduced volume should be investigated before providing this feature.

In selection of the dehumidifier or conditioner the relation of this item to the refrigeration plant is to be given careful consideration. Frequently, it is possible to make a saving in the refrigeration plant by providing more surface in the dehumidifier or conditioner. On the other hand, an excess of capacity in the refrigeration plant can be used to lower the apparatus dew-point (if the lower room humidity is satisfactory) resulting in a reduced quantity of dehumidified air and smaller dehumidifier.

In winter, where preheating coils in outside air intakes are required and subjected to entering air temperatures below freezing, the coils should be selected for operation at full capacity whenever the entering air temperature is below 35 F or where throttling of the steam supply to the coils is desirable, a type of coil that is designed especially for this must be used. In many cases the use of preheaters is not justified since the temperature of the mixture of outside and return air may be entirely satisfactory.

Where reheating coils are used after the supply fan, either as zone control reheaters or boosters, and are relatively near outlets, care is to be used to install these coils so that any stratification of temperature produced by the throttling of the steam or water supply does not result in cold air being delivered to one outlet and warm air to another. Some types of coils that do not produce stratification under throttled conditions are commercially available.

The selection of the refrigerating plant is in itself an economic study. The availability, consumption, and costs of condenser water are to be compared to the initial costs involved and water savings produced by the use of cooling towers or evaporative condensers. Whether or not a direct expansion or flooded system, cold water or brine type of plant will be used is not only a matter of initial cost but one of overall performance, operating economy, compactness, and in some cases, one of safety. Where water or brine is used as a cooling medium, the possibilities of using lower temperatures and decreased quantities of water or brine, with resultant savings in pumping power and line sizes, are to be considered with relation to the increased power consumption and probable increased cost of the refrigeration machine.

Air cleaning devices are to be selected according to the particular requirements of the project as well as to existing atmospheric conditions. In some applications such as certain types of stores or departments in large stores, lint screens should be provided for the return air as well as filters. Whether or not return air is to be filtered will depend upon the individual case and will be related to the amount of dirt or dust generated in or brought into the conditioned space from various sources. The type

of filter or cleaning device to be used depends largely on economic considerations. Obviously an expensive high efficiency cleaning device is not warranted where atmospheric dust or dirt is of such a nature that a less expensive, less efficient device will remove the more objectionable matter. Interior cleaning costs, dust and dirt damage, and hazard due to an accumulation of inflammable dust or dirt within the system are most important factors.

Automatic instruments are nearly always used in present practice for the control of temperature and humidity and to an increasing extent in the control of large refrigerating plants as well as small ones. Whether electric or pneumatic controls are to be used is a function of the particular requirements of the application and with certain exceptions a function of economy. Either electrically or pneumatically operated controls can be made to serve the same purpose though the individual case may favor one or the other. A detailed discussion of automatic controls may be found in Chapter 34. One point is to be emphasized. In general, the simpler the control system the better it will perform. Control systems seldom receive the maintenance they deserve and the fewer the instruments and devices the better the care. Further, it is poor policy to provide, at additional expense, instruments of extreme sensitivity to devices incapable of responding to the control demands.

Insulation is an important factor in air conditioning systems. Its economics with regard to steam and water or brine piping are well known and need no comment. The insulation of duct work is not merely a matter of economics but sometimes is a necessity from the standpoint of limiting the temperature rise of the air even when the ducts are in conditioned spaces. This temperature rise of the air should always be taken into account when apportioning the air and sizing the ducts and will indicate the necessity for insulation. In some cases, leakage of air from the duct where the duct is in a furred space or chase will eliminate the need for insulation by maintaining a reasonable temperature surrounding the duct.

There are practically no items of equipment associated with central air conditioning systems that are not subject to economic limitations as well as those of performance and duty and all of them should be considered in the selection.

ARRANGEMENT OF EQUIPMENT

A proper arrangement of equipment is essential for the proper functioning of any system. Where systems are to be installed in existing buildings the arrangement of equipment may be limited by structural or space considerations, but no compromises that may prevent satisfactory operation or maintenance should be considered.

The location of the apparatus room is often determined by building construction or available space. The closer the apparatus room to the conditioned space, the less expensive is the duct-work. On the other hand if the equipment generates noises it may be necessary to locate the room some distance from the spaces or provide adequate sound and vibration treatment. The scattering of wet apparatus throughout a building is to be avoided unless suitable precautions are taken. It must

be remembered that encroachment on spaces that are otherwise usable can be charged against the system as an operating cost.

In general the apparatus should be arranged to have straight line air flow. This is desirable but not always possible. Each change in direction is the source of air resistance, and in addition may cause eddy currents resulting in stratification. The usual order of equipment, beginning at the outside air intake is: outside air screen, outside air louvers, maximum and minimum outside air dampers, preheaters, return air connection, filters, conditioner, by-pass connection with or without reheaters, reheaters, fan and distributing ductwork. See Figs. 1, 2 and 3 for typical arrangements. Use of one or more of the methods of zoning may require a modification of this order but usually only after the dehumidifier or conditioner.

Outside air screens prevent the entry of large foreign matter, birds, etc. The use of louvers or a hood at the outside air intake prevents the entry of rain and snow. Both of these should be used on all systems. As pointed out earlier, the louvers and screens should be of sufficient size to permit the passage of the entire conditioned air quantity.

The minimum outside air damper usually covers the entire face of the preheater, which in turn is selected for the minimum outside air quantity. The maximum outside air damper is designed for the difference between the dehumidified air quantity and the minimum outside air. Its size can be such that its air resistance when open will equal the air resistance of the open minimum outside air damper plus that of the preheater if used. Where the spaces conditioned are very tight against air leakage, some type of relief or positive exhaust may be necessary when all outdoor air is introduced to the system to provide some means of egress for the air. Such reliefs require back-draft dampers to prevent infiltration when all outdoor air is not being used. A return fan properly dampered as indicated later is sometimes used.

The return air from the conditioned spaces usually is brought into the apparatus between the preheater and the filter. Just as it is necessary to make provision for using all outdoor air it is necessary to make provision for using all return air. Heating an unoccupied building or cooling it after a shut-down is easier if all the air used is return air. Consequently the return connections should be ample for this. In many cases higher velocities can be carried in the return ducts during such times, since the fan will exert a pull at the return connection nearly equal to the resistance of the outside air connection and the return ducts need not be larger than for normal return air quantities at normal velocities. In other cases it is possible that a reduced quantity of air due to the increased resistance of the return duct system may be satisfactory during the starting period. Where the return air system is extensive or complicated a return air fan is desirable. This fan can serve as a combination exhaust and return fan by arranging dampers in its discharge so that the necessary return air can be delivered back into the system and the remainder discharged outdoors. When all outdoor air is passed through the conditioner the entire return air quantity is discharged outdoors.

The by-pass connection normally connects the return air duct system into the apparatus casing between the conditioner and fan. Usually it is sized to handle about 50 per cent of the fan capacity where a variable

by-pass is used though extreme load variations may require a greater amount. It is at times good design to locate the reheater in the by-pass connection using a certain amount of by-pass air when heating is required. Since the relatively high resistance of the conditioner is to be balanced by the heating coil and by-pass connection, enough heating surface can be provided to raise the temperature of the by-pass air to the point where the mixture of by-pass air and conditioned air will have the required temperature. When a variable by-pass is used a damper working in opposition to the by-pass damper should be placed across the face of the dehumidifier, for unless the resistances of the two are most carefully balanced at all operating points the proper mixtures of air will not be obtained. The avoidance of by-passing outside air is again stressed. Where the by-pass is made a part of the dehumidifier or conditioner and located on the top or side of it, the return air connection should be made in such a way that stratification of return air is insured, baffles being provided to accomplish this purpose if necessary. Where return air and by-pass air connections are taken off a return duct system it may be necessary to install a back-draft damper between the return air connection and the by-pass connection, if the return duct system is extensive and the connections simple. In this instance, when the by-pass damper is at maximum opening it may be much easier for outside air to pass through the return damper, into the return duct connection and through the by-pass than for return air to pass through the by-pass connection into the fan. Air always takes the easiest path and if the dehumidifier resistance is high, and the return duct resistances low, this situation is apt to occur unless precautions are taken. A return fan instead of a back-draft damper may be required for this case if the failure of return air to reach the dehumidifier or conditioner is a serious matter under reduced load conditions.

The location and arrangement of the dehumidifier, humidifier, or conditioner with reference to each apparatus assembly is more or less standardized. In general, the outside air intake, preheaters, and return air connections precede the conditioner while the by-pass, reheaters and fan follow the dehumidifier. In the cases of the *blow through* system, where the fan is located ahead of the conditioner, the leakage of air at the conditioner is outward instead of inward and may be accompanied by water leakage unless the proper precautions are taken.

The location of the complete apparatus assembly including the dehumidifier will be dependent on the type of building, spaces available, structural characteristics, etc. The type of conditioner used may limit the location under certain conditions. Where cooling coils employing chilled water or brine as the cooling agent are used there are few restrictions with regard to location other than those of pumping power, working pressures, line costs, etc. Where open spray dehumidifiers are used very definite limitations present themselves, and these may require certain extraneous equipment to make the system workable. If several spray type dehumidifiers are located on different levels, a surge or storage tank to which the return water from each dehumidifier can be taken is required. Should the water level in the pan of the dehumidifiers be low in relation to that of the surge tank, return water pumps will be required, and these pumps will have to be operated until the water supply lines are drained in

order to prevent flooding of the lower dehumidifiers. Where spray dehumidifiers are on the same level, equalizing lines between the pans may be required if a storage tank is not provided.

All of the various pieces of equipment from the outdoor air intake through the fan usually are connected together by sheet metal casings. Frequently the building structure or specially constructed walls or partitions may be used to form all or portion of the casing. In any case the casing or connection must be sufficiently sturdy for the required duty. Sheet metal work must be well braced not only to prevent bellying or vibration under pulsations in air flow but to withstand the abuse of normal usage. Casings should be adequately braced wherever access doors are installed and all large panels should be adequately reinforced by angle iron.

Each apparatus layout is to be made with accessibility in mind. Where cooling coils are used space for removing and repairing or replacing the coils should be provided. Adequate space is to be provided for the servicing and replacement of eliminators. Filters must be so located that the proper cleaning, replacement or routine servicing can be accomplished without difficulty. Free access to the bearings of all moving machinery is a necessity. Provisions should be made for the complete removal and replacement of any part of the system that is subject to wear, deterioration or damage, whether it may be filter, fan wheel, motor, rotor, pump impeller or heat transfer surface.

DESIGN PROCEDURE

The customary design procedure is outlined herewith. For simplification the procedure is set up on the basis of a year 'round system. For summer only or winter only systems, the unrelated parts are to be omitted.

1. Selection of design conditions (inside and outside).
   a. Summer.
   b. Winter.
2. Determination of outside air requirements.
3. Determination of cooling load.
   a. Room sensible heat gain.
   b. Room latent heat gain.
   c. Room total heat gain.
   d. Grand total heat gain.
4. Determination of heating load.
   a. Room sensible heat loss.
   b. Room moisture loss.
   c. Humidification requirement.
   d. Total heating requirement.
5. Determination of apparatus dew-point and dehumidified or humidified air quantity.
   a. Summer (full load and part load).
   b. Winter.
6. Supply air temperature difference and quantity.
   a. Summer.
   b. Winter.
7. Equipment selection.
8. Equipment layout.

The foregoing steps are merely typical. Many applications will require at least a preliminary investigation of some of the latter steps before proceeding with the earlier steps.

Chapter 22

UNIT HEATERS, UNIT VENTILATORS, UNIT HUMIDIFIERS

Unit Heaters, Ratings, Unit Ventilators, Applications, Window Ventilators, Unit Humidifiers, Types of Units

IN other chapters, descriptions are given of heating, cooling, ventilating humidifying, and dehumidifying systems. The success of such systems has led to the production of factory-assembled equipment employing a majority of the principles of these complete systems. As a result, present day practice involves the use of unitary equipment in the majority of installations where capacity and application demands are within the limits of such units. Thus unit heaters, unit ventilators, and unit humidifiers described in this chapter, and cooling units, unit air conditioners, and attic fans described in Chapter 23 have come to occupy a place of their own in the industry.

Unitary Equipment

Unitary equipment was first applied in units of small capacity but increased experience in this field has led to an ever widening range of capacities and applications. In general a *unit* may be defined as a factory-made encased assembly of the functional elements indicated by its name, such as unit heater, unit ventilator, etc. These units are shipped substantially complete or built and shipped in sections so that the only field work necessary is the assembling together of the sections, without resorting to any field fabrication.

A unit may be complete in itself, employing its own direct means of air distribution and source of heating, in which case it thus represents a complete self-contained unit. Or it may be coupled with separate means of air distribution such as duct work and outlets, in which case it will still be considered as a unit system, as contrasted with the generally accepted term of a field fabricated central station system. The manufacturer of the unit is responsible for the output and performance of the unit under rated conditions, whereas the contractor installing the complete unitary system is normally held responsible for the performance of the complete system.

Unit equipment justifies its existence due to the following features:

1. Lower cost per unit capacity. Standardized design and volume production makes possible low cost factory assembly thereby eliminating individual design and handling of every part for each installation.

2. Flexibility and mobility of equipment. Unitary equipment can be readily located in existing buildings without the necessity of running large ducts through floors and many partitions. Such equipment can be shifted to meet changing requirements. Tenants may obtain the advantages of conditioning when the entire building is not equipped with a conditioning system. In industrial process work, the flexibility of unitary equipment is also advantageous.

3. Lower installation costs. The fact that the equipment arrives on the job in an assembled condition, coupled with the lesser problems of duct work and connecting piping, materially reduces installation costs

4. Small capacities. The small capacities available in unitary equipment have brought the advantages of controlled air conditions to a number of small offices, stores, shops, and individual rooms where specially designed and built central system equipment would have been uneconomic.

Definitions

With the growth of the unit equipment industry, it becomes increasingly evident that there is no sharp line of demarcation, on the basis of capacity, between a unit and a central plant system. The definitions contained in a Code, Standard Method of Rating and Testing Air Conditioning Equipment[1], have helped to clarify and identify the various types of equipment since the definitions are given on a purely functional basis. The following definitions are taken from the code:

1. A *Heating Unit* is a specific air treating combination consisting of means for air circulation and heating within prescribed temperature limits.

2. A *Heating Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for heating and maintaining humidity within prescribed limits.

3. A *Humidifying Unit* adds water vapor to and circulates air in a space to be humidified.

4. A *Free Delivery Type Unit* takes in air and discharges it directly to the space to be treated without external elements which impose air resistance.

5. A *Pressure Type Unit* is for use with one or more external elements which impose air resistance.

UNIT HEATERS

A unit heater consists of the combination of a heating element and fan or blower having a common enclosure and placed within or adjacent to the space to be heated. Generally no ducts are attached to inlets or outlets, although it is common practice with many unit heater applications to equip the heaters with directional outlets or adjustable louvers. While unit heaters are designed primarily to handle all recirculated air, they may be installed to handle either partial or total outdoor air.

Features

A wide variety of structural designs is available. All employ some form of heat transfer surface, supplied with steam, hot water, gas or electric heat. Air is always forced over or drawn through the heat transfer surface by a fan of either the propeller or centrifugal type. Heating surfaces may be in the form of non-ferrous or steel pipe coils, non-ferrous or steel pipe with extended surfaces, cast-iron, or pressed or built-up sections of the cartridge or automotive type.

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[1]Prepared by a Joint Committee of the *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association*, *National Electrical Manufacturers' Association* and *Air Conditioning Manufacturers' Association*.

Compared with the older method of heating by radiation, properly designed and applied unit heaters should:

1. Circulate air in the building at a rapid rate but without objectionable draft.
2. Reduce the temperature differential between the floor and ceiling.
3. Direct the heated air so that uniform temperature distribution will be obtained throughout the heated space.
4. Prevent or remove the cold stratum of air commonly found at the floor level.
5. Reduce the number of heating elements required and thereby decrease the cost and extent of the piping necessary.
6. Maintain a closer control of room temperature either manually or by means of simple thermostats.
7. Produce an economy in heating costs resulting from the sum total of the above advantages.
8. Provide a means of saving floor area or room space due to the compactness of the equipment and flexibility of application.

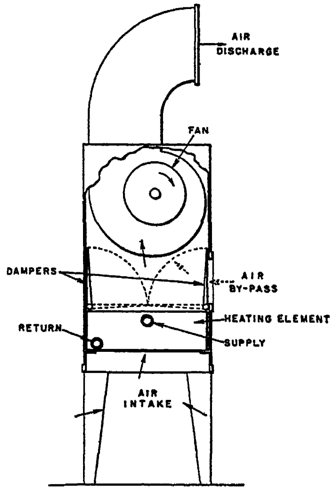

FIG. 1. FLOOR MOUNTED UNIT HEATER, HOUSED TYPE FAN

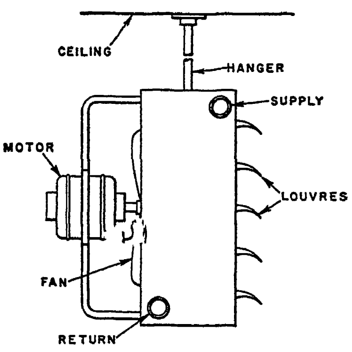

FIG. 2. SUSPENDED UNIT HEATER, PROPELLER TYPE FAN

Types of Units

There are two major types of unit heaters, centrifugal housed fan type and propeller fan type. The housed fan high velocity (1500 to 2500 fpm) discharge units, with outlets adjustable to deliver air in several directions, are able to project their heating effect over distances of from 30 ft to as much as 200 ft from the unit. This makes possible the location of these units at considerable distances from each other, thus reducing greatly the piping and loss of floor space due to the heating equipment. Figs. 1 and 3 illustrate the housed fan type of unit.

Propeller fan type of unit heater is used extensively to heat the small commercial establishment, although this type of unit is also available in very large sizes. Fig. 2 illustrates a propeller fan type of heater. A code[2] governing the number of sizes of propeller fan type units as well as

[2]Standards for Propeller Type Unit Heaters prepared and adopted by the *Industrial Unit Heater Association*, June, 1938.

standardization of fan motor types and the method of specifying outlet velocities has been adopted.

Ratings

Standard practice is to rate unit heaters in Btu per hour at a given temperature of air entering the heater and at a given steam pressure maintained in the coil. Steam at 2 lb pressure and air entering at 60 F are used as the standard basis of rating[3]. The capacity of a heater increases as the steam pressure increases, and decreases as the entering air temperature increases. The heating capacity for any condition of steam pressure and entering air temperature may be calculated approximately from any given rating by the use of factors in Tables 1 and 2. Table 1 is used for the blow-through type and Table 2 for the draw-through type of unit. The formulae given under *unit ventilators* for calculating capacities also apply to unit heaters.

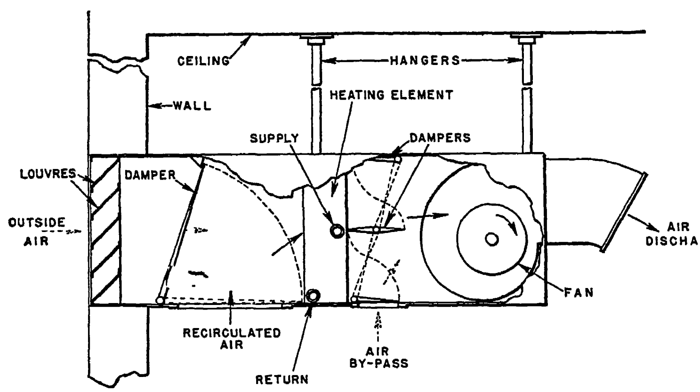

FIG. 3. SUSPENDED TYPE UNIT HEATER, HOUSED TYPE FAN

The temperature to be maintained[4] in the room, for recirculating heaters with intakes at the floor level, should be considered as the temperature of the air entering the heater. Where outside air is introduced, the temperature of the mixture must be calculated and used as the entering air temperature to the heater. Unit heaters taking in recirculated air at the floor level should maintain temperature differentials of less than 0.5 deg per foot of elevation when the maximum capacity of the heaters is required. This temperature difference per foot of elevation is less than the corresponding variations for spaces heated by direct radiation.

The temperature variation from floor to ceiling with suspended unit heaters taking air at some distance above the floor may reach as much as 1 deg per foot of elevation during the periods when the maximum capacity of the heaters is required. Thus this allowance should be made in calcu-

[3]A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Heaters (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 165).

[4]A.S.H.V.E. RESEARCH REPORT No. 958—Temperature Gradient Observations in a Large Heated Space by G. L. Larson, D. W. Nelson, and O. C. Cromer (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 243).

A.S.H.V.E. RESEARCH REPORT No. 1011—Tests of Three Heating Systems in an Industrial Type of Building, by G. L. Larson, D. W. Nelson, and John James (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 185).

lating the capacity of suspended heaters. High velocity discharge units (blower type illustrated in Fig. 3) will maintain slightly lower temperature differences than will low velocity units (propeller fan type illustrated in Fig. 2).

Unit heaters are customarily rated as free delivery type units. If outside air intakes, filters, or ducts on the discharge side are used with the heater, proper consideration should be given to the reduction in air and heating capacity that will result because of this added resistance.

The percentage of this reduction in capacity will depend upon the charactertistics of the heater and on the type, design and speed of the fans so that no specific percentage reduction can be assigned for all heaters at a given added resistance. In general, however, disc or propeller fan type units will experience a larger reduction in capacity than housed centrifugal fan units for a given added resistance and a given heater will have a larger reduction in capacity as the fan speed is lowered. When confronted with this problem the ratings under the conditions expected should be secured from the manufacturer.

Boiler Capacity

The capacity of the boiler should be based on the rated capacity of the heaters at the lowest entering air temperature that will occur, plus an allowance for line losses. Ordinarily for recirculating heaters the lowest entering temperature will occur at the beginning of the heating period and is usually taken as 40 F, while for ventilators taking air from outdoors the lowest entering temperature will be the extreme outdoor temperature expected in the district. No greater allowance in boiler capacity beyond the calculated heat demand need be added in order to supply unit heaters than for any other type of system.

It is unwise to install a single unit heater as the sole load on any boiler, particularly if the unit heater motor is started and stopped by thermostatic control. The wide and sudden fluctuations of load that occur under such conditions would require closer attendance to the boiler than is usually possible in a small installation. Where oil or gas is used to fire the boiler, it is possible by means of a pressurestat to control the boiler, in response to this rapid fluctuation. In most cases, however, and particularly where the boiler is coal-fired, it is advisable to use two or more smaller heating units instead of one large unit.

Steam pressures below 5 lb can be used with safety for recirculating unit heaters when their coils are designed for the purpose and when proper provision is made for returning the condensate. If ventilators are to take in air that may be at a temperature below freezing, however, a steam pressure of not less than 5 lb should be maintained on the convector or a corresponding differential in pressure between the supply and returns be maintained by means of a vacuum.

Piping Connections

Piping connections for unit heaters are similar to those for other types of fan blast heaters. The piping around the unit heaters must strictly conform to the system requirements while at the same time permitting the heaters themselves to function as intended. The basic piping principles for steam systems are discussed in Chapter 15.

TABLE 1. CONSTANTS FOR DETERMINING THE CAPACITY OF **Blow**-THROUGH TYPE UNIT HEATERS FOR VARIOUS STEAM PRESSURES AND TEMPERATURES OF ENTERING AIR

(Based on Steam Pressure of 2 lb Gage and Entering Air Temperature of 60 F)

Steam Pressure Lb per Sq In	Temperature of Entering Air											
	−10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
0	1.538	1.446	1.369	1.273	1.191	1.110	1.034	0.956	0.881	0.809	0.739	0.671
2	1.585	1.495	1.405	1.320	1.237	1.155	1.078	1.000	0.926	0.853	0.782	0.713
5	1.640	1.550	1.456	1.370	1.289	1.206	1.127	1.050	0.974	0.901	0.829	0.760
10	1.730	1.639	1.545	1.460	1.375	1.290	1.211	1.131	1.056	0.982	0.908	0.838
15	1.799	1.708	1.614	1.525	1.441	1.335	1.275	1.194	1.117	1.043	0.970	0.897
20	1.861	1.769	1.675	1.584	1.498	1.416	1.333	1.251	1.174	1.097	1.021	0.952
30	1.966	1.871	1.775	1.684	1.597	1.509	1.429	1.346	1.266	1.190	1.115	1.042
40	2.058	1.959	1.862	1.771	1.683	1.596	1.511	1.430	1.349	1.270	1.194	1.119
50	2.134	2.035	1.936	1.845	1.755	1.666	1.582	1.498	1.416	1.338	1.262	1.187
60	2.196	2.094	1.997	1.902	1.811	1.725	1.640	1.555	1.472	1.393	1.314	1.239
70	2.256	2.157	2.057	1.961	1.872	1.782	1.696	1.610	1.527	1.447	1.368	1.293
75	2.283	2.183	2.085	1.990	1.896	1.808	1.721	1.635	1.552	1.472	1.392	1.316
80	2.312	2.211	2.112	2.015	1.925	1.836	1.748	1.660	1.577	1.497	1.418	1.342
90	2.361	2.258	2.159	2.063	1.968	1.880	1.792	1.705	1.621	1.541	1.461	1.383
100	2.409	2.307	2.204	2.108	2.015	1.927	1.836	1.749	1.663	1.581	1.502	1.424

Note: To determine capacity at any steam pressure and entering temperature, multiply constant from table by rated capacity at 60 F entering and 2 lb pressure.

TABLE 2. CONSTANTS FOR DETERMINING THE CAPACITY OF **Draw**-THROUGH TYPE UNIT HEATERS FOR VARIOUS STEAM PRESSURES AND TEMPERATURES OF ENTERING AIR

(Based on Steam Pressure of 2 lb Gage and Entering Air Temperature of 60 F)

STEAM PRESSURE LB PER SQ IN.	TEMPERATURE OF ENTERING AIR											
	−10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
0	1.483	1.405	1.329	1.253	1.178	1.105	1.032	0.962	0.892	0.822	0.754	0.688
2	1.520	1.442	1.363	1.290	1.215	1.141	1.069	1.000	0.930	0.861	0.792	0.728
5	1.565	1.485	1.410	1.334	1.260	1.187	1.114	1.045	0.975	0.906	0.838	0.771
10	1.637	1.558	1.480	1.403	1.328	1.253	1.182	1.112	1.042	0.973	0.903	0.838
15	1.688	1.610	1.533	1.458	1.382	1.310	1.239	1.168	1.099	1.028	0.960	0.895
20	1.728	1.649	1.572	1.498	1.421	1.350	1.278	1.208	1.138	1.070	1.002	0.936
30	1.803	1.725	1.648	1.572	1.497	1.423	1.352	1.281	1.212	1.145	1 078	1.010
40	1.864	1.787	1.710	1.637	1.563	1.491	1.420	1.350	1.282	1.215	1.148	1.081
50	1.927	1.850	1.773	1.700	1.628	1.554	1.483	1.416	1.347	1.278	1.211	1.145
60	1.973	1.897	1.820	1.748	1.673	1.601	1.531	1.463	1.394	1.325	1.260	1.194
70	2.018	1.943	1.869	1.795	1.722	1.651	1.582	1.512	1.443	1.377	1.310	1.243
75	2.043	1.970	1.895	1.822	1.750	1.680	1.609	1.540	1.471	1.402	1.333	1.268
80	2.064	1.988	1.914	1.841	1.770	1.698	1.629	1.560	1.491	1.422	1.354	1.288
90	2.102	2.028	1.951	1.878	1.804	1.732	1.661	1.590	1.523	1.457	1.387	1.321
100	2.150	2.071	1.994	1.919	1.845	1.770	1.700	1.630	1.560	1.492	1.425	1.359

Note: To determine capacity at any steam pressure and entering temperature, multiply constant from table by rated capacity at 60 F entering and 2 lb pressure.

Rapid condensation of steam, especially during heating-up periods, is characteristic of this type of equipment. The piping must be planned to accommodate this rapid condensation, must keep the surfaces free of water; while on the supply side the piping must be ample to carry a full supply of steam to the surfaces to take the place of that condensed.

Adequate size of pipe is thus essential to all heating surfaces over which there is a forced flow of air. Especially is this true where the fan is operated under start-and-stop control and where the air handled may be made up either wholly or partly of cold air from outside the building. In such installations the condensation rate may vary rapidly and the necessity for ample pipe capacity is especially acute.

A method of connecting a unit heater to a one-pipe gravity system is illustrated in Fig. 4. In those cases where the unit heater is to be con-

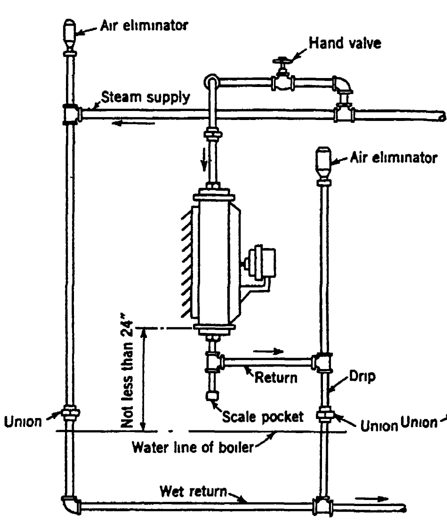

FIG. 4. UNIT HEATER CONNECTION TO ONE-PIPE GRAVITY STEAM SYSTEM

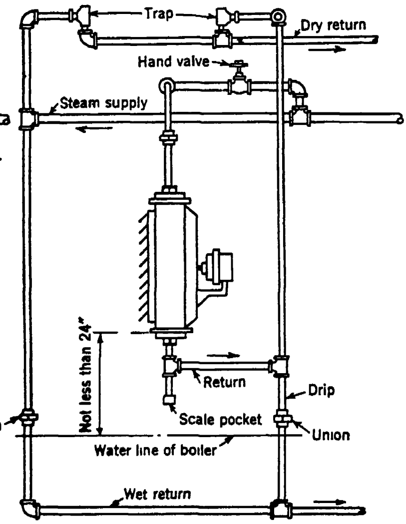

FIG. 5. UNIT HEATER CONNECTION TO GRAVITY SYSTEM WITH WET AND DRY RETURNS

nected to a dry return instead of a wet return it is necessary to provide a water pocket or loop about 5 ft in depth to prevent steam passing into the return and thus into other equipment.

A method of connection is shown in Fig. 5, where there is a wet return and a dry return. In this case the condensate from the heater and the drip from the supply main drop to the wet return by gravity, while the air passes upward through the traps to the dry return and is vented from the system at any suitable location.

A sketch of an arrangement where there is a dry return line through which both air and condensate pass to be handled by some suitable means, such as a condensate pump and receiver is given in Fig. 6. The return line is not subjected to vacuum, and consequently all arrangements must facilitate gravity flow of the condensate toward the receiver. Traps must pass air and condensate rapidly to keep the return piping only partially full of water.

Since unit heaters are often constructed with sufficient strength to resist high pressures, use of high pressure steam in them is a common practice. In Fig. 7 the condensate and air reach the return overhead through traps, and check valves are located in the return piping.

For two-pipe closed gravity return systems, the return from each unit should be fitted with a heavy duty or blast trap, and an automatic air valve should be connected into the return header of each unit. Pressure drop must be compensated for by elevation of the heater above the water line of the boiler or of the receiver.

In pump and receiver systems the air may be eliminated by individual air valves on the heaters, or it may be carried into the returns the same as for vacuum systems and the entire return system be free-vented to the atmosphere, provided all units, drip points, and radiation are properly trapped to prevent steam entering the returns.

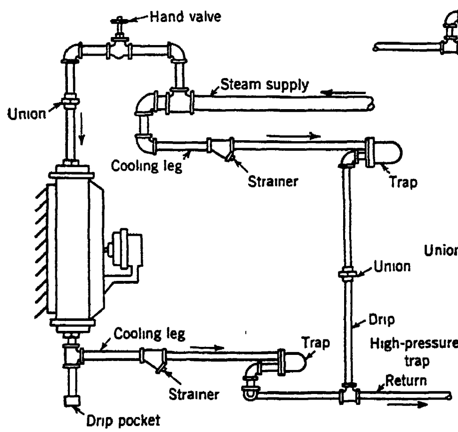

FIG. 6. UNIT HEATER CONNECTION FOR VACUUM OR VAPOR SYSTEM DISCHARGING CONDENSATION INTO DRY RETURN

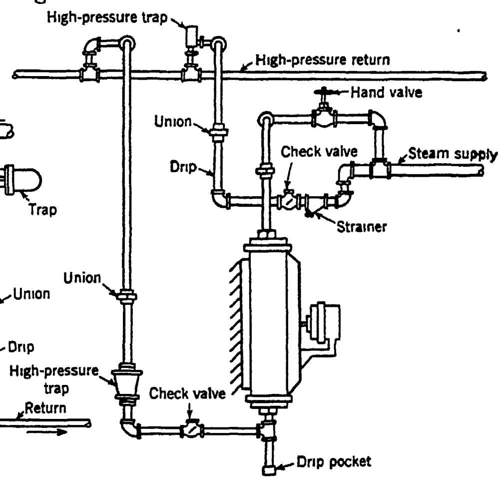

FIG. 7. METHOD OF CONNECTING UNIT HEATER TO HIGH PRESSURE RETURN

On vacuum or open vented systems the return from each unit should be fitted with a large capacity trap to discharge the water of condensation and with a thermostatic air valve for eliminating the air, or with a heavy-duty trap for handling both the condensation and the air, provided the air finally can be eliminated at some other point in the return system.

For high pressure systems the same kind of traps may be used as with vacuum systems, except that they must be constructed for the pressure used. If the air is to be eliminated at the return header of the unit, a high pressure air valve can be used; otherwise the air may be passed with the condensate through the high-pressure return trap, with some danger of return pipe corrosion and the problem of its elimination at some other point in the system.

Application

Unit heaters are used principally for commercial and industrial applications such as display rooms, garages, factories, factory offices and to

some extent for office applications where appearance is not a major factor.

Unit heaters may be adapted to a number of industrial processes, such as drying and curing, with which the use of heated air in rapid circulation with uniform distribution is of particular advantage. They may be used for moisture absorption, such as fog removal in dye houses, or for the prevention of condensation on ceilings or other cold surfaces of buildings in which process moisture is given off. When such conditions are severe, it is necessary that the heaters draw air from outside in enough volume to provide a rapid air change and that they operate in conjunction with ventilators or fans for exhausting the moisture-laden air. (See discussion of Condensation in Chapter 4.)

There are three major factors to consider in the application of unit heaters, as follows:

1. Location of Unit.
2. Air Distribution.
3. Heating Medium.

Heaters may be distributed through the central portions of a room discharging toward exposed surfaces, or may be spaced around the walls, discharging along the walls and inward as well, especially when there are considerable roof losses.

Suspended type units are located in an elevated position withdrawing air from this higher level and discharging the heated air down into the working zone. This type of installation is illustrated in Figs. 2 and 3. Suspended type units provide excellent temperature distribution.

In closely occupied spaces where direct air drafts into the working zone are not permitted, the floor mounted unit will give more uniform temperature distribution. These units draw the cold air from the floor and discharge the heated air above the working zone.

In general, it is better to direct the discharge from the unit heaters in such fashion that rotational circulation of the entire room content is set up by the system rather than to have the heaters discharge at random and in counter-directions.

Various types and makes of unit heaters are illustrated in the *Catalog Data Section* of this edition. Usually hot blasts of air in working zones are objectionable, so heaters mounted on the floor should have their discharge outlets above the head line and suspended heaters should be placed in such manner and turned in such direction that the heated air stream will not be objectionable in the working zone. In the interest of economy, however, the elevation of the heater outlet and the direction of discharge should be so arranged that the heated air shall be brought as close to the head line as possible, yet not into the working zone. In general, the higher the elevation of the unit, the greater the volume and velocity required to bring the warm air down to the working zone, and consequently, the lower the required temperature of the air leaving the unit.

Low or high pressure steam as well as hot water are generally used in unit heaters. Direct-fired units are also available. Superheated steam can be satisfactorily used in unit heaters provided the capacity is based on saturated steam temperature and not on the total temperature. If

unusually high superheat is used, trouble may be experienced from the excessive expansion and contraction of the heating elements.

Electric, Direct-Fired, and Turbine-Driven Units

The foregoing discussion relates generally to units in which steam or hot water is used as the heating medium. Electric unit heaters are applied where electric power is abundant and cheap and where other forms of fuel are scarce and expensive. The low first cost, easy control, and inexpensive installation of this type of heating have also accounted for many other installations in which electricity has conveniently provided heat for short periods of time. (See Chapter 44.)

A recent development in gas burning equipment is the direct-fired industrial unit heater. These heaters are of the warm-air type and are equipped with fans which cause the air to pass over the heating surfaces at a fairly high velocity and then direct the warm air into the space to be heated. As is the case with the steam-fed unit heaters, the gas-fired appliances may be used for heating stores, shops, and warehouses. They usually are suspended in the space to be heated and in most instances leave the entire floor and wall area free for commercial use. Partial or complete automatic control also may be secured on appliances of this type. This type of heater is often used for temporary heat during building construction or where the installation of a steam or hot water plant is for some reason not justified. For permanent installations, it is usually advisable to provide an exhaust duct from the gas-fired unit heaters to remove products of combustion from the occupied space. While this is not necessary in large open industrial plants, in smaller closed rooms, it becomes essential.

Where high pressure steam is available it is sometimes used to drive a steam turbine direct-connected to the unit heater. The exhaust from this turbine, reduced in pressure, is then passed into the heating coil where it is condensed and returned to the boiler.

UNIT VENTILATORS[5]

Unit ventilators are similar in principle to unit heaters since ventilators incorporate an encased heating surface through which outside air is forced by means of a blower or fan and may or may not have provision for recirculation of air. While unit heaters are largely used for commercial and industrial applications, unit ventilators are intended primarily for schools, offices, and semi-commercial establishments. A typical unit ventilator is illustrated in Fig. 8.

Specifications

Unit ventilators usually consist of a semi-decorative cabinet containing the following necessary or optional parts:

1. Outside air inlet.
2. Inlet damper for closing the opening to the outside air inlet when the unit is not in use.

[5]A roof ventilator is sometimes termed a *unit ventilator*. For information on roof ventilators, see Chapter 42.

3. Adhesive or dry type filters for cleaning the air (optional).
4. A heating element usually of special design and intended for low pressure steam.
5. Motor and fan assembly.
6. Mixing chamber where warm and cold air streams are brought together.
7. Outdoor air inlet and recirculating air mixing damper (optional).
8. Discharge grille or diffuser.
9. Temperature control arrangement.

Functions and Features

The primary functions and features of a unit ventilator are:

1. To supply a given quantity of outdoor air for ventilation or to mix indoor and outdoor air. (See A.S.H.V.E. Ventilation Standards, Chapter 47.)

2. To warm the air to approximately the room temperature if the unit is intended for ventilation only, or to a higher temperature if it is intended to take care of all or a part of the heat transmission losses from the room.

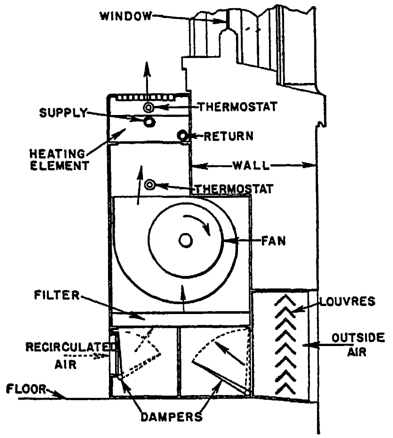

FIG. 8. TYPICAL UNIT VENTILATOR SHOWING ONE OF MANY ARRANGEMENTS OF DAMPERS AND HEATING COILS

3. To control the temperature of the air delivered so as to prevent both cold drafts and overheating. (See Chapter 34.)

4. To deliver air to the room in such a manner that proper distribution is obtained without drafts.

5. To recirculate room air for the purpose of heating or promoting comfort when ventilation is unnecessary. (Ordinances should be consulted.)

6. To perform all its functions without objectionable noise.

7. To clean the air properly.

In general the features of this type of unit are quite similar to those given for unit heaters.

Ratings

Unit ventilators are customarily furnished with two ratings, one established by anemometer readings and the other by condensation. The latter is for standard air. For the former, capacities vary from 750 to 10,000 cfm. Each size may be equipped with radiators for various rates

is practically possible. The following notes will serve as a guide for these selections:

1. The *barometric pressure*, represented by B_0, is the actual pressure at the site of the chimney and not the pressure reduced to sea level datum.

In general, the barometric pressure decreases approximately 0.1 in. of mercury per 100 ft increase in elevation.

2. The *unit weight of a cubic foot* of chimney gases at 0 F and sea level barometric pressure is given by the equation:

$$W_c = 0.131 CO_2 + 0.095\ O_2 + 0.083\ N_2 \tag{3}$$

In this equation CO_2, O_2 and N_2 represent the percentages of the parts by volume of the carbon dioxide, oxygen and nitrogen content, respectively, of the gas analysis. For ordinary operating conditions, the value of W_c may be assumed at 0.09.

The density effect on the chimney gases due to superheated water vapor resulting from moisture and hydrogen in the fuel, or due to any air infiltrations in the chimney proper are disregarded. Though water vapor content is not disclosed by Orsat analysis, its presence tends to reduce the actual weight per cubic foot of chimney gases.

3. The *atmospheric temperature* is the actual observed temperature of the outside air at the time the analysis of the operating chimney is made. The mean atmospheric temperature in the temperate zone is approximately 62 F.

4. The *chimney gas temperature* decreases from the breeching connection to the top of the stack. This drop in temperature depends upon the material and construction of the stack, its tightness or freedom from leaks, its area, its height, and the velocity of the gases through it. The same chimney will suffer different temperature losses depending upon the capacity under which it is working and the variable atmospheric conditions. No general equation covering all these variables has been suggested, but from observations on chimneys varying in diameter from 3 to 16 ft and in height from 100 to 250 ft the following equation was deduced[2];

$$T_c = \frac{3.13\ T_1 \left[\left(\frac{H_b}{3}\right)^{0.96} - 1\right]}{H_b - 3} \tag{4}$$

where

T_1 = absolute temperature at the center of the connection from the breeching, degrees Fahrenheit.

H_b = the height of the stack above center line connection to breeching, feet.

5. The *coefficient of friction* between the chimney gases and a sooted surface has been taken by many workers in this field as a constant value of 0.016 for the conditions involved. This value, of course, would be less for a new unlined steel stack than for a brick or brick-lined chimney, but in time the inside surface of all chimneys regardless of the materials of construction becomes covered with a layer of soot, and thus the coefficient of friction has been taken the same for all types of chimneys and in general constant for all conditions of operation. For reasons of simplicity and convenience to the reader, this constant value of 0.016 has been employed in the development of the various special equations and charts shown in this chapter.

In important chimney design, especially when the construction or the materials are unusual, it is recommended that use be made of Reynolds number[3] in determining the friction factor, *f*.

6. The *length of the friction duct* is the vertical distance between the bottom of the breeching opening and the top of the chimney. Ordinarily this distance is approximately equal to the height of the chimney above the grate level.

7. Assuming no air infiltration the *amount of gases flowing and being discharged* is, of course, equal to the amount of gases generated in the combustion chamber of the

[2]Notes on Power Plant Design, by E. F. Miller and James Holt (*Massachusetts Institute of Technology*, 1930).

[3]For more complete discussion see Flow of Fluids in Closed Conduits, by R. J. S. Pigott (*Mechanical Engineering*, August, 1933).

Substituting in Equation 4:

$$H_t = 24{,}000 + 0.24 \times 0.075 \times 60 \times 1000\ (70-0) = 99{,}600 \text{ Btu per hour}$$

$$t_y = \frac{24{,}000}{0.24 \times 0.075 \times 60 \times 1000} + 70 = 92.2 \text{ F.}$$

When part of the air handled by the unit is taken from the room and the remainder from the outside,

$$H_t = 0.24 W_o\ (t_y - t_o) + 0.24\ W_i\ (t_y - t) \tag{5}$$

where

W_o = weight of air, pounds per hour taken from out-of-doors.
W_i = weight of air, pounds per hour taken from the room.

$$W_o = d_o\ 60\ Q_o \tag{6}$$

$$W_i = d_i\ 60\ Q_i \tag{7}$$

where

d_o = density of air, pounds per cubic foot at temperature t_o.
d_i = density of air, pounds per cubic foot at temperature t.
Q_o = volume of air taken in from the outside, cubic feet per minute.
Q_i = volume of air taken in from the room, cubic feet per minute.

$$t_y = \frac{H}{0.24\ (W_o + W_i)} + t \tag{8}$$

$$H_t = H + 0.24\ d_o\ 60\ Q_o\ (t - t_o) \tag{9}$$

Equations 5, 6, 7, 8, and 9 may be used in the same manner as is illustrated above for Equations 1, 2, 3, and 4. It may be noted in Equation 9, representing the total heat requirements, that as the quantity Q_o is diminished the heat requirements for the unit diminish very materially.

In Example 1, if the quantity of air taken in from the outside is reduced to zero, or all of the air handled by the unit is recirculated, the total heat requirements H_t reduce from 99,600 to 24,000 Btu per hour, or to about one fourth. Such a unit handling one third of its air volume from the outside and two thirds from the room would show a total heat requirement of $24{,}000 + \frac{99{,}600 - 24{,}000}{3} = 59{,}200$ Btu per hour. Units designed and operated on this principle show an average heat requirement and, therefore, a boiler capacity requirement of less than 50 per cent of that required for units taking all their air from the outside.

If all of the air is recirculated, the total heat required is the same as the heat loss of the room, or

$$H_t = H = 0.24\ W\ (t_y - t) \tag{10}$$

If the heat loss of the room is to be taken care of by the direct heating surface, the unit ventilators will be required to warm the air introduced for the ventilating requirements. Therefore:

$$H_v = 0.24\ W\ (t_y - t_o) \tag{11}$$

In this case t_y should be equal to or slightly higher than t. If the unit ventilator were of such capacity as to exactly provide for the ventilating

requirements, the direct radiation would be selected on the usual basis. However, it is necessary to employ a unit which may not exactly meet the ventilating requirements, since standard units are usually rated in terms of the volume of air that will be delivered at a certain temperature t_v for an initial temperature of t_o. Therefore a certain amount of heat (H_h) may be available from the unit ventilator for heating purposes, as previously stated, and the amount of equivalent direct heating surface may, if desired, be deducted from the amount required for heating the room.

Applications

Items to be considered in the application of unit ventilators include the following:

1. Combination with other means of heating.
2. Location of units.
3. Method of venting.

In a *split* system the unit is used primarily for ventilation. Air is delivered to the room at very near the room temperature, and enough separate direct heaters are placed in the room to warm it to the desired temperature, independently of the unit. Their principal advantage lies in offsetting the cooling effect of window and wall surfaces long before these can be heated to room temperature and in retaining heat for this purpose after the ventilation is shut down.

Where the unit ventilator selected has a capacity more than sufficient to warm the air needed to meet the ventilating requirements, a corresponding reduction may be made in the amount of direct heating surface installed. The greater the amount of excess capacity of the unit, the more efficient will be the temperature regulation of the room. The split system permits the heating of the room during failure of electric current, since the direct radiators will furnish heat, but it permits a careless operator to avoid operating the ventilating equipment.

A combined system employs the unit ventilator alone, its capacity being sufficient both for ventilation and for supplying the heat loss. Direct heating surface is omitted altogether. It becomes necessary then that the fan be running whenever the room is to be heated but this also gives assurance of ventilation, especially if automatic dampers are used in the air intake from out-of-doors and in the recirculating intake arranged so as to give a certain quantity of air from the outside (commensurate with weather conditions) whenever the unit is operating and after the room is heated. The cost of installation of a combined system is usually less than that of a split system and there is less danger of overheating, but if the electric energy fails there will be practically no heating.

The location of the unit ventilator in a room is important. Wherever possible it should be placed against an outside wall. It is difficult to obtain proper air distribution if the unit is erected either on an inside wall or in a corner of the room. Standard units discharge the air stream upward, but for special cases units may be installed to discharge air horizontally. Units may be set away from the wall or partially recessed into the wall to save space without materially affecting the results. The air inlet may enter the cabinet at the back at any point from top to bottom

The size and location of the vent[7] outlet is important. In many cases the sizes for public buildings are regulated by law, but the location of the vents generally is left to the discretion of the engineer.

Best results have been obtained with a velocity through the vent openings nearly equal to that at which the air is introduced into the room, thus maintaining a slight pressure in the room. Calculated velocities at the vent openings of from 600 to 800 fpm produce the best diffusion results from this system.

The cross-sectional area of the vent flue itself may be figured on the basis of 15 sq in. of flue for each 100 cfm. Thus the vent flue area of a flue for a room equipped with one 1200 cfm unit ventilating machine would be 180 sq in. The area of vent flue opening from the room may be figured on the basis of 25 sq in. per 100 cfm.

In school buildings provided with wardrobes or cloakrooms the vents may be so located that the air shall pass through these spaces, heating and ventilating them with air which otherwise would be passed to the outside

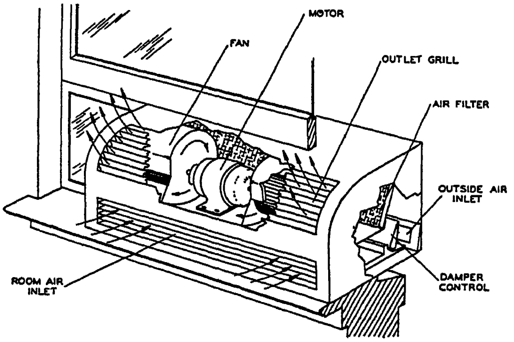

Fig. 9. Typical Window Ventilator

without being used to the best advantage. Many state codes for ventilation of public buildings make this arrangement mandatory.

There has been much controversy over the use of corridor ventilation in school building practice, one group holding the view that when each classroom has a separate vent flue there is a minimum fire risk and less likelihood of cross-contamination, while others emphasize the economy features of the corridor discharge and minimize the fire contamination, and other hazards.

WINDOW VENTILATORS

A window ventilator consists of filters and motor driven fans enclosed in a cabinet to be mounted on the window sill of homes or offices. These units accomplish ventilation, air cleaning, and air circulation. The direction of air discharge is manually adjustable for seasonal operation. Fig. 9 illustrates a unit of this type.

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[7]A.S.H.V.E. Research Report No. 936—Investigation of Air Outlets in Class Room Ventilation, by G. L. Larson, D. W. Nelson, and R. W. Kubasta (A.S.H.V.E. Transactions, Vol. 38, 1932, p. 463).

A.S.H.V.E. Research Report No. 1017—Air Supply to Classrooms in Relation to Vent Flue Openings, by F. C. Houghten, Carl Gutberlet, and M. F. Lichtenfels (A.S.H.V.E. Transactions, Vol. 41, 1935, p. 279).

UNIT HUMIDIFIERS

A unit humidifier consists essentially of some type of equipment for adding moisture to the air, usually a fan to draw the air through the humidifier, and in some cases tempering coils and filters, all encased in a single cabinet. These units are generally used in conjunction with heating systems which do not provide the necessary humidification during winter operation.

In any type of unit humidifier, the process of adding moisture to the air requires heat from a heating coil, water, or air itself.

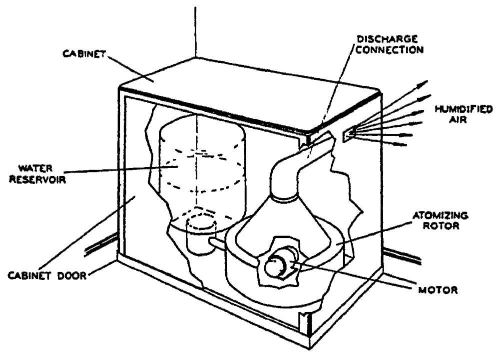

FIG. 10. TYPICAL UNIT HUMIDIFIER OF THE SPRAY TYPE FOR USE IN THE ROOM BEING HUMIDIFIED

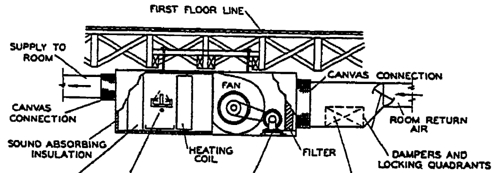

FIG. 11. TYPICAL UNIT HUMIDIFIER OF THE SPRAY TYPE WITH STEAM COIL TO PREHEAT THE AIR FOR RESIDENCES

Types of Units

Small unit humidifiers in decorative casings are made for applications where it is desired to place the unit directly in the room to be humidified. These units are usually of the atomizing type and are completely self-contained. The humidifier water is supplied by a reservoir which must be refilled at intervals. In most units the fine spray of water is mixed with some room air and the mixture is discharged directly into the room. The heat required for humidification in this method is obtained by transforming some of the sensible heat of the air to latent heat. A unit of this type is illustrated in Fig. 10.

Another type of small unit humidifier employs the principle of vaporizing the water by the direct application of heat. One method commonly used is to immerse an electric heating element in a reservoir of water

to heat it until some of the water is vaporized into the air stream. This type of unit is usually used in the same range of capacities as the spray type described previously.

A third type of unit humidifier used extensively is the larger spray type of unit to deliver enough humidifying capacity for a residence. In this type of unit either the water or air is heated. Fig. 11 illustrates a typical unit of this type. These units usually include air filters and in some cases provide ventilation air by means of an outside air duct connection to the unit. The units are available for either floor or ceiling mounting and are usually placed in a central location in the basement with short supply and return duct connections from the first floor. Room air is brought into the unit through the return duct connection and first passes over a tempering coil heated by steam or hot water, then is humidified by passing through some type of spray humidifier. Surplus moisture is removed by an eliminator and the humidified air is delivered to the room through a duct connection. Since a large percentage of the tempering coil capacity is transformed into latent heat during the humidifying process, the unit does not generally eliminate any existing steam radiation but does tend to improve comfort conditions by supplying heating during the off-period of furnace operation.

For a complete discussion of the principles of the various methods of humidification, refer to Chapter 27.

Chapter 23

UNIT AIR CONDITIONERS, COOLING UNITS, ATTIC FANS

Unit Air Conditioners, Functions, Types, Application, Cooling Units, Attic Fans

A UNIT is an assembly of the functional elements indicated by its name, such as air conditioning unit, room cooling unit, etc. A unit of this type may be complete in itself employing its own direct means of air distribution and source of refrigeration or heating, in which case it represents a complete self-contained unit. Or it may be coupled with separate means of refrigeration and air distribution, in which case it will still be considered a unit system in comparison with customary field fabricated central station systems.

The code, Standard Method of Rating and Testing Air Conditioning Equipment[1], defines the various types of unitary equipment:

1. A *Cooling Unit* is a specific air treating combination consisting of means for air circulation and cooling within prescribed temperature limits.
2. An *Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for maintaining temperature and humidity within prescribed limits.
3. A *Cooling Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for cooling and maintaining humidity within prescribed limits.
4. A *Self-Contained Air Conditioning or Cooling Unit* is one in which a condensing unit is combined in the same cabinet with the other functional elements. Self-contained air conditioning units are classified[2] according to the method of rejecting condenser heat (water cooled, air cooled, and evaporatively cooled), method of introducing ventilation air (no ventilation, ventilation by drawing air from outside, ventilation by exhausting room air to the outside, or ventilation by a combination of the last two methods), and method of discharging air to the room (free delivery or pressure type).
5. A *Free Delivery Type Unit* takes in air and discharges it directly to the space to be treated without external elements which impose air resistance.
6. A *Pressure Type Unit* is for use with one or more external elements which impose air resistance.

UNIT AIR CONDITIONERS

This equipment takes the form of an encased assembly including the apparatus necessary to perform either some or all of the functions of

[1]Prepared by a Joint Committee of the *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association*, *National Electrical Manufacturers' Association* and *Air Conditioning Manufacturers' Association*.

[2]Standard Method of Rating and Testing Self-Contained Air Conditioning Units for Comfort Cooling prepared by a Joint Committee of the *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association*, *National Electrical Manufacturers' Association*, and *Air Conditioning Manufacturers' Association*.

cooling, dehumidifying, filtering, ventilation, air circulation, heating, and humidifying. Control of the air conditions is provided by manual switches, automatic devices or a combination of the two. The controls are usually mounted on the units.

The various elements required to produce the effects on the conditioned air are discussed herewith under separate headings.

Heating

Heating in the air conditioning unit is ordinarily accomplished by a heating coil of the non-ferrous finned tube type supplied with either steam or hot water. The steam or hot water is supplied from an external source.

In some cases the heating element may be an electric heater. Where electric power is low in cost, air conditioning units may provide heat from encased or open strip heaters (see Chapter 44). Radiant electric heaters are seldom used except as their radiant heat is absorbed by some receiving wall and then transmitted to the air in the form of convected heat.

Humidifying

Humidifying the air requires a source of heat which may be supplied by applying heat directly to the humidifier water, by applying heat to the air to be humidified, or by picking up heat directly from the air.

The oldest and best known method of humidifying air is by means of a spray. The simplest system is that in which the spray water is furnished from a constant water source, and the excess is permitted to run to waste. The spray effect may be accomplished either by a direct atomizing type which breaks the water down into fine particles by passing it through nozzles, or the water may be directed in a fine jet to a flat surface or target. In these methods some of the sensible heat of the air is transformed into latent heat. These methods are normally inefficient in the use of water. In some cases, the spray is impinged against a heated surface, thereby evaporating some of the water to increase the humidifying capacity. While this is practical in some instances, there is always the danger of scale formation where hard water is employed. A direct steam spray is used in industrial applications but is seldom used in air conditioning units for comfort work due to the resulting odors.

A popular method is to use a drip humidifier. In this type, the water flow is controlled by a solenoid valve and the water is poured into a pan. The pan contains a series of small holes through which the water passes and drips over a built-up section of galvanized screening. The air passes through the screens and picks up moisture.

Another method of humidifying is the evaporative pan type. This is used in comfort conditioning units and consists of a container offering as much water surface as possible and equipped with means of heating the water. The heat may be applied either electrically, by steam, hot water, or by circulation of the water from a heated space through the evaporating pan. Since the humidification is accomplished by surface evaporation only, if low pressure steam or hot water is used, it is essential that the air stream be directed across the surface and that the evaporating surface be large. The evaporative pan type of humidification limits the water wastage and is usually supplied with water through a float valve.

Due to the collection of salts in this evaporating pan such humidifying systems require occasional drainage and cleaning.

Other methods of humidification attempted in air conditioning units are through the use of wetted fabrics, porous earthenware plates, or other capillary surfaces. These methods rely upon the capillary absorption of the moisture from the liquid level into the portion exposed to the air. They have a tendency to lose their effectiveness due to the resulting deposit of mineral salts at the evaporating surfaces thereby clogging the pores and reducing the contact of the air with the water. Also they frequently become foul and often support bacterial growth.

Cooling and Dehumidifying

The cooling and dehumidifying effects on air are produced either simultaneously as in the case of a direct expansion cooling coil, or separately as in the case of an adsorption process and separate cooling coil.

In conditioning units, the use of surface cooling is probably the most common method of producing reduction in dry-bulb temperature of the air and dehumidification simultaneously. The type of surface employed may be cast or fabricated from tubes. In present day practices finned tubes or plate fins through which tubes are passed form the most generally used cooling surface. The detailed fabrication of this surface and the arrangement of the tubes will depend largely upon the type of refrigerant for which it is intended.

The simplest construction is that in which chilled water or brine is used as the refrigerating medium. With direct expansion refrigerant it is usually necessary to provide a special arrangement of headers so that proper distribution of refrigerant through all the surface is obtained. In some cases, ordinary brine coils can be used when operated as a flooded refrigerant system. In some units a combination of a direct spray and a refrigerant surface is used, the spray being directed against the surface. Such systems claim the advantage of air washing together with the maintenance of a clean and effective cooling coil.

When surface coolers are used, adequate protection in the form of filters or at least lint screens are necessary to prevent fouling of the surface from the air borne dirt. Surfaces not so protected frequently become completely matted with lint, grease, and similar dirt.

The sources of refrigeration used with these surface type conditioning units are discussed in Chapter 25. However, they may be divided into the following groups:

1. Direct expansion refrigerant in which the liquid refrigerant is evaporated within the coils of the unit. The vapor from these coils may be recompressed in centrifugal, rotary, or reciprocating type compressors, and the refrigerant again returned to the evaporator coil.

2. Indirect refrigeration by means of:
   a. Cold well water.
   b. Cold city water.
   c. Artificial refrigerated water provided by direct expansion of refrigerant in a water cooler, direct steam jet refrigeration, or by the melting of ice.

Another direct means of cooling and dehumidification is through the use of ice. In such units the ice is brought into as intimate contact as

possible with the air handled. Provision is made for the removal of the moisture as rapidly as it is formed from the melting of the ice. Ice is also used to cool water which is circulated through the sprays.

Other methods of dehumidification accomplished by direct contact with the transfer medium are by means of the so-called *adsorption* and *absorption* systems. (See Chapter 24.) It must be recognized that these methods of dehumidification do not in themselves provide cooling. The substance removes the water vapor from the air thereby heating it. This highly dehumidified air may then be cooled either by partial dehumidification or by direct contact with a cooling medium of cold water or direct expansion refrigerant.

Filtering

The filtering or air cleaning function of an air conditioning unit is accomplished in a variety of ways depending upon the amount of filtering required. In unit systems where filtering alone is considered satisfactory,

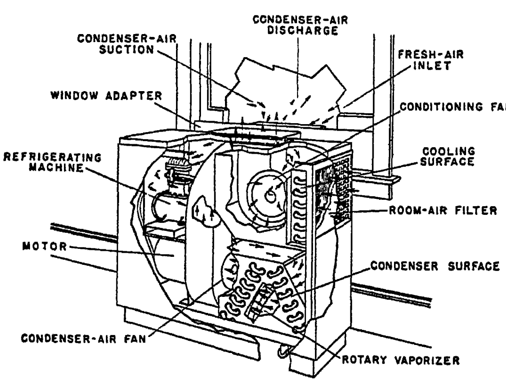

FIG. 1. SELF-CONTAINED ROOM TYPE AIR CONDITIONING UNIT FOR COOLING

the degree of filtering varies widely and in proportion to the actual needs. If the air is chiefly recirculated with but little outside air used for ventilation, filtering requirements are largely limited to keeping the coils in a clean and operable condition. Thus such units are frequently furnished with simple lint screens of low resistance and formed of moderately close meshed wire. Where outside air is used for ventilation, more complete filtering of dust particles is necessary and for this purpose, there are a large number of filters available on the market. Some of these filters are of the so-called *throw-away* type, constructed of inexpensive material so that when they become dirty or clogged they may be thrown away and replaced with new ones. All of these filtering methods are described in detail in Chapter 29.

Ventilating

The ventilating function or introduction of outdoor air is an important consideration in air conditioning units for comfort cooling. While a unit that recirculates all its air capacity is still considered an air conditioning

unit, the better type system provides for the introduction of a certain proportion of outdoor air. In some instances one of several units may operate entirely on outside air, while in other cases only a portion of the air handled by the unit is drawn from out-of-doors. In such cases a damper is provided either in the unit or in the duct connections for controlling the proportion of outdoor air.

Types of Units

Several types and designs of air conditioning units are available for selection. New designs are constantly appearing, with new improvements, greater capacities, wider range of application, and superior construction. Air conditioning units may be classified into the following types:

1. Self-contained air conditioning units.
   a. Room air conditioners for mounting either on the floor or window sill. The condensers are either air, water, or evaporatively cooled.
   b. Store air conditioners for mounting either inside the conditioned space and discharging air directly from the unit, or located outside the conditioned space with ducts connected to the unit. The condensers in this type of unit are water cooled.

2. Remote air conditioning units. These may be either the suspended type or floor type. Design of the floor type of unit varies depending upon the type of application. Units for multiple installation in office buildings or hotels, units for individual offices, and commercial refrigeration units are some of the varieties manufactured.

The self-contained room air conditioning units are finished in decorative cabinets to harmonize with the interiors of residences or offices. For the operation of this unit it is only necessary that it be located adjacent to a window or shaft to which air connections can be made and to plug in the motors to a convenient light socket. A unit of this type is illustrated in Fig. 1. In this particular unit, the conditioned air enters on the side, passing through a grille, filter, and cooling coil, and is delivered vertically to the room through a special motor and fan assembly. Refrigeration is furnished by a reciprocating compressor driven from a motor located in the base. This compressor utilizes an air cooled condenser. Air is drawn into the base by a fan mounted on the compressor motor, so arranged that the air passes through the refrigeration condenser and is again discharged out through the window connection. A novel feature of this design is that the condensate from the cooling coil is sprayed over the condenser surface and there vaporized, thus eliminating the need for drain connections. One advantage of this type of conditioning unit is that it may be removed from the occupied space during the winter season when cooling is not needed.

Other self-contained room air conditioners are available with water cooled or evaporatively cooled condensers. In the water cooled unit, a water connection and drain must be made to the unit, thus reducing the mobility of the unit. The evaporative cooled condenser model also requires a small water connection to supply the necessary water to evaporatively cool the condenser.

Self-contained room air conditioners have been made in sizes from $\frac{1}{3}$ to $1\frac{1}{2}$ hp. The $\frac{1}{3}$ and $\frac{1}{2}$ hp sizes are usually window type units with air cooled condensers.

Self-contained air conditioning units for stores and other commercial establishments have achieved prominence in the last few years. These units range in capacity from 1 to 15 hp. The equipment is enclosed in steel casings and in sizes up to 10 hp is finished to harmonize with the interior of commercial establishments. The units use water cooled condensers and are designed for floor mounting. Units in sizes up to and including 5 hp usually include air distributors to discharge the air directly into the conditioned space, thus eliminating the cost of duct work. The air distributors are adjustable to direct the air flow in such a manner to provide even air distribution in the space being conditioned. These units may use 100 per cent recirculated air or may supply quantities of ventilation air by means of a duct connected from outdoors to the inlet of the unit.

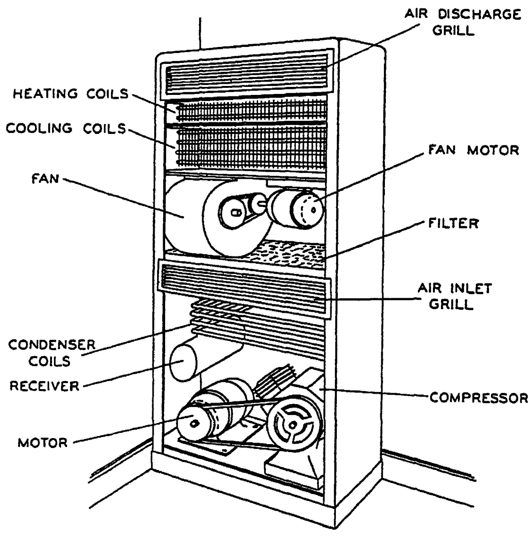

FIG. 2. SELF-CONTAINED STORE TYPE AIR CONDITIONING UNIT

Self-contained store units above 5 hp are usually located adjacent to the conditioned space and ducts connected from the unit to outlets in the conditioned space. This is necessary because it is not usually possible to evenly distribute the large volumes of air handled by these units from a single air distributor. The larger units are therefore not as decoratively finished since they are placed outside the conditioned space.

Another problem in the design of these units is to provide a means of removing the heat of compression and the heat of the motor from the unit. This may be done by a small water cooled cooling coil placed inside the condensing unit compartment. The same water is passed through the water cooled condenser. In the larger sizes, the enclosure around the condensing unit is perforated or screened so that the ambient air removes the condensing unit heat. This is possible since the units are placed outside the conditioned space and appearance of the unit is not a major factor. A third method used with units located in the

conditioned space is to pass some of the recirculated air through the condensing unit compartment then up into the air conditioned section. This method reduces the net cooling effect of the unit since some of the cooled air is used to remove heat from the condensing unit enclosure.

Store units usually provide for the inclusion of heating coils and humidifying equipment as optional equipment where winter ventilation and circulation are desired.

A typical self-contained store air conditioning unit is illustrated in Fig. 2. In this particular unit the air enters a grille located at the front of the unit, passes through a filter, and is discharged by a blower through a cooling and heating coil to an adjustable discharge distributor. The air is delivered in a manner to insure good distribution without being directed at the occupants. In some designs the air may be discharged

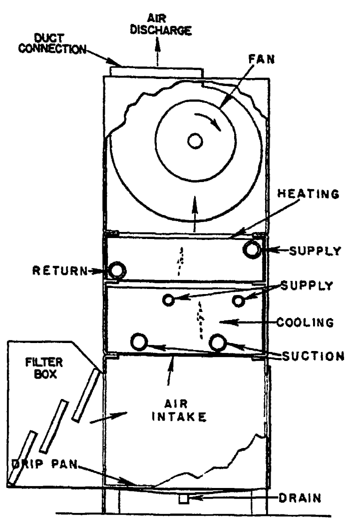

FIG. 3. VERTICAL REMOTE TYPE AIR CONDITIONING UNIT, YEAR 'ROUND

from the sides of the unit as well as from the front. The refrigerating effect is furnished by a reciprocating compressor belt connected to a motor, all mounted on a resilient base to reduce vibration. The heat dissipated by the condensing unit is generally removed by water cooling. Panels are removable for servicing and replacement of filters. The motor starter and other controls are mounted inside the enclosure and the unit may be operated either as a cooling unit or circulating unit by using the manual switches mounted on the side panel, or it may be automatically controlled by means of a thermostat.

Remotely located air conditioning units vary widely in details of construction and are different from the self-contained type of unit in that the sources of refrigeration or heating are not enclosed in the unit.

The vertical floor type remote unit shown in Fig. 3 consists of a fan section, housing one or more fans, mounted on a coil section in which are located a heating coil and a cooling coil, which may be built for

either direct expansion refrigerant, chilled water, or brine. These two sections are supported on a third or drip pan section. The distributing duct system is attached to the fan outlets and return and outside air connections are made to the drip pan. A filter box is illustrated attached to the drip pan section.

A horizontal remote type of air conditioning unit is illustrated in Fig. 4. This unit is similar in construction and operation to the vertical type explained previously. In the smaller sizes this type is installed in the

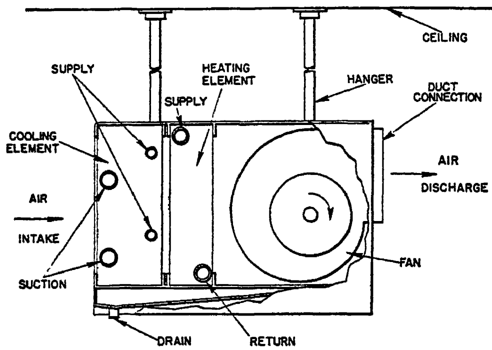

FIG. 4. HORIZONTAL REMOTE TYPE AIR CONDITIONING UNIT, YEAR 'ROUND

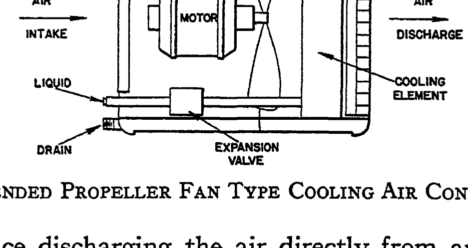

FIG. 5. SUSPENDED PROPELLER FAN TYPE COOLING AIR CONDITIONING UNIT

conditioned space discharging the air directly from an air distributor. A common type of suspended unit for exposed location utilizing a propeller type fan and suitable for summer conditioning only is illustrated in Fig. 5. Such units are equipped with either a direct expansion coil or one for chilled water or brine circulation. The outer cabinet is made of wood-grained steel or baked enamel and is insulated from the cool air chamber to prevent external condensation. The drip from the coil is collected in an insulated drip pan and carried to a drain. The inlet to the unit is provided with a lint screen to protect the cooling surface. Such units are normally used for recirculation only but may be connected for

ventilation through short full-size ducts. Similar units are available with twin housed fans of the same general construction, although usually such fans draw the air instead of blowing it through the coils.

A spray type remote conditioning unit is illustrated in Fig. 6. This spray type unit, which is similar to the arrangement given in Fig. 3, provides for the complete washing of the air and the cooling coil. For winter operation the spray provides means for humidification. The units may be obtained with by-pass dampers as shown in Fig. 6, to provide control of cooling in summer and humidification in winter. The spray type unit without the cooling coil may be used for humidification and heat control. This type of air conditioning unit is used in industrial process air conditioning as well as for comfort air conditioning.

All of these types of remote air conditioning units are usually located

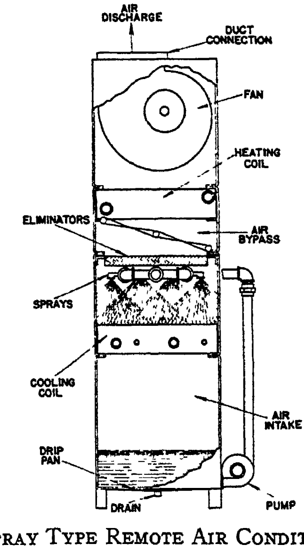

FIG. 6. SPRAY TYPE REMOTE AIR CONDITIONING UNIT

outside the conditioned space with duct work from the unit to the conditioned space and are used mostly in commercial application.

Another type of remote room air conditioner is the type used for multiple installations in office buildings or hotels. An all-year-round floor type remote heating and cooling unit for an exposed location and with direct expansion coil supplied with refrigerant from a remotely located compressor is shown in Fig. 7. A cooling coil for use with chilled water may be substituted for the direct expansion coil indicated. The fans below the separate cooling and heating elements deliver the air against deflectors thereby obtaining distribution across the face of the element and preventing condensate from dripping down into the fans. The plate upon which the fans are mounted serves as the drip pan from which the water is conducted to the drain. Separate elements are used for heating and cooling without manual control. When the unit is used for summer conditioning only, the heating coil may be omitted. The illustration indicates an evaporative type humidifier and drain pan.

Other units are available in which a target spray humidifier is substituted for the evaporative type thereby supplying humidification in winter for application in rooms with other existing heat sources. However, this spray will not provide a great deal of humidification unless the water or air passing through the unit is heated.

Still another remote type of unit is available in which the fans are mounted at the top of the unit delivering directly through a grille and drawing their air supply through the cooling and heating coils. Other variations in proportion and details of construction of this general arrangement are common. With this type of unit, ventilation is usually provided by means of a separate duct connected to the inlet of the unit.

An entirely different arrangement of the remote air conditioner for multiple installation is shown in Fig. 8. This places both the air inlet and the discharge at the top of the unit. The fan at one side discharges the air downward to the bottom where it turns and passes horizontally

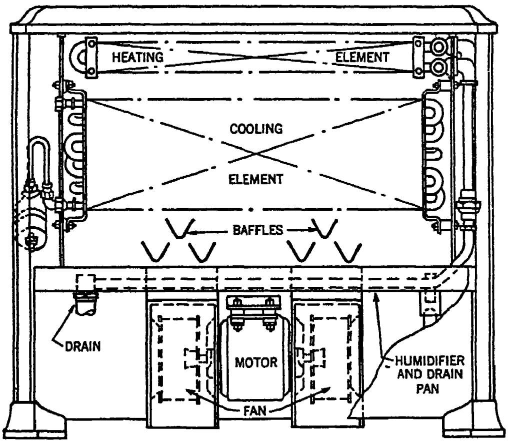

Fig. 7. Floor Type Remote Room Air Conditioning Unit for Heating and Cooling

through an atomizing spray air washer. The path then continues upward through eliminators, a cooling surface and a heating surface before it leaves the unit. With steam or hot water connected to the heating element, this unit gives controlled temperature, humidity, air cleaning, and air movement in both summer and winter. Excess water is run to waste. Acoustical treatment of the housing and outlet baffles permits installation where noise requirements are exacting.

Application

In the application of unit air conditioners it is important to consider the following points:

1. Location of the unit.
2. Air distribution.
3. Use of multiple units in lieu of a central plant system.
4. Self-contained units vs. remote air conditioners.
5. Water usage of self-contained units and methods of water conservation.

In locating air conditioning units, the characteristics of the conditioned space, the building construction, the type of system employed, the duct connections, the accessibility of the unit for servicing, as well as the sources of power, water, refrigeration, heating, and drain connections should be considered.

Locating units in the conditioned space demands serious attention to insure proper air distribution. If ventilation air is required, or if the condenser of a self-contained unit is of the air cooled type, the proximity to a source of outdoor air should be considered when locating the units. Self-contained units with water cooled condensers should be placed close to the water supply and drain, and care must be exercised that the ambient temperature is never below 32 F to prevent freezing the water in the condenser. It is, of course, important to locate the unit so that panels may be easily removed so that parts are easily accessible in case of trouble.

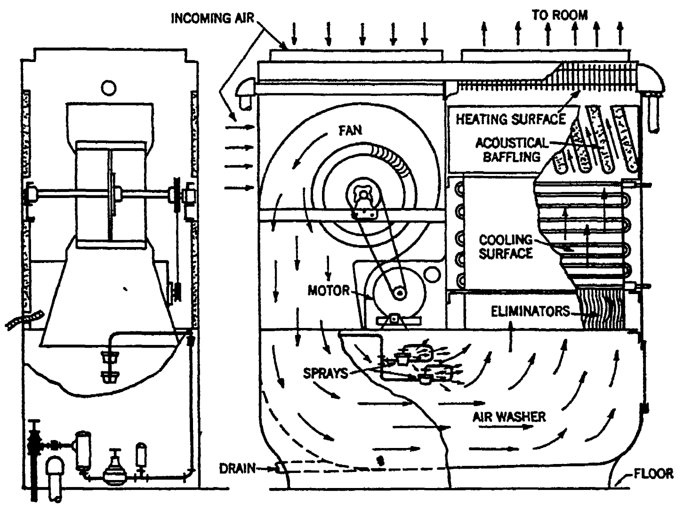

Fig. 8. Remote Type Room Air Conditioning Unit with Top Inlet and Outlet

Location of the remote type of unit also requires consideration of the source of refrigeration or heating. In the smaller sizes these units are placed in the conditioned space. The larger sizes are frequently located externally to the occupied and conditioned space and are connected thereto by means of delivery and return ducts. Such an arrangement permits the location of the conditioning unit convenient to either the source of refrigeration or outside air or both. It frequently permits the use of the basement or of space less valuable than that on the level or floor of the occupied zone. The design then approaches that of a *Central System*, (see Chapter 21). Oftentimes the same type of unit may find application in an exposed position for one job and in a concealed location for another. Frequently conditioning units are built into the structure or into the architectural design of a room so that they are entirely concealed except for the discharge and return grilles which are designed so as to correspond to the decorative scheme of the room.

must be supplied to the condensing unit enclosure cooling coil, thus defeating the purpose of the evaporative condenser to conserve water.

Ratings

There are two codes governing the rating and testing of air conditioning units. The first code, Standard Method of Rating and Testing Air Conditioning Equipment[3], covers all types of air conditioning units except the self-contained type. The latter is covered by the Standard Method of Rating and Testing Self-Contained Air Conditioning Units for Comfort Cooling[4]. The two codes are necessary because of the basic difference caused by the heat given up by the self-contained units.

Self-contained air conditioning unit ratings are expressed in the code in terms of the effect produced on air such as:

1. The net total room cooling effect in Btu per hour. This is the actual heat removed from the room and is equal to the gross cooling effect less the heat given back to the room by the unit.
2. The net room dehumidifying effect in Btu per hour.
3. The net room sensible cooling effect in Btu per hour.
4. The sensible heating effect in Btu per hour.
5. The humidifying effect in pounds per hour.
6. The total air capacity in cubic feet per minute of standard air.

The standard rating basis as given in the code for self-contained units is tabulated in Table 1.

The standard rating basis for air conditioning units is similar to that given previously for self-contained units except the relative humidity of the entering air is specified as 50 per cent (66.7 F wet-bulb) instead of specifying the wet-bulb temperature as 67 F and the suction saturated refrigerant temperature is specified as 40 F for comfort cooling since an air conditioning unit does not include a condensing unit. The suction saturated refrigerant temperature of a self-contained air conditioning unit is not given in Table 1 as a basis of rating since this temperature is the temperature obtained when the self-contained unit operates as a system at the conditions given in Table 1. It is expected that the entering air conditions for the standard rating will agree when the code, Standard Method of Rating and Testing Air Conditioning Equipment, is revised.

COOLING UNITS

Cooling units may be used either in comfort cooling or commercial applications. As applied to industrial product conditioning and processing they are similar in construction to unit heaters described in Chapter 22 except that the heat transfer surface is supplied with refrigeration instead of heat. They are normally installed within the space to be served, or at least closely adjacent thereto.

Product cooling was originally accomplished by means of stationary pipe coils. This was later supplemented with the forced fan bunker systems in which air was passed over banks of coils. Today the coils

[3]Loc. Cit. Note 1.
[4]Loc. Cit. Note 2.

and fan are encased in an enclosure and controls are provided to maintain an average coil surface temperature. Thus, for any installation, the depth of coil, air flow, and face area determine the relation between dry-bulb temperature reduction and wet-bulb temperature reduction. Occasionally they are provided to receive outside air in which case this air is invariably filtered or washed to prevent any possible contamination of the product.

Features of Units

The principal field for cooling units is in cold storage plants, fur storage, fruit packing houses, provision stores, brewery fermentation and stock rooms, candy plants, and other industrial process work. In replacing bunker and wall coils in meat storage plants, cooling units give distinct advantages in compactness, lower first cost and maintenance expense, ease of defrosting, freedom from drip and the maintenance of sanitary conditions, as well as uniform temperature and humidity under variable load conditions. Cooling units by means of their positive air

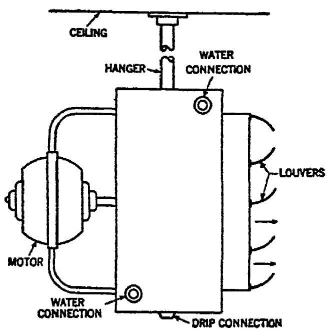

FIG. 9. CEILING TYPE COOLING UNIT

circulation prevent dead-air spots, frequently objectionable in this industry.

Types of Units

Cooling units are provided in two major types similar to unit heaters, either floor mounted with housed fan, or suspended with propeller type fans. Normally, air outlet velocities are lower than for heating, due largely to the effect of high velocities on the product. Cooling units are normally of the free delivery type although they occasionally are supplemented with duct work to provide more careful air distribution.

Typical cooling units are shown in Figs. 9, 10, and 11. Fig. 9 indicates a suspended type cooling unit which may be designed with or without a moisture eliminator. If high air velocities are maintained, an eliminator will be necessary to prevent the drops of moisture from being carried through with the air. The condensation that occurs is collected in a drip pan and removed from the system through a drain pipe. Fig. 10 indicates a typical floor-mounted unit of the housed fan type. The illustration shows a common form of distributing outlet designed to give low outlet velocities together with a controlled distribution. In process work, it is

often important that direct air distribution does not impinge on the product. Cooling units are normally constructed of galvanized steel or non-ferrous material in order to reduce the corrosive effect of their constant wetted condition.

Ratings

Since cooling units are mostly used in low temperature applications, the standard basis of rating is different than for air conditioning units. The Standard for Rating and Testing Air Conditioning Equipment states that the standard rating shall be based on air entering the cooling unit at 45 F dry-bulb and 85 per cent relative humidity, and the suction saturated refrigeration temperature shall be 30 F for commercial cooling.

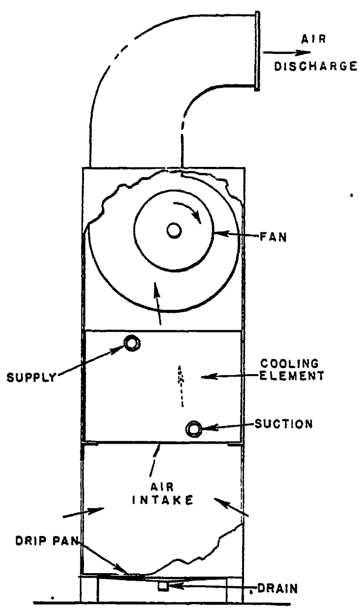

Fig. 10. Surface Type Cooling Unit

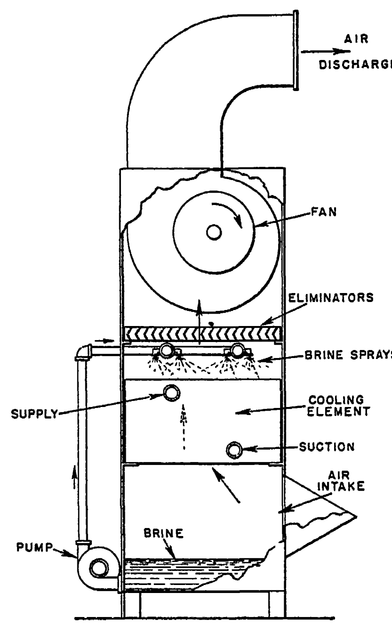

Fig. 11. Brine Spray Type Cooling Unit

Ratings of cooling units may be expressed in Btu per hour, or in tons of refrigeration. When ratings at other than standard conditions are given, the quantity, temperature, and relative humidity of the entering air should be specified, together with the refrigerant temperature within the coil. When chilled water or brine is used, the rate of circulation of the cooling media as well as its entering temperature should be given.

Defrosting

Cooling units are often called upon to operate in rooms where a temperature below freezing is maintained and low refrigerant temperatures are required. This results in the collection of frost on the heat transfer surface which in turn leads to a rapid loss in capacity and requires eventual defrosting. Such defrosting is accomplished by the following methods:

1. When the room is above freezing the source of refrigeration is cut off and the fan allowed to operate until the unit has defrosted.

2. A reversal of the refrigeration system may be provided and the so-called hot gas defrosting method used. This is accomplished by reversing the flow of the hot gas so that it is delivered directly from the compressor to the evaporator cooling unit. As soon as the ice and frost have been melted, the system is again returned to its normal cycle.

3. Where brine is used as a refrigerant, heated brine may be sent through the cooler to remove the ice.

4. When the room is at very low temperatures, warm air defrosting is sometimes used by providing for the admission and removal of warm air from outside the cooled space.

5. The surface may be sprayed with a strong brine solution.

In order to prevent the collection of frost in low temperature rooms where high latent heat loads are present, unit coolers equipped with a constant brine spray are frequently used. These are normally of the housed fan type similar to Fig. 10, but equipped with a pump for recirculating brine at intervals to maintain a non-freezing mixture as shown in Fig. 11.

COSTS

The following factors influence the cost of unit air conditioning installations:

1. Since the cost of the total job involves material cost plus installation labor and since through the use of unitary equipment, material costs can be kept to a minimum, every effort should be made to simplify installation.

2. Self-contained units in the small sizes now available, probably represent the lowest cost individual installations. They have, however, their limitations.

3. The floor type all-year-round air conditioning units for the occupied space with a remotely controlled compressor, heating sources being either the existing heat system or steam connections to the unit, probably afford the lowest cost all-year-round service for most individual rooms.

4. For multiple rooms or offices, the remotely located air conditioning unit with remote source of refrigeration probably represents the most economical installation. The larger self-contained air conditioners are particularly adaptable to stores, residences and small commercial installations.

Costs of operation vary widely depending upon the cost of power and water. Water costs in the larger installations are being materially reduced through the use of cooling towers and special types of condensers. It is difficult to make any comparison of operating costs of cooling as contrasted with heating equipment because the relative operating expense depends upon many factors including climatic conditions; *i.e.*, in the South the cost of operating cooling equipment greatly exceeds the operating cost of heating equipment, whereas in colder climates where cooling equipment is used about two months per year, the heating costs are probably higher than those for cooling.

ATTIC FANS

Attic fans, used during the warm months of the year to draw large volumes of outside air through a house, offer a means of using the comparative coolness of outside evening and night air to bring down the inside temperature.

Because the low static pressures involved are usually less than 1/8 in. of water, disc or propeller fans are generally used instead of the blower or housed types. The fans should have quiet operating characteristics, and they should be capable of giving about 20 to 30 air changes per hour in northern areas. In the South the usual specification requires one air change per minute which provides appreciable air movement in addition to the cooling effect.

Types

Open attic fans are units in which the fan is installed in a gable or dormer and one or more grilles are provided in the ceilings of the rooms below. Outdoor air, which enters the house through open windows, is drawn into the attic through the grilles, and is discharged out-of-doors by the fan. An attic stairway may be used in place of the grilles. It is essential that the roof and the attic walls be free from air leaks.

Boxed-in fans are units in which the fan is installed within the attic in a box or housing directly over a central ceiling grille, or in a bulkhead enclosing an attic stair. The fan may be connected by a duct system to the grilles in individual rooms. Outdoor air entering through the windows of the rooms below is discharged into the attic space and escapes to the outside through louvers, dormer windows, or screened openings under the eaves.

Location

The locations of the fan, the outlet openings, and the grilles should be selected after consideration of the room and attic arrangement in order to give uniform air distribution in the individual rooms served. If the outlet for the air is not on the side away from the direction of the prevailing wind, openings should be provided on all sides. Kitchens should be separately ventilated because of the fire hazard, and to prevent the spread of cooking odors.

Costs

The capacity of attic fans is from 3,000 to 30,000 cfm with the trend toward units in the range of capacity from 7,000 to 15,000 cfm.

Some typical data on an attic fan installation in an average six room house of frame construction containing 14,000 cu ft and located in the southern part of this country are:

Installed cost	\$100 to \$400, average \$200
Fan data	12,000 cfm average, 500 watts input
Operating period	April 15 to October 15, intermittently as weather conditions demand
Power consumption	500 kwhr per year for 8 months' operation

In northern climates the figures would be considerably reduced. A smaller capacity fan can be effectively used and the cost of an installation ranges from \$60.00 upwards.

Chapter 24

COOLING, DEHUMIDIFICATION AND DEHYDRATION

Definitions and Methods, Adsorbents, Absorbents, Nature of Processes, Temperature—Pressure—Concentration Relations, Dehydration Methods, Auxiliaries, Controls, Performance, Economics

THE addition or abstraction of heat to or from air, whether sensible or latent, requires (*a*) a medium held at the necessary temperature or vapor pressure to produce a flow of heat or moisture and (*b*) sufficient contact between the air and the medium to achieve the desired final condition. The medium may be solid or liquid. It may be used (*a*) directly, as in a water or brine spray, or (*b*) indirectly, as with a steam radiator or direct expansion cooling coil.

The contact is obtained through the use of exposed surface, to which the molecules of air are brought into direct physical proximity, thereby producing the heat interchange. These molecules then re-mix with uncontacted molecules in the air stream. The completeness of the interchange is a function of the number of such successive contacts, and is a measure of the efficiency of the surface[1]. The contacting surface may be that of the medium directly, such as a finely atomized spray or the bed of a solid dehydrating agent; or a chilled or warmed metal surface, as a coil; or a combination of medium and surface, such as a packed tower, where the medium produces the interchange and the surface provides the necessary contact area.

DEFINITIONS AND METHODS

There are several basic methods of producing the necessary difference in temperature or vapor pressure between air and the medium employed to achieve cooling or dehumidification, or both simultaneously:

Cooling of air involves its reduction in temperature due to the abstraction of sensible heat. It is always a result of contact with a medium held at a temperature lower than that of the air. Cooling may be accompanied by moisture addition (evaporation), by moisture extraction (dehumidification), or by no change of moisture content whatever. Such moisture change, if present, is considered as a secondary or by-product effect. As

[1]The Contact Mixture Analogy Applied to Heat Transfer with Mixtures of Air and Water Vapor, by W. H. Carrier (*A.S.M.E. Transactions*, Vol. 59, 1937).

previously stated, the medium may be directly in contact with the air (as water, brine, or ice), or indirectly through a barrier wall (as cooling surface). When the latter method is used, and the surface temperature is held above the air dew-point, only cooling occurs without moisture interchange.

Evaporative Cooling involves the adiabatic exchange of heat between air and a water spray or wetted surface. The water assumes the wet-bulb temperature of the air, which remains constant during its traverse of the exchanger. No heat is added or abstracted from the medium (water), which is continually recirculated. Cooling of the air occurs due to the temperature difference between entering air, and water at the wet-bulb temperature. Humidification occurs as a result of the vapor pressure exerted by the water which is higher than that corresponding to the entering air dew-point. Since this is an adiabatic exchange, the enthalpy of the air remains constant, while the dew-point rises and the dry-bulb falls, and the loss of sensible heat exactly equals the gain in latent heat (neglecting radiation losses). The maximum available temperature reduction is the total difference between entering dry- and wet-bulbs (wet-bulb depression). Equipment achieving the complete reduction is termed *completely* saturating or 100 per cent efficient, since the air leaves in a saturated state. Equipment utilizing only a portion of the wet-bulb depression is termed *partially* saturating.

Evaporative cooling is being used advantageously in many parts of the country. It is particularly applicable (1) in districts where the relative humidity is normally low during the cooling season, and (2) in applications where the cooling load is principally a sensible load.

Dehumidification of air, in its broadest connotation, means simply the removal of moisture. Usage in the art has restricted the application of the term, so that the former broad meaning is now properly covered by the complementary names dehumidification and dehydration. *Dehumidification* usually refers to the *condensation* of water vapor from air due to its contact with a chilled medium (see Cooling). This type of heat exchange invariably includes temperature reduction due to removal of sensible heat, which reduction may be considered a by-product effect.

Dehydration refers specifically to the removal of water vapor from air due to its contact with a dehydrating agent. The primary distinction between dehumidification and dehydration is the vapor pressure exerted at the surface of the contacting medium. In the case of dehumidification, this surface vapor pressure is always the same as that which would be exerted by a body of water (or ice) at that same surface temperature. In the case of a dehydrating agent, the surface vapor pressure is always *lower* than that exerted by water at the same temperature, and the effectiveness of the medium as a dessicant is largely a function of the amount by which this vapor pressure can be lowered at the working temperature involved.

Thus it is evident that the primary function of a dehydrating agent is to establish a vapor pressure difference between the air and the medium in order to secure thereby a removal of moisture (latent heat) from the air. In the simplest type of process, no heat is abstracted from the medium itself, and the process is essentially an adiabatic one in which the latent heat lost by the air is converted to sensible heat which raises the air temperature by an equivalent amount. This process is therefore an energy exchange, similar to, but the reverse of, adiabatic saturation.

Combination Methods. It is evident that two or more of the above processes—cooling, evaporative cooling, dehumidification and dehydration—may be combined by the proper application of interchangers in sequence. Such combinations are dictated by the availability of prime sources of energy and the economic justification of each.

This chapter discusses in detail the engineering and economic principles involved in the application of dehydration. For similar discussion of the other processes, refer to the following material: Cooling and dehumidification by the use of surface interchangers (cooling coils), see Chapter 26. Cooling, dehumidification and evaporative cooling with air washers, see Chapter 27. For sources of cooling involving city and well water and cooling towers, see Chapter 27, while for mechanical refrigeration and ice, refer to Chapter 25. For the thermodynamics of evaporative cooling, see Chapter 1.

DEHYDRATING AGENTS

Dehydrating agents may be divided into two general classifications:

1. *Adsorbent*—A material which has the ability to condense water vapor on its surface without itself being changed physically or chemically. Certain solid materials, such as silica gel, activated alumina and activated carbon have this property.

2. *Absorbent*—A material which has the ability to take up water vapor but which changes physically, chemically, or both, during the cycle. Calcium chloride is an example of a solid material while liquid materials include lithium chloride, calcium chloride, lithium bromide and ethylene glycol.

Adsorbents

These substances are characterized by a physical structure containing a great number of extremely small pores but still retaining sufficient mechanical strength to resist the wear and handling to which they are subjected. To be suitable for dehydration purposes such substances must fulfill the following requirements:

1. Possess suitable vapor pressure characteristics.
2. Be available at an economical cost.
3. Adsorb sufficient moisture per pound of material to avoid excessive bed dimensions.
4. Be chemically stable, resisting contamination from impurities.
5. Physically rugged to resist breakdown from handling, abrasion, etc.
6. Withstand breakdown from indefinitely repeated reactivation cycles.
7. Possess practical and efficient reactivation temperatures.

Aluminum Oxide (Alumina), in a porous, amorphous form is a solid adsorbent frequently called by the common name activated alumina. It contains small amounts of hydrated aluminum oxide, very small amounts of soda, and various metallic oxides. A good grade of activated alumina will show 92 per cent of Al_2O_3, and its soda content will be combined with silica and alumina into an insoluble compound. This substance also has the property of adsorbing certain gases and certain vapors other than water vapor—a property which is sometimes useful in air conditioning installations. It is available commercially in granules ranging from a fine powder to pieces approximately 1.5 in. in diameter. It has high adsorptive capacity per unit of weight and is non-toxic. It may be repeatedly re-activated after becoming saturated with adsorbed moisture without practical loss of its adsorptive ability. In the grade frequently used for air drying the re-activation may be accomplished at temperatures under 350 F. Specific gravity is 3.25 and the pores are reported to occupy 58 per cent of the volume of each particle. For most estimating purposes the volume-weight relation on a dry basis may be taken as 50 lb per cubic foot although in the smaller sizes the packed weight may be as much as 64 lb per cubic foot.

Silicon Dioxide (Silica), in a special form obtained by suitably mixing sulphuric acid with sodium silicate, is another solid adsorbent and is commonly called silica gel. Its capillary structure is exceedingly small, so small that its exact structure has to be deduced rather than observed. The gel is available commercially in a wide variety of sizes of granules ranging from 4 to 300 mesh. It has high adsorptive capacity per unit of weight, it is non-toxic, and may be repeatedly re-activated without

practical deterioration. Re-activation may be accomplished at temperatures of air up to 600 F although it is frequently accomplished with air or other gases at temperatures not over 300 F. Volume of the capillary pores is reported to be from 50 to 70 per cent of the total solid volume. For most estimating purposes the volume-weight relation can be assumed as from 38 to 40 lb per cubic foot on a dry basis. It also has the property of absorbing certain gas and vapors other than water vapor.

Other substances having properties which make them available as solid adsorbents include lamisilate and charcoal but details of their physical properties are not available.

Nature of Adsorption Process

The adsorbent does not go into solution but water vapor is extracted from the air-vapor stream passing through the bed of adsorbent material and is caught and retained in the capillary pores. The exact nature of the process which goes on during adsorption is not known but it is stated that the action is brought about by surface condensation, and also by a difference between the vapor pressure of the water condensing inside the pores and the partial pressure of the water vapor in the air-vapor mixture. The adsorbing process in the bed can continue until the vapor pressures come into equilibrium. The amount of vapor adsorbed will depend on the adsorbent substances being used, but for any single substance the amount depends on the temperature of the bed as well as on the partial pressure of the air-vapor mixture being passed over it.

As the bed of material adsorbs moisture, its vapor pressure approaches that of the contacting air and the rate of adsorption gradually slows down so that equilibrium may not be reached for 24 to 48 hours. Because of this diminishing rate of adsorption, commercially designed systems do not permit the state of equilibrium to be reached but generally operate on a 10 to 30 min contact time—the period of most rapid adsorption.

As the process of adsorption goes on heat is liberated in the bed. The heat so liberated is the latent heat of the water vapor condensed together with the so-called heat of wetting. For a pound of water vapor at 60 F the latent heat released by condensation is approximately 1057 Btu. The heat of wetting for silica gel, for example, is about 200 Btu, making a total heat of adsorption of approximately 1257 Btu per pound of water adsorbed from the air-vapor mixture passing through the silica gel bed. The heat of wetting varies with the substance being used as the adsorbent while the latent heat of condensation depends only on the temperature and pressure of the water vapor.

Temperature—Pressure—Concentration Relations

Since the adsorptive ability of an adsorbent depends on the temperature of the bed and on the partial pressure difference between the pores and the air-vapor mixture, it is important to know the pressures and temperatures at which pressure equilibrium is reached.

Evidently the equilibrium conditions represent the limits beyond which adsorption of vapor cannot continue. The relationship can be shown graphically and Fig. 1 is such a chart for silica gel. Charts of like nature can be plotted for other adsorbent materials.

The equilibrium conditions for a gel bed maintained at constant temperature while the water vapor adsorption is allowed to continue until pressure equilibrium is reached is shown in Fig. 1. Each curve on the chart shows a certain dew-point temperature, and therefore a certain pressure of the saturated water vapor.

As an example in the interpretation of the chart consider the case when moist air at a temperature of 80 F and a partial vapor pressure of 0.5 in. of mercury flows through a bed of silica gel which is at a temperature of 80 F. The chart indicates that the equilibrium of pressure between the

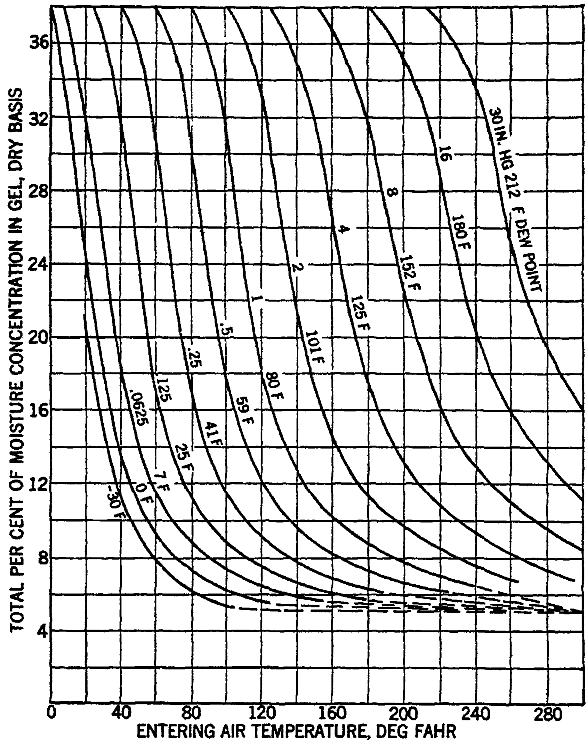

FIG. 1. TEMPERATURE—VAPOR PRESSURE—CONCENTRATION RELATION FOR A SILICA GEL BED AT CONSTANT TEMPERATURE

air-vapor mixture and the bed is reached when the dry bed has adsorbed moisture to the extent of 30 per cent of the weight when dry. When this happens the bed can adsorb no more moisture unless its temperature is changed.

While charts of this kind can show the limiting properties of the substances they are seldom directly applicable to the solution of air conditioning problems unless considerable additional information is available. This takes the form of performance data covering the characteristics of the equipment in which the adsorbent bed is placed. Such performance data are presented later in this discussion.

Absorbents

Any absorbent substance may be used as an air drying agent if it has a vapor pressure lower than the vapor pressure in the air-vapor mixture from which the moisture is to be removed.

Solid Absorbents. The substances used are in general the solid forms of the liquid absorbents, more commonly calcium chloride due to its low cost. At present they are used principally in small dessicating chambers, and in small dryers of the cartridge type, through which air is forced under pressure.

Liquid Absorbents. These are characteristically water solutions of materials in which the vapor pressure is reduced to a suitable level by governing the concentration of the solution. In addition to having suitable vapor pressure characteristics a practical absorbent must also be widely available at economical cost, be non-corrosive, odorless, non-toxic, non-inflammable, chemically inert against any impurities in the air stream, stable over the range of use and especially it must not precipitate out at the lowest temperature to which the apparatus is exposed. It must have low viscosity and be capable of being economically regenerated or concentrated after having been diluted by absorbing moisture.

Water solutions, or brines, of the chlorides or bromides of various inorganic elements such as lithium chloride and calcium chloride are the absorbents most frequently used in connection with air conditioning applications and detailed attention is confined to these two in this chapter.

Nature of Absorption Process

The application consists of bringing the air-vapor stream into intimate contact with the absorbent, permissibly by passing the air stream through a finely divided spray of the brine but more generally by passing the air over a contacting pack where the liquid absorbent presents a large surface to the air stream. The difference in vapor pressure causes some of the vapor in the air-vapor mixture to migrate into the brine. Here it condenses into liquid water and decreases the concentration of the absorbent.

As the water vapor is added to the absorbent and condenses, it gives up its latent heat of condensation which tends to raise the temperature of both the absorbent and the moist air stream. For every pound of water absorbed and condensed the heat added to the air stream and the brine combined is obtainable from steam tables. For instance, at 60 F the amount of this heat is about 1057 Btu. In addition to this heat there is involved also the so-called heat of mixing which is frequently considerable.

A more complicated cycle involves heat removal from the contacting medium, either within or external to the interchanger. Thus the temperature of the medium may be higher than, equal to, or lower than that of the air, depending on the agent used and the function to be performed. In such a cycle, the dehydration process may be accompanied by cooling or heating, or neither, and such effect, if present, may be either a necessary by-product of the process, or for the specific purpose of obtaining both latent and sensible heat removal simultaneously. The heat thus produced in the bed is to a large extent transferred to the air being dried, and in the average air conditioning installation must be removed by passing the air through an aftercooler.

Temperature—Pressure—Concentration Relations

Since the absorption process can continue only as long as there is a difference in vapor pressure between the absorbent and the air-vapor mixture and since at a given temperature of the absorbent the vapor pressure depends on the concentration of the solution, evidently there must be a relation between these quantities which if known would state the limits of the process. The relationship would also depend on the absorbent being used, and would have to be determined for each substance used as an absorbent. This relationship is shown graphically in

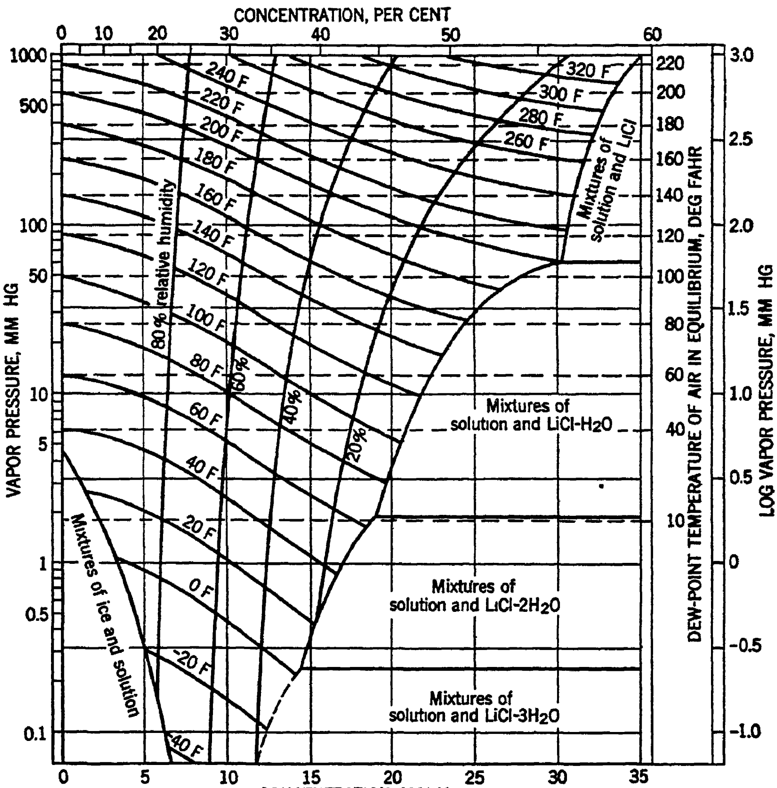

FIG. 2. TEMPERATURE—PRESSURE—CONCENTRATIONS FOR LITHIUM CHLORIDE

Fig. 2 for lithium chloride, and Fig. 3 presents similar data for calcium chloride. These charts are essentially similar to Fig. 1, and their direct usefulness is limited by much the same considerations. Other physical properties of lithium chloride are shown in Tables 1, 2 and 3.

In Fig. 2 and Table 1 the unit of concentration is the *mol.* A M molal solution is defined as a solution containing $M \times 42.37$ grains of anhydrous lithium chloride per 1000 grains of water. The formula connecting concentration in mols with weight in per cent is equivalent to: $(100 \times M \times 42.37) \div [1000 + (M \times 42.37)]$.

Table 1. Properties of Lithium Chloride Solutions

Concentration Pound Mols (42.4 Lb) *LiCl* per 1000 Lb Water	Concentration Per Cent by Weight	Specific Gravity at 100 F	Viscosity (Millipoise)		Partial Heat of Mixing at 0 F, Btu per Lb	Temp. Coeff. of Partial Heat of Mixing Btu per Lb per F	Specific Heat at 70 F	Boiling Point F at (760 mm Hg)	Freezing Point, F
			80 F	180 F					
0	0.0	1.000	8.61	3.48	0.00	0.000	0.998	212.0	32.0
2	7.8	1.037	11.19	4.56	2.04	−0.014	0.901	215.8	16.3
4	14.5	1.076	14.42	6.01	7.24	−0.036	0.831	221.5	− 5.8
6	20.2	1.111	18.62	7.78	16.70	−0.069	0.778	228.9	−34.2
8	25.3	1.143	24.32	10.00	31.90	−0.109	0.739	238.1	−69.0
10	29.7	1.172	32.28	12.91	51.10	−0.143	0.710	248.4	−90.0
12	33.7	1.199	43.45	16.56	75.70	−0.160	0.687	258.8	−40.0
14	37.4	1.225	60.26	21.28	90.80	−0.167	0.666	268.9	1.0
16	40.4	1.248	82.04	27.10	124.80	−0.176	0.647	277.9	36.5
18	43.3	1.270	113.80	35.48	145.00	−0.186	0.631	285.8	58.1
20	46.0	1.291		46.45	162.00	−0.194		293.2	86.4
22	48.4			60.67	171.00	−0.200		300.2	133.0
24	50.3			84.33	177.00	−0.200		307.0	156.0
26	52.4				182.00	−0.210		313.0	180.0
28	54.3				191.00	−0.210		318.0	190.0
30	56.1				194.00	−0.210		323.0	195.0
32	57.5				198.00	−0.220		328.0	280.0

Table 2. Dew-Point of Air in Equilibrium with Lithium Chloride Solutions Concentration in Pound Mols (42.4 lb) Lithium Chloride per 1000 lb Water

Dew-Point at Zero Conc.	Concentration of Lithium Chloride														
	2.0	4.0	6.0	8.0	10 0	12.0	14.0	16 0	18.0	20.0	22.0	24.0	26.0	28.0	30.0
320	315.2	308.7	299.9	290.2	279.7	269.4	259.6	251.5	244.1	236.5	230.0	223.8	218.6	214.5	210.3
300	295.4	289.1	280.5	270.9	260.6	250.5	240.8	232.6	225.4	218.0	211.8	205.8	200.8	196.9	192.8
280	275.6	269.5	261.1	251.7	241.5	231.6	222.2	214.0	206.7	199.7	193.5	187.8	183.2	179.3	175.2
260	255.8	250.0	241.9	232.6	222.7	212.8	203.5	195.5	188.4	181.7	175.4	170.0	165.6	162.0	158.4
240	236.0	230.4	222.5	213.5	203.8	194.2	185.0	177.1	170.0	163.6	157.5	152.2	148.3	144.6	140.5
220	216.2	210.8	203.2	194.4	184.9	175.5	166.4	158.6	151.6	145.3	139.6	134.6	130.7	127.3	124.2
200	196.4	191.2	183.9	175.4	166.1	156.7	148.0	140.3	133.5	127.3	121.9	117.0	113.3	110.1	
180	176.6	171.6	164.7	156.4	147.3	138.1	129.6	122.1	115.5	109.4	104.2	99.6	96.0		
160	156.8	152.1	145.4	137.4	128.6	119.7	111.3	103.9	97.4	91.6	86.6	82.2			
140	137.0	132.6	126.1	118.4	109.9	101.3	93.1	85.9	79.5	73.8	69.0				
120	117.2	113.0	106.8	99.4	91.1	82.7	74.7	67.8	61.5	56.0					
110	107.3	103.2	97.2	89.9	81.9	73.5	65.6	58.8	52.6	47.1					
100	97.4	93.4	87.5	80.5	72.7	64.4	56.6	49.8	43.7	38.2					
90	87.5	83.6	77.9	71.0	63.3	55.2	47.6	40.8	34.8	29.3					
80	77.6	73.8	68.4	61.6	54.0	46.1	38.5	31.8	25.9	20.6					
70	67.7	64.0	58.7	52.2	44.8	37.0	29.5	22.9	17.2	12.0					
60	57.8	54.3	49.1	42.7	35.5	27.9	20.5	14.0	8.3						
40	38.0	34.7	29.9	23.9	16.9	9.6	2.4	−3.9							
20		15.1	10.7	5.0	−1.7	−8.7	−15.4								
0		−4.5	−8.6	−13.9	−20.2	−27.0	−33.3								

Example 1. Determine the dew-point, wet-bulb, relative humidity and absolute humidity of air in equilibrium at 100 F with pure lithium chloride solution of density 1.270.

Solution. From Table 1 the concentration of a solution of density 1.270 at 100 F is 18.0 *M*. From Fig. 2 the dew-point of 18 *M* lithium chloride at 100 F is 43.7 F. From Table 6, Chapter 1, the partial pressure of water over the solution is 0.2858 in. of Hg, and the absolute humidity is 42.00 grains per pound dry air. From the psychrometric chart, the wet-bulb is 66.3 F, and the relative humidity is 15 per cent.

Example 2. Determine the boiling point, and freezing point of 18 *M* lithium chloride solutions.

Solution. From Table 1, boiling point (standard) is 285.8 F, freezing point is 58.1 F.

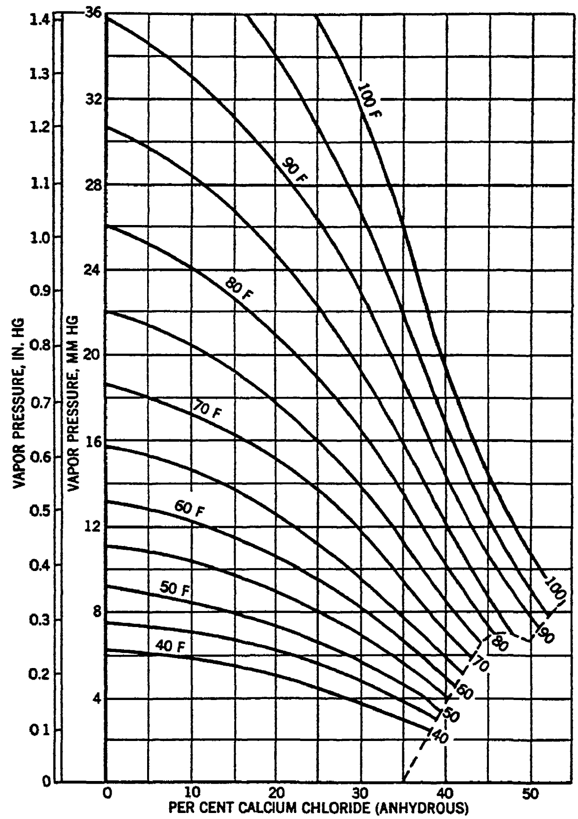

FIG. 3. TEMPERATURE—PRESSURE—CONCENTRATIONS FOR CALCIUM CHLORIDE

Example 3. Calculate the heat of vaporization of 1 lb of water from a large amount of 18 *M* lithium chloride solution at the boiling point.

Solution. The heat of boiling is equal to the heat of mixing plus the heat of boiling pure water at the same temperature. The heat of mixing from Table 1 at 18 *M* and 285.8 F is $145 - (0.186 \times 285.8) = 92$ Btu per pound. The heat of vaporization of water from steam tables at 285.8 F is 920 Btu per pound. Therefore the heat of vaporization of water from the solution is $920 + 92 = 1012$ Btu per pound.

Example 4. One thousand pounds of air per minute at 100 F dry-bulb with a dew-point of 70 F and a relative humidity of 39 per cent is passed over 18 *M* lithium chloride solution. The rate of flow of the solution is 200 gpm and the entering temperature is 80 F. The air leaves the absorber at 85 F dry-bulb and dew-point of 35 F. Calculate

by precooling the brine in the solution cooler to a predetermined temperature, which is usually below that of the air, by city, well or chilled water.

The excess water of condensation, which dilutes the brine, is removed in the solution concentrator. This is a low pressure steam heat exchanger which over-concentrates a portion of the weak liquor and returns it to the main brine reservoir for re-pumping. The concentrator operates in the manner of an evaporative condenser, whereby moisture is evaporated from the brine by the heating coils into a stream of regeneration air, taken

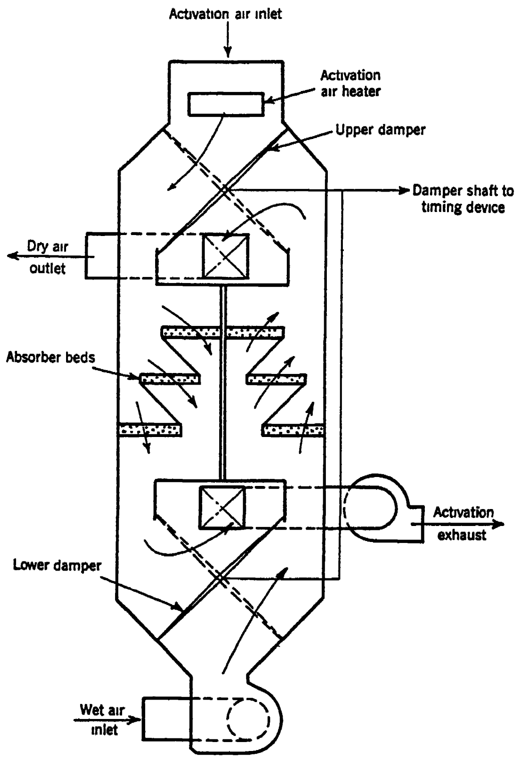

FIG. 5. SOLID ADSORBENT DEHYDRATOR—STATIONARY BED TYPE

from and rejected to the outside atmosphere. Low pressure steam is normally used for heating the brine. When it is desirable or necessary to use gas or electricity, an auxiliary low pressure steam boiler is usually added to the equipment. Concentrators operating on a simple boiler principle have not as yet been commercially practical.

It should be noted that the solution concentration phase is the reverse of the dehydration process. During concentration the aqueous vapor pressure of the solution is greater than that of the surrounding air, while during dehydration, the reverse is the case. Utilization of this principle permits winter humidification, by heating (instead of cooling) the solution pumped to the contactor. Water is thereby evaporated into, instead of

condensed out of, the conditioned air stream. This requires dilution of the brine externally to the contactor, rather than concentration.

Another type of liquid dehydrator utilizes an integral interchanger which employs the same type of solution concentrator as described for the system with the external interchanger. However, the dehydration contactor and solution cooler are combined by placing a cooling coil directly in the wet air stream. This coil provides the contacting surface between air and the warm concentrated solution which is sprayed over the cooling surface. By circulating a cooling medium through this coil, control of solution temperature (hence its vapor pressure) is accomplished directly in the air stream.

ESTIMATING OF LOADS

Where equipment is used which removes sensible and latent heat simultaneously such as a chilled water or direct expansion dehumidifier,

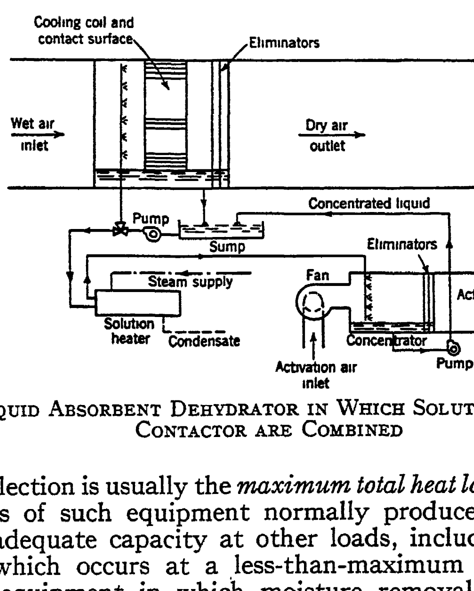

FIG. 6. LIQUID ABSORBENT DEHYDRATOR IN WHICH SOLUTION COOLER AND CONTACTOR ARE COMBINED

the basis of selection is usually the *maximum total heat load.* The operating characteristics of such equipment normally produce satisfactory dew-points with adequate capacity at other loads, including the maximum latent load which occurs at a less-than-maximum total load. With dehydration equipment in which moisture removal is achieved independently of sensible cooling, it is necessary that equipment be chosen for the maximum load of each functional element. The sensible cooler should be selected for the maximum sensible load; the dehydrator for the maximum latent load. These loads need not occur simultaneously, and in fact, rarely do.

In estimating the maximum latent heat load for comfort applications, it is considered good practice to select an outside design dew-point for the locality which is exceeded on not more than 5 per cent of the days during the season. A smaller percentage, or even the maximum dew-point recorded, may be advisable for rigorous industrial applications.

Due consideration should also be given to moisture seepage through building materials and vapor infiltration through openings. These items

become important at dew-points below 50 F and have an extreme effect upon load and equipment selection at dew-points below 35 F.

LOCATION OF EQUIPMENT

All types of dehydration equipment are, in general, applicable in one of several possible locations in the system air flow diagram. The choice of the type of equipment and its location is dependent upon the work to be performed, the capacity of the dehydration equipment to remove moisture and the type of energy available for activation or regeneration.

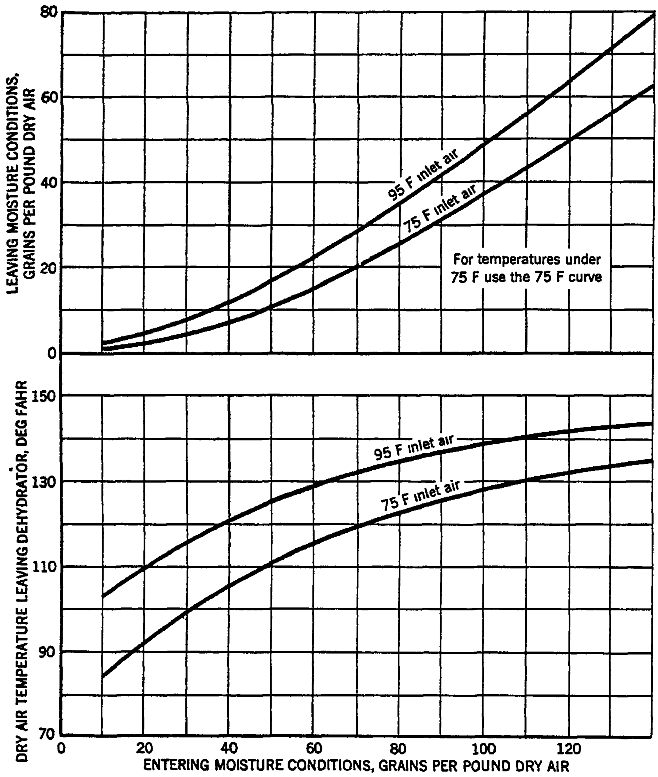

FIG. 7. SILICA GEL DEHYDRATOR PERFORMANCE DATA

Dehydration apparatus may be located: (*a*) to treat outside air only, (*b*) to treat return air only, or (*c*) to treat a mixture of outside and return air.

EQUIPMENT AUXILIARIES

Precoolers. When cold water is available, it is generally economical to use this water in a precooling coil in the outside air stream. The dehumidifying accomplished by this coil reduces the load on the dehydrator; and moreover, lowering the temperature of the inlet air to the dehydrator results in a higher dehydrator moisture removal efficiency.

Dry Air Coolers. Particularly with the solid adsorbent process, and to a lesser extent with liquid absorbents, a dry air cooler is employed to remove sensible heat from the dehydrated air whenever it leaves the dehydrator at an elevated temperature. A cooling coil using city water is usual practice, and is considered economical whenever the difference between effluent air and entering water temperatures is greater than 15 F.

Sensible Heat Coolers. Since the normal conditioning system requires sensible heat removal, auxiliary equipment may be needed for this function. This is almost always in the form of cooling surface using water, brine or direct expansion refrigerant. It is located on the leaving side of the dehydrator, but frequently treats in addition a large volume of room air which is not circulated through the dehydrator for moisture reduction.

CONTROLS

The use of dehydration equipment makes possible the use of a relatively simple control system with a humidistat or, alternatively, a wet-bulb controller, to regulate the operation of the dehydrator, and a thermostat

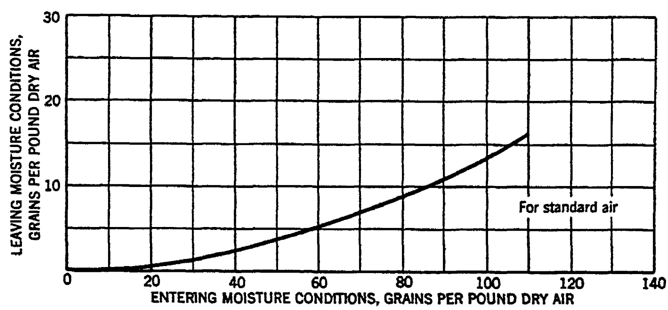

FIG. 8. ACTIVATED ALUMINA DEHYDRATOR PERFORMANCE DATA

to control the sensible cooling apparatus. Functionally, the relative humidity control may consist of one of the following:

1. *Stop—Start*—Where the humidistat starts the dehydrator on rising humidity and stops it on falling humidity.

2. *Bypass*—Where the humidistat modulates face and bypass dampers located at the wet air inlet of the dehydrator. Thus the quantity of air passing through the dehydrator is proportioned in accordance with the change in latent heat load.

3. *Vapor Pressure Control* (used with liquid absorbents)—Where the humidistat directly controls the temperature or concentration of the contacting solution, thereby matching the latent heat removal to the load requirement.

EQUIPMENT PERFORMANCE

It is recognized that, whereas the curves relating temperature and vapor pressure of the several dehydration agents (Figs. 1, 2 and 3) accurately define the equilibrium limits for these materials, these curves cannot be used for predicting performance of available equipment. This is because (*a*) the materials themselves can only be utilized efficiently within certain ranges of moisture concentration, and (*b*) the degree to

which the vapor pressure of the air being treated approaches that at the surface of the material depends upon the completeness of the contact. For this reason, actual moisture removing capacity is determined from performance curves of the several materials under practical conditions of temperature, concentration and contacting efficiency as shown in Figs. 7, 8 and 9. While these curves by no means explore all the performance possibilities, they may be considered to be representative of sound design

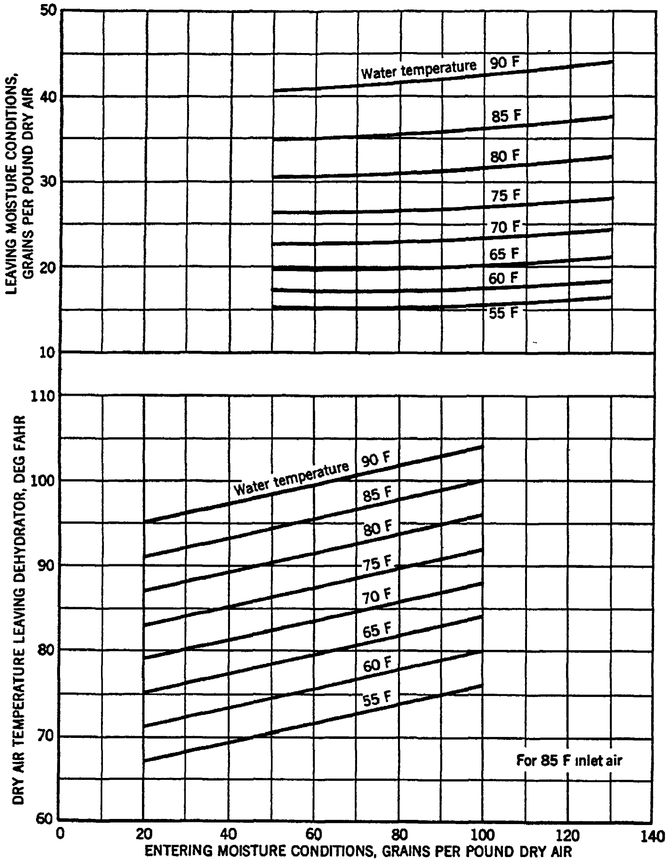

FIG. 9. LITHIUM CHLORIDE CONTACTOR PERFORMANCE DATA

and application practice. Representative data on heat input and water consumption are given in Table 4.

ECONOMICS OF DEHYDRATION

Almost all summer comfort air conditioning, as well as much industrial and commercial air conditioning, requires both of the functions of sensible and latent heat removal from air. Each of the methods of cooling and dehumidification has as its objective either the removal of sensible or latent heat, or both simultaneously. Choice of method and

medium therefore depends solely on whether (*a*) method and medium are physically able to accomplish the desired result with practical equipment, and (*b*) method and medium are justifiable economically.

Referring particularly to the problem of moisture removal, it may be stated that either dehumidification using chilled water, brine or direct expansion refrigerant, or dehydration using solid or liquid agents, is *equally practical* from the viewpoint of engineering performance for the vast majority of comfort and for a great many industrial applications;

TABLE 4. PERFORMANCE COEFFICIENTS AND WATER REQUIREMENTS OF SOLID AND LIQUID DEHYDRATION PROCESSES

Moisture Grains per Lb		Temperature Deg F		Thermal Performance Ratio[a]	Cooling Water Consumption[b]	
In	Out	In	Out		Temp. Deg F	Gpm per 1000 Btu per Hr of Latent Heat
Solid Adsorbent[c]						
130	65	85	138	0.39		
130	45	85	154	0.35		
65	30	85	118	0.28		
45	12	85	115	0.23		
Liquid Absorbent[d]						
130	65	85	99	0.45	85	0.23
130	45	85	101	0.45	85	0.44
130	45	85	87	0.41	60	0.26
65	30	85	83	0.41	60	0.64

[a]Thermal performance ratio is defined as latent heat removed from entering air divided by the heat input into the regenerator. Latent heat may be actually abstracted from system or converted to sensible heat, depending on process. No credit is given in the performance ratio for abstraction of sensible heat; where it occurs, it is considered a by-product effect. Values are approximate and, while they can be construed as typical, may vary considerably with design and economic application.

[b]Cooling water shown is that required solely to produce the latent heat removal. Additional water is required by both processes for reducing effluent temperatures below those listed. Gallons per minute shown are necessarily an economic approximation, weighing amount of surface against water consumption.

[c]Solid adsorbent based on the use of gas for activation.

[d]Liquid absorbent based on the use of steam at 12 lb per square inch gage for regeneration.

that is, their fields of application overlap. It cannot properly be stated that one method or material is superior *functionally* to another, since all have the same objective, and each is capable of attaining that objective.

For this reason, the choice of method and agent can normally be considered strictly on its economic merits. The choice of dehydration, mechanical refrigeration, natural cold water or ice, must be justified by the initial investment of available equipment, the availability and cost of prime energy sources, and the charges properly allocable to space occupied, labor of operation, maintenance, etc.

ECONOMIC COMPARISONS

It is evident that it is not possible to set forth definite rules governing the choice of dehydration equipment in preference to other methods of

dehumidification. It is only possible to state certain general conditions which tend to make dehydration favorable or unfavorable.

Dehydration tends to be favorable where:

1. Steam or gas is available at a cost substantially lower than electricity.
2. Required dry-bulb temperature is high or unimportant in comparison to maintenance of proper relative humidity.
3. Sensible cooling can be supplied by low cost city, well, or river water available at the proper temperature. For comfort conditioning, this temperature cannot normally be higher than 65 F.
4. An abnormally high room latent heat load or a large outside air latent load is encountered (such as in a dance hall, theater, restaurant, etc.).
5. Abnormally low room dew-points are required (such as 40 F or lower for some manufacturing operations).
6. Low temperature water is available but high in cost or limited in quantity.
7. In low temperature driers a complementary heat exchange can be utilized. In such cases, the sensible heat of the dry air from the dehydrator is reduced by the evaporation of moisture within the drier.

Of the factors just enumerated Item 1 is the most important influence, and if favorable, it indicates the desirability of considering dehydration. The other items are of lesser importance as criteria, but each has a direct influence in the economic considerations.

Dehydration tends to be unfavorable where:

1. Electricity is low in cost.
2. Normal comfort dew-points are required with a preponderantly sensible heat load.
3. Mechanical refrigeration is required for sensible heat removal.
4. Water temperature is too high for sensible heat removal. For comfort conditioning, this usually means water above 65 F.
5. Water is available in adequate quantity and at such temperature that it can be used directly for both sensible and latent removal, or can be further chilled more cheaply by mechanical refrigeration. For comfort conditioning, this normally means water below 55 F.

No single item just mentioned will necessarily disqualify dehydration, but will tend to require several favorable factors to make it a possibility for selection.

The previously outlined criteria are general and inclusive. When analyzed with respect to the possible fields of application, it is evident that dehydration equipment can be used, within its legitimate economic limits, for: air conditioning for human comfort, commercial cooling for food products requiring low humidities, industrial air conditioning for processes, and industrial drying. Attention is called to those particularly favorable industrial conditioning and drying applications in which the dried air can be used at effluent temperature without further treatment.

REFERENCES

Direct Evaporative Cooling for Homes in the Southwest, by A. J. Rummel (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1940, p. 621).

Chemical Dehumidification Agents, by F. R. Bichowsky (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1940, p. 627).

Chapter 25

REFRIGERATION

Mechanical Refrigeration, Characteristics of Compression System, Absorption Systems, Expansion Valves, Condensers, Evaporators and Coolers, Refrigerant Pipe Sizes, Ice Systems, Equipment Selection, Reverse Cycle

COOLING and dehumidification in air conditioning work usually requires refrigeration equipment. The localities where cold water from a natural source is at a sufficiently low temperature for comfort air conditioning are rare, and evaporative cooling is generally restricted to sections of the country where humidities are naturally low.

The important difference between the refrigeration equipment used for comfort air conditioning and that used for commercial refrigeration is the use of a relatively higher evaporator temperature. This temperature is usually above freezing in air conditioning refrigeration equipment. The higher evaporator temperature (that is high suction pressure) affects the design of the system used, and makes possible the use of systems that are not always practical for commercial refrigeration.

MECHANICAL REFRIGERATION

The fundamentals of mechanical refrigeration systems are similar, although they differ in the methods used for compression of the refrigerant vapor.

Refrigerant vapor, usually saturated or slightly superheated, is drawn into the compressor as diagrammed in Fig. 1. It is then compressed and discharged at a higher pressure to a condenser. The vapor is condensed as it contacts a heat transfer surface over which is flowing a cooling medium such as water, air or a combination of the two. The liquid refrigerant flows to the evaporator through an expansion valve which reduces its pressure and regulates its flow. In the evaporator, the refrigerant absorbs heat from the medium which is to be cooled. When this medium is water or brine, the evaporator is known as a water or brine cooler and the refrigeration system, if used for air cooling, is known as an indirect system. When the medium cooled is air, the evaporator is known as a direct expansion cooler and the system is known as a direct expansion system.

Fundamentally, the function of the system is to absorb heat at one temperature and pump it to a higher temperature, where it may be

Table 1. Properties of Ammonia

Sat. Temp F	Abs. Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		100 F Superheat		200 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
0	30.42	0.02419	9.116	42.9	611.8	0.0975	1.3352	666.8	1.4439	720.3	1.5317
2	31.92	0.02424	8.714	45.1	612.4	0.1022	1.3312	667.6	1.4400	721.2	1.5277
4	33.47	0.02430	8.333	47.2	613.0	0.1069	1.3273	668.4	1.4360	722.2	1.5236
5	34.27	0.02432	8.150	48.3	613.3	0.1092	1.3253	668.8	1.4340	722.6	1.5216
6	35.09	0.02435	7.971	49.4	613.6	0.1115	1.3234	669.3	1.4321	723.1	1.5196
8	36.77	0.02440	7.629	51.6	614.3	0.1162	1.3195	670.1	1.4281	724.1	1.5155
10	38.51	0.02446	7.304	53.8	614.9	0.1208	1.3157	670.9	1.4242	725.0	1.5115
12	40.31	0.02451	6.996	56.0	615.5	0.1254	1.3118	671.7	1.4205	725.9	1.5077
14	42.18	0.02457	6.703	58.2	616.1	0.1300	1.3081	672.5	1.4168	726.8	1.5039
16	44.12	0.02462	6.425	60.3	616.6	0.1346	1.3043	673.4	1.4130	727.8	1.5001
18	46.13	0.02468	6.161	62.5	617.2	0.1392	1.3006	674.2	1.4093	728.7	1.4963
20	48.21	0.02474	5.910	64.7	617.8	0.1437	1.2969	675.0	1.4056	729.6	1.4925
22	50.36	0.02479	5.671	66.9	618.3	0.1483	1.2933	675.8	1.4021	730.5	1.4889
24	52.59	0.02485	5.443	69.1	618.9	0.1528	1.2897	676.6	1.3985	731.4	1.4853
26	54.90	0.02491	5.227	71.3	619.4	0.1573	1.2861	677.3	1.3950	732.4	1.4816
28	57.28	0.02497	5.021	73.5	619.9	0.1618	1.2825	678.1	1.3914	733.3	1.4780
30	59.74	0.02503	4.825	75.7	620.5	0.1663	1.2790	678.9	1.3879	734.2	1.4744
32	62.29	0.02508	4.637	77.9	621.0	0.1708	1.2755	679.7	1.3846	735.1	1.4710
34	64.91	0.02514	4.459	80.1	621.5	0.1753	1.2721	680.4	1.3812	736.0	1.4676
36	67.63	0.02521	4.289	82.3	622.0	0.1797	1.2686	681.2	1.3779	736.8	1.4643
38	70.43	0.02527	4.126	84.6	622.5	0.1841	1.2652	681.9	1.3745	737.7	1.4609
39	71.87	0.02530	4.048	85.7	622.7	0.1863	1.2635	682.3	1.3729	738.2	1.4592
40	73.32	0.02533	3.971	86.8	623.0	0.1885	1.2618	682.7	1.3712	738.6	1.4575
41	74.80	0.02536	3.897	87.9	623.2	0.1908	1.2602	683.1	1.3696	739.0	1.4559
42	76.31	0.02539	3.823	89.0	623.4	0.1930	1.2585	683.4	1.3680	739.5	1.4542
44	79.38	0.02545	3.682	91.2	623.9	0.1974	1.2552	684.2	1.3648	740.4	1.4510
46	82.55	0.02551	3.547	93.5	624.4	0.2018	1.2519	684.9	1.3616	741.3	1.4477
48	85.82	0.02558	3.418	95.7	624.8	0.2062	1.2486	685.6	1.3584	742.2	1.4445
50	89.19	0.02564	3.294	97.9	625.2	0.2105	1.2453	686.4	1.3552	743.1	1.4412
52	92.66	0.02571	3.176	100.2	625.7	0.2149	1.2421	687.1	1.3521	744.0	1.4382
54	96.23	0.02577	3.063	102.4	626.1	0.2192	1.2389	687.8	1.3491	744.8	1.4351
56	99.91	0.02584	2.954	104.7	626.5	0.2236	1.2357	688.5	1.3460	745.7	1.4321
58	103.7	0.02590	2.851	106.9	626.9	0.2279	1.2325	689.2	1.3430	746.5	1.4290
60	107.6	0.02597	2.751	109.2	627.3	0.2322	1.2294	689.9	1.3399	747.4	1.4260
62	111.6	0.02604	2.656	111.5	627.7	0.2365	1.2262	690.6	1.3370	748.2	1.4231
64	115.7	0.02611	2.565	113.7	628.0	0.2408	1.2231	691.3	1.3341	749.1	1.4202
66	120.0	0.02618	2.477	116.0	628.4	0.2451	1.2201	691.9	1.3312	749.9	1.4172
68	124.3	0.02625	2.393	118.3	628.8	0.2494	1.2170	692.6	1.3283	750.8	1.4143
70	128.8	0.02632	2.312	120.5	629.1	0.2537	1.2140	693.3	1.3254	751.6	1.4114
72	133.4	0.02639	2.235	122.8	629.4	0.2579	1.2110	694.0	1.3226	752.4	1.4086
74	138.1	0.02646	2.161	125.1	629.8	0.2622	1.2080	694.6	1.3199	753.3	1.4059
76	143.0	0.02653	2.089	127.4	630.1	0.2664	1.2050	695.3	1.3171	754.1	1.4031
78	147.9	0.02661	2.021	129.7	630.4	0.2706	1.2020	695.9	1.3144	755.0	1.4004
80	153.0	0.02668	1.955	132.0	630.7	0.2749	1.1991	696.6	1.3116	755.8	1.3976
82	158.3	0.02675	1.892	134.3	631.0	0.2791	1.1962	697.2	1.3089	756.6	1.3949
84	163.7	0.02684	1.831	136.6	631.3	0.2833	1.1933	697.8	1.3063	757.4	1.3923
86	169.2	0.02691	1.772	138.9	631.5	0.2875	1.1904	698.5	1.3040	758.3	1.3896
88	174.8	0.02699	1.716	141.2	631.8	0.2917	1.1875	699.1	1.3010	759.1	1.3870
90	180.6	0.02707	1.661	143.5	632.0	0.2958	1.1846	699.7	1.2983	759.9	1.3843
92	186.6	0.02715	1.609	145.8	632.2	0.3000	1.1818	700.3	1.2957	760.7	1.3818
94	192.7	0.02723	1.559	148.2	632.5	0.3041	1.1789	700.9	1.2932	761.5	1.3793
96	198.9	0.02731	1.510	150.5	632.6	0.3083	1.1761	701.5	1.2906	762.2	1.3768
98	205.3	0.02739	1.464	152.9	632.9	0.3125	1.1733	702.1	1.2881	763.0	1.3743
100	211.9	0.02747	1.419	155.2	633.0	0.3166	1.1705	702.7	1.2855	763.8	1.3718
102	218.6	0.02756	1.375	157.6	633.2	0.3207	1.1677	703.3	1.2830	764.6	1.3693
104	225.4	0.02764	1.334	159.9	633.4	0.3248	1.1649	703.8	1.2805	765.3	1.3668
106	232.5	0.02773	1.293	162.3	633.5	0.3289	1.1621	704.3	1.2780	766.1	1.3643
108	239.7	0.02782	1.254	164.6	633.6	0.3330	1.1593	705.0	1.2755	766.9	1.3619
110	247.0	0.02790	1.217	167.0	633.7	0.3372	1.1566	705.5	1.2731	767.6	1.3596
112	254.5	0.02799	1.180	169.4	633.8	0.3413	1.1538	706.1	1.2708	768.3	1.3573
114	262.2	0.02808	1.145	171.8	633.9	0.3453	1.1510	706.6	1.2684	769.1	1.3550
116	270.1	0.02817	1.112	174.2	634.0	0.3495	1.1483	707.2	1.2661	769.8	1.3527
118	278.2	0.02827	1.079	176.6	634.0	0.3535	1.1455	707.7	1.2636	770.5	1.3503
120	286.4	0.02836	1.047	179.0	634.0	0.3576	1.1427	708.2	1.2612	771.3	1.3479
122	294.8	0.02846	1.017	181.4	634.0	0.3618	1.1400	708.6	1.2587	772.0	1.3455
124	303.4	0.02855	0.987	183.9	634.0	0.3659	1.1372	709.1	1.2563	772.8	1.3431
126	312.2	0.02865	0.958	186.3	633.9	0.3700	1.1344	709.6	1.2538	773.5	1.3407
128	321.2	0.02875	0.931	188.8	633.9	0.3741	1.1316	710.0	1.2513	774.2	1.3383

removed by an available cooling medium. In order to conserve refrigerant, virtually all refrigeration systems are completely closed and the same refrigerant is recirculated.

The fundamental heat equations (disregarding losses) which should be kept in mind are: (1) the heat absorbed in the evaporator plus the heat added to the refrigerant during compression equals the heat rejected by the condenser; (2) the heat added to the refrigerant during compression is equal to the input to the compressor shaft less the heat dissipated from the compressor to the surroundings.

In the case where the compressor is driven by an electric motor, the heat due to compression is equal to the motor input less the electrical motor losses, less the power transmission losses and less the heat dissipated from the compressor to the surroundings.

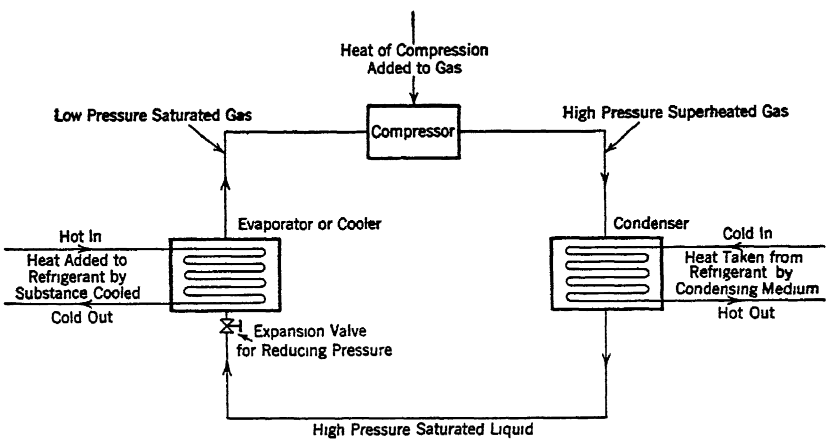

FIG. 1. MECHANICAL REFRIGERATION SYSTEM

Refrigerants

There are many substances which might be used as refrigerants in mechanical refrigeration systems, but in practice the choice is limited by a wide variety of considerations including availability, cost, safety, chemical stability and adaptability to the type of refrigerating system to be used.

In this chapter detailed consideration is limited to six substances, viz: ammonia, dichlorodifluoromethane (F_{12}), methyl chloride, carbon dioxide, monofluorotrichloromethane (F_{11}), and water, properties for each of which are given in Tables 1, 2, 3, 4, 5 and 6. Each table gives the principal physical properties of the saturated substance, and all are arranged in uniform fashion. In all except the water table, columns are included which give the heat content and entropy of the superheated vapor at two selected points. The first four refrigerants named are used in reciprocating and rotary compressors. The last two are used in centrifugal compressors. Water is also used in steam jet equipment.

TABLE 2. PROPERTIES OF DICHLORODIFLUOROMETHANE(F_{12})

Sat Temp. F	Abs. Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct	Entropy
0	23.87	0.0110	1.637	8.25	78.21	0 01869	0.17091	81.71	0.17829	85.26	0.18547
2	24 89	0.0110	1.574	8.67	78.44	0.01961	0 17075	81.94	0.17812	85.51	0.18529
4	25.96	0.0111	1.514	9.10	78.67	0 02052	0.17060	82.17	0 17795	85.76	0.18511
5	26.51	0 0111	1.485	9.32	78.79	0 02097	0.17052	82.29	0.17786	85.89	0.18502
6	27.05	0.0111	1.457	9.53	78.90	0.02143	0.17045	82.41	0.17778	86.01	0.18494
8	28.18	0.0111	1.403	9.96	79.13	0.02235	0.17030	82.66	0.17763	86 26	0.18477
10	29.35	0 0112	1.351	10.39	79.36	0.02328	0 17015	82.90	0.17747	86.51	0.18460
12	30.56	0.0112	1.301	10 82	79.59	0 02419	0.17001	83.14	0.17733	86.76	0.18444
14	31.80	0.0112	1.253	11.26	79.82	0.02510	0.16987	83.38	0.17720	87.01	0.18429
16	33 08	0.0112	1.207	11.70	80.05	0 02601	0.16974	83.61	0.17706	87.26	0 18413
18	34.40	0.0113	1 163	12.12	80.27	0 02692	0.16961	83.85	0.17693	87.51	0.18397
20	35.75	0.0113	1.121	12.55	80.49	0.02783	0.16949	84.09	0.17679	87.76	0.18382
22	37.15	0.0113	1.081	13.00	80.72	0.02873	0.16938	84.32	0.17666	88.00	0 18369
24	38.58	0.0113	1.043	13.44	80.95	0.02963	0.16926	84.55	0.17652	88.24	0.18355
26	40.07	0 0114	1.007	13.88	81.17	0.03053	0.16913	84.79	0.17639	88.49	0.18342
28	41.59	0.0114	0.973	14.32	81.39	0.03143	0.16900	85.02	0.17625	88.73	0.18328
30	43.16	0.0115	0.939	14.76	81.61	0.03233	0.16887	85.25	0.17612	88.97	0.18315
32	44.77	0.0115	0.908	15.21	81.83	0.03323	0.16876	85.48	0.17600	89.21	0.18303
34	46.42	0 0115	0.877	15.65	82.05	0.03413	0.16865	85.71	0.17589	89.45	0 18291
36	48.13	0.0116	0.848	16.10	82.27	0 03502	0 16854	85.95	0.17577	89.68	0.18280
38	49.88	0.0116	0 819	16.55	82.49	0.03591	0.16843	86.18	0.17566	89.92	0.18268
39	50.78	0 0116	0.806	16.77	82.60	0.03635	0.16838	86.29	0.17560	90.04	0.18262
40	51.68	0.0116	0.792	17.00	82.71	0.03680	0.16833	86.41	0.17554	90.16	0.18256
41	52.70	0.0116	0.779	17.23	82.82	0.03725	0.16828	86.52	0.17549	90 28	0.18251
42	53.51	0.0116	0.767	17.46	82.93	0.03770	0.16823	86.64	0.17544	90.40	0.18245
44	55.40	0 0117	0.742	17.91	83.15	0 03859	0.16813	86.86	0.17534	90.65	0.18235
46	57.35	0.0117	0.718	18.36	83.36	0 03948	0 16803	87.09	0.17525	90.89	0.18224
48	59.35	0.0117	0.695	18.82	83.57	0.04037	0.16794	87.31	0.17515	91.14	0.18214
50	61.39	0.0118	0.673	19.27	83 78	0.04126	0.16785	87.54	0.17505	91.38	0.18203
52	63.49	0.0118	0.652	19.72	83.99	0.04215	0.16776	87.76	0.17496	91.61	0.18193
54	65.63	0.0118	0.632	20.18	84.20	0.04304	0.16767	87.98	0.17486	91.83	0.18184
56	67.84	0.0119	0.612	20.64	84.41	0.04392	0.16758	88.20	0.17477	92.06	0.18174
58	70.10	0 0119	0.593	21.11	84.62	0.04480	0.16749	88.42	0.17467	92 28	0.18165
60	72.41	0.0119	0.575	21.57	84.82	0.04568	0.16741	88.64	0.17458	92.51	0.18155
62	74.77	0.0120	0.557	22.03	85.02	0 04657	0.16733	88.86	0.17450	92.74	0.18147
64	77.20	0.0120	0.540	22.49	85.22	0.04745	0 16725	89.07	0.17442	92.97	0.18139
66	79.67	0.0120	0.524	22.95	85.42	0.04833	0.16717	89.29	0.17433	93.20	0.18130
68	82.24	0.0121	0.508	23.42	85.62	0.04921	0.16709	89.50	0.17425	93.43	0.18122
70	84.82	0.0121	0.493	23.90	85.82	0.05009	0.16701	89.72	0.17417	93 66	0.18114
72	87.50	0.0121	0.479	24 37	86.02	0.05097	0.16693	89.93	0.17409	93.99	0.18106
74	90.20	0.0122	0.464	24.84	86.22	0 05185	0.16685	90.14	0.17402	94.12	0.18098
76	93.00	0.0122	0.451	25.32	86.42	0.05272	0.16677	90.36	0.17394	94.34	0 18091
78	95.85	0 0123	0.438	25.80	86.61	0.05359	0.16669	90.57	0.17387	94 57	0 18083
80	98.76	0.0123	0.425	26.28	86 80	0.05446	0.16662	90.78	0.17379	94.80	0.18075
82	101.70	0.0123	0.413	26.76	86.99	0.05534	0.16655	90.98	0.17372	95.01	0.18068
84	104.8	0.0124	0.401	27.24	87.18	0.05621	0.16648	91.18	0.17365	95.22	0.18061
86	107.9	0.0124	0.389	27.72	87.37	0.05708	0.16640	91.37	0.17358	95.44	0.18054
88	111.1	0.0124	0 378	28.21	87.56	0.05795	0.16632	91.57	0.17351	95.65	0.18047
90	114.3	0.0125	0.368	28.70	87.74	0.05882	0.16624	91.77	0.17344	95.86	0.18040
92	117.7	0.0125	0.357	29.19	87.92	0.05969	0.16616	91.97	0.17337	96.07	0.18033
94	121.0	0.0126	0.347	29.68	88.10	0.06056	0.16608	92.16	0.17330	96.28	0.18026
96	124.5	0 0126	0.338	30.18	88.28	0.06143	0.16600	92.36	0 17322	96.50	0.18018
98	128.0	0.0126	0.328	30.67	88.45	0.06230	0.16592	92.55	0.17315	96.71	0.18011
100	131.6	0.0127	0.319	31.16	88.62	0.06316	0.16584	92.75	0.17308	96 92	0.18004
102	135.3	0.0127	0.310	31.65	88.79	0.06403	0.16576	92.93	0.17301	97.12	0.17998
104	139.0	0.0128	0.302	32.15	88.95	0.06490	0.16568	93.11	0.17294	97.32	0.17993
106	142.8	0.0128	0.293	32.65	89.11	0.06577	0.16560	93.30	0.17288	97.53	0.17987
108	146.8	0.0129	0.285	33.15	89.27	0.06663	0.16551	93.48	0.17281	97 73	0.17982
110	150.7	0.0129	0.277	33.65	89.43	0.06749	0.16542	93.66	0.17274	97.93	0.17976
112	154.8	0.0130	0.269	34.15	89.58	0.06836	0.16533	93.82	0.17266	98 11	0.17969
114	158.9	0.0130	0.262	34.65	89.73	0.06922	0.16524	93.98	0.17258	98.29	0.17961
116	163.1	0.0131	0.254	35.15	89.87	0.07008	0.16515	94.15	0.17249	98.48	0.17954
118	167.4	0.0131	0.247	35.65	90.01	0.07094	0.16505	94.31	0.17241	98.66	0.17946
120	171.8	0.0132	0.240	36.16	90.15	0.07180	0.16495	94.47	0.17233	98.84	0 17939
122	176.2	0.0132	0.233	36.66	90 28	0.07266	0.16484	94.63	0.17224	99.01	0.17931
124	180.8	0.0133	0 227	37.16	90.40	0.07352	0.16473	94.78	0.17215	99.18	0.17922
126	185.4	0.0133	0.220	37.67	90.52	0.07437	0.16462	94.94	0.17206	99.35	0.17914
128	190.1	0.0134	0 214	38.18	90.64	0.07522	0.16450	95.09	0.17196	99.53	0.17906
130	194.9	0.0134	0.208	38.69	90.76	0.07607	0.16438	95.25	0.17186	99.70	0.17897
132	199.8	0.0135	0.202	39.19	90.86	0.07691	0.16425	95.41	0.17176	99.87	0.17889
134	204 8	0.0135	0.196	39.70	90.96	0.07775	0.16411	95.56	0.17166	100.04	0.17881
136	209.9	0.0136	0.191	40.21	91.06	0.07858	0.16396	95.72	0.17156	100.22	0.17873
138	215.0	0.0137	0.185	40.72	91.15	0.07941	0.16380	95.87	0.17145	100.39	0.17864
140	220.2	0.0138	0.180	41.24	91.24	0.08024	0.16363	96.03	0.17134	100.56	0.17856

Table 3. Properties of Methyl Chloride

Sat. Temp F	Abs. Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		100 F Superheat		200 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
0	18.73	0.0162	5.052	14.4	192.4	0.0328	0.4197	215.6	0.467	237.2	0.507
2	19.60	0.0162	4.856	15.1	193.1	0.0344	0.4196	216.2	0.466	237.7	0.505
4	20.47	0.0163	4.661	15.8	193.8	0.0360	0.4195	216.7	0.465	238.2	0.504
5	20.91	0.0163	4.563	16.2	194.1	0.0368	0.4195	217.0	0.464	238.5	0.503
6	21.39	0.0163	4.476	16.6	194.4	0.0376	0.4194	217.3	0.464	238.8	0.502
8	22.34	0.0164	4.303	17.3	195.1	0.0391	0.4193	217.9	0.463	239.4	0.501
10	23.30	0.0164	4.129	18.1	195.8	0.0407	0.4192	218.5	0.463	240.0	0.500
12	24.38	0.0164	3.984	18.8	196.3	0.0423	0.4184	219.0	0.462	240.5	0.499
14	25.46	0.0164	3.839	19.6	196.7	0.0439	0.4176	219.5	0.462	241.0	0.498
16	26.55	0.0165	3.693	20.3	197.2	0.0454	0.4168	220.0	0.461	241.5	0.498
18	27.63	0.0165	3.548	21.1	197.6	0.0472	0.4160	220.5	0.461	242.0	0.497
20	28.71	0.0166	3.403	21.8	198.1	0.0486	0.4152	221.0	0.460	242.5	0.496
22	29.98	0.0166	3.288	22.5	198.5	0.0501	0.4148	221.5	0.459	243.0	0.495
24	31.25	0.0166	3.172	23.3	198.9	0.0516	0.4143	222.0	0.459	243.6	0.495
26	32.53	0.0167	3.057	24.0	199.3	0.0532	0.4139	222.4	0.458	244.1	0.494
28	33.80	0.0167	2.941	24.8	199.7	0.0547	0.4134	222.9	0.458	244.7	0.494
30	35.07	0.0168	2.826	25.5	200.1	0.0562	0.4130	223.4	0.457	245.2	0.493
32	36.55	0.0168	2.734	26.2	200.5	0.0577	0.4124	223.9	0.456	245.7	0.492
34	38.03	0.0169	2.642	27.0	200.9	0.0592	0.4118	224.3	0.455	246.2	0.492
36	39.51	0.0169	2.549	27.7	201.4	0.0607	0.4111	224.8	0.455	246.7	0.491
38	40.99	0.0169	2.457	28.5	201.8	0.0622	0.4105	225.2	0.454	247.2	0.491
39	41.73	0.0170	2.411	28.8	202.0	0.0629	0.4102	225.5	0.453	247.4	0.490
40	42.47	0.0170	2.365	29.2	202.2	0.0637	0.4099	225.7	0.453	247.7	0.490
41	43.33	0.0170	2.328	29.6	202.4	0.0644	0.4096	225.9	0.453	248.0	0.490
42	44.18	0.0171	2.290	29.9	202.6	0.0651	0.4093	226.1	0.452	248.3	0.489
44	45.89	0.0171	2.216	30.7	203.0	0.0666	0.4087	226.6	0.451	248.8	0.489
46	47.61	0.0171	2.141	31.4	203.3	0.0680	0.4081	227.0	0.451	249.4	0.488
48	49.32	0.0172	2.067	32.2	203.7	0.0695	0.4075	227.5	0.450	249.9	0.488
50	51.03	0.0172	1.992	32.9	204.1	0.0709	0.4069	227.9	0.449	250.5	0.487
52	53.00	0.0172	1.931	33.7	204.4	0.0724	0.4063	228.2	0.448	251.0	0.486
54	54.97	0.0173	1.870	34.4	204.7	0.0739	0.4056	228.6	0.448	251.5	0.486
56	56.94	0.0173	1.810	35.2	205.1	0.0754	0.4050	228.9	0.447	252.0	0.485
58	58.91	0.0173	1.749	35.9	205.4	0.0769	0.4043	229.3	0.447	252.5	0.485
60	60.88	0.0174	1.688	36.7	205.7	0.0784	0.4037	229.6	0.446	253.0	0.484
62	63.13	0.0174	1.638	37.4	206.0	0.0798	0.4030	229.9	0.445	253.5	0.483
64	65.37	0.0174	1.588	38.2	206.3	0.0812	0.4024	230.3	0.444	254.0	0.483
66	67.62	0.0175	1.539	38.9	206.6	0.0827	0.4017	230.6	0.443	254.5	0.482
68	69.86	0.0175	1.489	39.7	206.9	0.0841	0.4011	231.0	0.442	255.0	0.482
70	72.11	0.0176	1.439	40.4	207.2	0.0855	0.4004	231.3	0.441	255.5	0.481
72	74.66	0.0176	1.398	41.1	207.5	0.0869	0.3998	231.6	0.440	256.0	0.480
74	77.21	0.0177	1.357	41.9	207.7	0.0883	0.3992	232.0	0.439	256.5	0.480
76	79.76	0.0177	1.315	42.6	208.0	0.0898	0.3985	232.3	0.439	256.9	0.479
78	82.31	0.0178	1.274	43.4	208.2	0.0912	0.3979	232.7	0.438	257.4	0.479
80	84.86	0.0178	1.233	44.1	208.5	0.0926	0.3973	233.0	0.437	257.9	0.478
82	87.74	0.0178	1.199	44.8	208.7	0.0940	0.3967	233.3	0.436	258.4	0.478
84	90.62	0.0179	1.165	45.6	209.0	0.0953	0.3960	233.6	0.435	258.9	0.477
86	93.50	0.0179	1.130	46.3	209.2	0.0967	0.3954	233.9	0.435	259.4	0.477
88	96.38	0.0180	1.096	47.1	209.5	0.0980	0.3947	234.2	0.434	259.9	0.476
90	99.26	0.0180	1.062	47.8	209.7	0.0994	0.3941	234.5	0.433	260.4	0.476
92	102.49	0.0180	1.033	48.6	209.9	0.1008	0.3935	234.8	0.433	260.8	0.476
94	105.72	0.0181	1.005	49.3	210.2	0.1022	0.3929	235.1	0.432	261.2	0.475
96	108.94	0.0181	0.9764	50.1	210.4	0.1035	0.3922	235.4	0.432	261.6	0.475
98	112.17	0.0182	0.9478	50.8	210.7	0.1049	0.3916	235.7	0.431	262.0	0.474
100	115.40	0.0182	0.9193	51.6	210.9	0.1063	0.3910	236.0	0.431	262.4	0.474
102	119.00	0.0183	0.8952	52.3	211.1	0.1076	0.3903	236.4	0.430	262.8	0.474
104	122.60	0.0183	0.8712	53.1	211.3	0.1090	0.3897	236.8	0.430	263.2	0.473
106	126.20	0.0184	0.8471	53.8	211.4	0.1103	0.3890	237.1	0.429	263.5	0.473
108	129.80	0.0184	0.8231	54.6	211.6	0.1117	0.3884	237.5	0.429	263.9	0.472
110	133.40	0.0185	0.7990	55.3	211.8	0.1130	0.3877	237.9	0.428	264.3	0.472
112	137.42	0.0185	0.7786	56.1	212.0	0.1144	0.3871	238.1	0.427	264.6	0.471
114	141.44	0.0185	0.7583	56.8	212.2	0.1157	0.3864	238.3	0.427	264.8	0.470
116	145.46	0.0186	0.7379	57.6	212.4	0.1171	0.3858	238.6	0.426	265.1	0.470
118	149.48	0.0186	0.7176	58.3	212.6	0.1184	0.3851	238.8	0.426	265.3	0.469
120	153.50	0.0187	0.6972	59.1	212.8	0.1198	0.3845	239.0	0.425	265.6	0.468

TABLE 4. PROPERTIES OF CARBON DIOXIDE

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		50 F Superheat		100 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
0	305.5	0.01570	0.29040	18.8	138.9	0.0418	0.3024	153.7	0.3342	167.5	0.3612
2	315.9	0.01579	0.28030	19.8	138.8	0.0440	0.3014	153.7	0.3330	167.6	0.3600
4	326.5	0.01588	0.27070	20.8	138.8	0.0461	0.3005	153.7	0.3318	167.7	0.3588
5	332.0	0.01592	0.26610	21.3	138.8	0.0472	0.3000	153.7	0.3312	167.7	0.3582
6	337.4	0.01596	0.26140	21.8	138.7	0.0483	0.2994	153.7	0.3306	167.8	0.3576
8	348.7	0.01605	0.25260	22.9	138.7	0.0504	0.2982	153.7	0.3293	167.9	0.3563
10	360.2	0.01614	0.24370	24.0	138.7	0.0526	0.2970	153.7	0.3281	168.0	0.3550
12	371.9	0.01623	0.23540	25.0	138.6	0.0548	0.2958	153.7	0.3270	168.1	0.3538
14	383.9	0.01632	0.22740	26.1	138.6	0.0571	0.2946	153.7	0.3259	168.2	0.3526
16	396.2	0.01642	0.21970	27.2	138.5	0.0593	0.2933	153.7	0.3249	168.3	0.3513
18	408.9	0.01652	0.21210	28.3	138.4	0.0616	0.2921	153.7	0.3238	168.5	0.3501
20	421.8	0.01663	0.20490	29.4	138.3	0.0638	0.2909	153.7	0.3227	168.6	0.3489
22	434.0	0.01673	0.19790	30.5	138.2	0.0662	0.2897	153.7	0.3214	168.7	0.3479
24	448.4	0.01684	0.19120	31.7	138.1	0.0686	0.2885	153.7	0.3202	168.8	0.3470
26	462.2	0.01695	0.18460	32.9	138.0	0.0710	0.2873	153.7	0.3189	168.9	0.3460
28	476.3	0.01707	0.17830	34.1	137.9	0.0734	0.2861	153.7	0.3177	169.0	0.3451
30	490.8	0.01719	0.17220	35.4	137.8	0.0758	0.2849	153.7	0.3164	169.1	0.3441
32	505.5	0.01731	0.16630	36.7	137.7	0.0781	0.2834	153.7	0.3158	169.2	0.3431
34	522.6	0.01744	0.16030	37.9	137.4	0.0804	0.2820	153.7	0.3151	169.3	0.3421
36	536.0	0.01759	0.15500	39.1	137.2	0.0828	0.2805	153.7	0.3145	169.4	0.3411
38	551.7	0.01773	0.14960	40.4	136.9	0.0851	0.2791	153.7	0.3138	169.5	0.3401
39	559.7	0.01780	0.14700	41.0	136.8	0.0862	0.2783	153.7	0.3135	169.5	0.3396
40	567.8	0.01787	0.14440	41.7	136.7	0.0874	0.2776	153.7	0.3132	169.6	0.3391
41	576.0	0.01794	0.14185	42.3	136.5	0.0887	0.2768	153.7	0.3127	169.6	0.3386
42	584.3	0.01801	0.13930	42.9	136.3	0.0899	0.2761	153.7	0.3122	169.7	0.3381
44	601.1	0.01817	0.13440	44.3	136.1	0.0924	0.2745	153.7	0.3112	169.8	0.3371
46	618.2	0.01834	0.12970	45.6	135.7	0.0950	0.2730	153.7	0.3101	169.9	0.3362
48	635.7	0.01851	0.12500	47.0	135.4	0.0975	0.2714	153.7	0.3091	170.0	0.3352
50	653.6	0.01868	0.12050	48.4	135.0	0.1000	0.2699	153.7	0.3081	170.1	0.3342
52	671.9	0.01887	0.11610	49.8	134.5	0.1027	0.2681	153.7	0.3069	170.2	0.3333
54	690.6	0.01906	0.11170	51.2	133.9	0.1054	0.2663	153.7	0.3057	170.3	0.3324
56	709.5	0.01927	0.10750	52.6	133.4	0.1081	0.2644	153.7	0.3046	170.5	0.3315
58	728.8	0.01948	0.10340	54.0	132.7	0.1108	0.2626	153.7	0.3034	170.6	0.3306
60	748.6	0.01970	0.09940	55.5	132.1	0.1135	0.2608	153.7	0.3022	170.7	0.3297
62	769.0	0.01995	0.09545	57.0	131.3	0.1164	0.2584	153.7	0.3012	170.8	0.3289
64	789.4	0.02020	0.09180	58.6	130.6	0.1194	0.2560	153.7	0.3002	170.9	0.3281
66	810.3	0.02048	0.08800	60.2	129.7	0.1223	0.2535	153.7	0.2991	171.0	0.3273
68	831.6	0.02079	0.08422	61.9	128.7	0.1253	0.2511	153.7	0.2981	171.1	0.3265
70	853.4	0.02112	0.08040	63.7	127.5	0.1282	0.2487	153.7	0.2971	171.2	0.3257
72	875.8	0.02152	0.07654	65.5	126.0	0.1321	0.2450	153.7	0.2962	171.3	0.3250
74	898.2	0.02192	0.07269	67.3	124.5	0.1360	0.2414	153.7	0.2953	171.4	0.3242
76	921.3	0.02242	0.06875	69.4	122.8	0.1398	0.2377	153.7	0.2945	171.5	0.3235
78	944.8	0.02300	0.06473	71.6	120.9	0.1437	0.2341	153.7	0.2936	171.6	0.3227
80	968.7	0.02370	0.06064	73.9	118.7	0.1476	0.2304	153.7	0.2927	171.7	0.3220
82	993.0	0.02456	0.05648	76.4	116.6	0.1578	0.2195	153.7	0.2920	173.8	0.3215
84	1017.7	0.02553	0.05223	79.4	113.9	0.1679	0.2087	153.7	0.2914	176.0	0.3209
86	1043.0	0.02686	0.04789	83.3	110.4	0.1781	0.1978	153.7	0.2907	178.2	0.3204
87.8	1069.9	0.03454	0.03454	97.0	97.0	0.1880	0.1880	153.7	0.2901	180.1	0.3199

Types of Compressors

There are many different types of compressors, using various refrigerants. Each type has its advantages for its particular application, and those generally used for air conditioning are of the following types:

1. Reciprocating compressors (commonly referred to as piston type).
2. Centrifugal compressors.
3. Steam jet.

Reciprocating compressors are available in a wide range of sizes and types. Any of a number of refrigerants, including dichlorodifluoromethane

(F_{12}), methyl chloride, ammonia, carbon dioxide, and sulphur dioxide may be used in reciprocating machines. The first of these is used extensively in direct expansion systems of comfort air conditioning.

Compressors may be classified into two general types, (*a*) open type, (*b*) enclosed type. If the driving mechanism is external to the compressor, then the shaft must be brought out through the crankcase and a shaft seal or stuffing box must be used to prevent escape of the refrigerant. This type of compressor is known as an open-type compressor. When the driving mechanism is located within the crankcase of the compressor in such a way as to avoid the necessity of a shaft seal, the compressor is known as the completely enclosed or hermetically sealed type.

Open type compressors may be further classified as belt driven and directly connected. A great number of direct-driven units are now being used which generally operate at higher rotational speeds than the belt-driven type.

The present tendency is toward forced lubrication of the bearings of compressors by means of an oil pump driven from the crankshaft, although there are many splash lubricated compressors on the market. The chief advantages of the forced lubricated compressor are that the lubrication system requires less energy for its operation than the splash type, the oil can be easily filtered before it enters the bearings, and less oil is usually required.

The compressor capacity must be selected for and matched to the maximum load for the installation on which it is to be used. Air-conditioning loads, however, vary over a wide range, and a wide fluctuation in air conditions may result during periods of light load if on-and-off control of full compressor capacity is used. To prevent such undesirable fluctuation, several methods are employed to vary the capacity of reciprocating compressors, such as:

1. By-passing one or more cylinders, of a multi-cylinder compressor, from discharge to suction.
2. Rendering the suction valves of one or more cylinders of a multi-cylinder compressor inoperative. This is usually accomplished by depressing the suction valves.
3. Varying the speed of the compressor, usually by using variable speed or two-speed electric motors.
4. Using clearance pockets to control the quantity of refrigerant pumped.
5. Restricting the suction inlet to one or more of the cylinders of a multi-cylinder compressor either by an automatic modulating valve or by an *on* and *off* valve.

All of these methods, with the exception of variable speed, result in a slightly lowered overall compressor efficiency when in use, since the mechanical losses remain constant whereas the quantity of refrigerant pumped is lowered.

Centrifugal compressors are used with very low pressure refrigerants; usually both evaporator and condenser work below atmospheric pressure. Water and monofluorotrichloromethane (F_{11}) are the refrigerants commonly used in centrifugal machines.

Compression of the refrigerant is accomplished by means of centrifugal force; therefore, this type of compressor is inherently suitable for large volumes of refrigerant at low pressure differentials. Two or more stages are usually required and high speeds are necessary to obtain good efficiency.

Table 5. Properties of Monofluorotrichloromethane (F_{11})

Sat Temp. F	Abs. Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht Ct.	Entropy	Ht. Ct.	Entropy
0	2.59	0.01020	13.700	7.81	90.4	0.0178	0.1975	93.9	0.2049	97.4	0.2120
5	2.96	0.01024	12.100	8.81	91.2	0.0200	0.1974	94.7	0.2047	98.2	0.2117
10	3.38	0.01028	10.700	9.82	92.0	0.0222	0.1973	95.5	0.2045	99.0	0.2114
15	3.85	0.01032	9.530	10.80	92.8	0.0243	0.1971	96.3	0.2043	99.8	0.2111
20	4.36	0.01036	8.490	11.90	93.7	0.0264	0.1970	97.2	0.2041	100.7	0.2109
25	4.94	0.01040	7.580	12.90	94.5	0.0286	0.1969	98.0	0.2039	101.5	0.2107
30	5.57	0.01045	6.770	13.90	95.3	0.0307	0.1969	98.8	0.2038	102.3	0.2105
35	6.27	0.01049	6.080	14.90	96.1	0.0328	0.1968	99.6	0.2037	103.1	0.2103
40	7.03	0.01053	5.460	16.00	96.8	0.0349	0.1968	100.3	0.2036	103.8	0.2101
45	7.88	0.01057	4.920	17.00	97.6	0.0370	0.1967	101.1	0.2035	104.6	0.2099
50	8.79	0.01062	4.440	18.10	98.4	0.0391	0.1967	101.9	0.2034	105.4	0.2098
55	9.80	0.01066	4.020	19.10	99.2	0.0412	0.1967	102.7	0.2033	106.2	0.2097
60	10.90	0.01071	3.640	20.20	100.0	0.0432	0.1967	103.5	0.2033	107.0	0.2096
65	12.10	0.01076	3.300	21.30	100.8	0.0453	0.1967	104.3	0.2032	107.8	0.2094
70	13.40	0.01081	3.000	22.40	101.5	0.0473	0.1967	105.0	0.2032	108.5	0.2093
75	14.80	0.01086	2.740	23.50	102.2	0.0493	0.1967	105.7	0.2031	109.2	0.2092
80	16.30	0.01091	2.500	24.50	102.9	0.0513	0.1966	106.4	0.2030	109.9	0.2090
85	17.90	0.01096	2.280	25.60	103.6	0.0533	0.1966	107.1	0.2029	110.6	0.2089
90	19.70	0.01101	2.090	26.70	104.4	0.0553	0.1966	107.9	0.2028	111.4	0.2088
95	21.60	0.01106	1.918	27.80	105.1	0.0573	0.1966	108.6	0.2028	112.1	0.2087
100	23.60	0.01111	1.761	28.90	105.7	0.0593	0.1965	109.2	0.2027	112.7	0.2085
105	25.90	0.01116	1.620	30.10	106.4	0.0613	0.1965	109.9	0.2026	113.4	0.2084

Table 6. Properties of Water

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From +32 F							
				Heat Content		Entropy		50 F Superheat		100 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
32	0.0887	0.01602	3296.0	0.00	1073.0	0.0000	2.1826	1096.9	2.2277	1120.8	2.2688
35	0.1000	0.01602	2941.0	3.02	1074.4	0.0062	2.1724	1098.3	2.2172	1122.2	2.2581
40	0.1217	0.01602	2441.0	8.05	1076.8	0.0163	2.1555	1100.6	2.2000	1124.5	2.2406
45	0.1475	0.01602	2034.0	13.07	1079.2	0.0262	2.1390	1102.9	2.1832	1126.7	2.2234
50	0.1780	0.01602	1702.0	18.08	1081.5	0.0361	2.1230	1105.2	2.1667	1129.0	2.2066
55	0.2140	0.01603	1430.0	23.08	1083.9	0.0459	2.1073	1107.5	2.1506	1131.3	2.1902
60	0.2561	0.01603	1206.0	28.08	1086.2	0.0556	2.0920	1109.8	2.1349	1133.5	2.1742
65	0.3054	0.01604	1021.0	33.08	1088.6	0.0652	2.0771	1112.2	2.1196	1135.8	2.1585
70	0.3628	0.01605	868.0	38.07	1090.9	0.0746	2.0625	1114.5	2.1046	1138.1	2.1432
75	0.4295	0.01606	740.0	43.06	1093.2	0.0840	2.0483	1116.7	2.0900	1140.3	2.1283
80	0.507	0.01607	632.9	48.05	1095.5	0.0933	2.0344	1119.0	2.0758	1142.5	2.1138
85	0.596	0.01609	543.3	53.04	1097.8	0.1025	2.0208	1121.2	2.0619	1144.7	2.0996
90	0.698	0.01610	467.9	58.03	1100.0	0.1116	2.0075	1123.4	2.0483	1146.8	2.0857
95	0.815	0.01612	404.2	63.01	1102.3	0.1206	1.9946	1125.6	2.0350	1148.9	2.0721
100	0.949	0.01613	350.3	68.00	1104.6	0.1296	1.9819	1127.9	2.0220	1151.1	2.0588
105	1.101	0.01615	304.4	72.98	1106.8	0.1384	1.9695	1130.2	2.0093	1153.2	2.0458

For properties of steam at high temperatures, see Table 8, Chapter 1.

The evaporator is usually constructed as an integral part of the centrifugal type condensing unit, to chill water which is then circulated to the air conditioning system. This is done because it would not be economical to pipe these large volumes of refrigerant any distance.

Centrifugal compressors like reciprocating compressors can be divided into two general types, open and enclosed. In general, the open type compressor is geared to the driving mechanism, and operates at higher speed than the driving motor or turbine. A modern completely enclosed direct-driven, centrifugal compressor is illustrated in Fig. 2.

The compressor capacity can be varied by controlling the condensing pressure. This is accomplished by regulating the quantity and temperature of the condenser cooling water. The capacity falls off with increasing condensing pressure. Centrifugal compressors are seldom

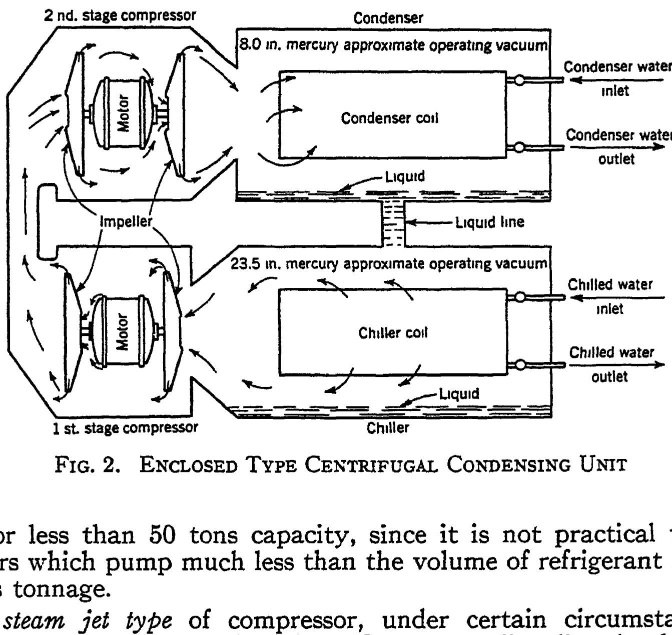

FIG. 2. ENCLOSED TYPE CENTRIFUGAL CONDENSING UNIT

built for less than 50 tons capacity, since it is not practical to make impellers which pump much less than the volume of refrigerant required for this tonnage.

The steam jet type of compressor, under certain circumstances, is desirable for use in air conditioning. Steam supplies directly the power used for compressing the refrigerant, thus eliminating the losses connected with other methods of supplying energy. As the compression ratio between the evaporator and condenser under normal circumstances is large, the mechanical efficiency of the equipment is somewhat lower than that of the positive mechanical type compressor. The condensing water requirements are considerably greater, as both the refrigerant and the impelling steam must be condensed.

The steam jet system functions on the principle that water under high vacuum will vaporize at low temperatures. Steam jet boosters or compressors of the type commonly used in power plants for various processes will produce the necessary low absolute pressure to cause evaporation of the water.

A diagrammatic representation of a typical steam ejector water cooling system is shown in Fig. 3. The figures correspond to an average representative system. The water to be cooled enters the evaporator and is cooled to a temperature corresponding to the vacuum maintained. Because of the high vacuum, a small amount of the water introduced in the evaporator is flashed into steam. As this requires heat, and the only source of heat is the rest of the water in the evaporator tank, this other water is almost instantly cooled to a temperature corresponding to the boiling point determined by the vacuum maintained. The amount of water flashed into steam is a small percentage of the total water circulated through the evaporator, amounting to approximately 11 lb per hour per ton of refrigeration developed. The remainder of the water at the

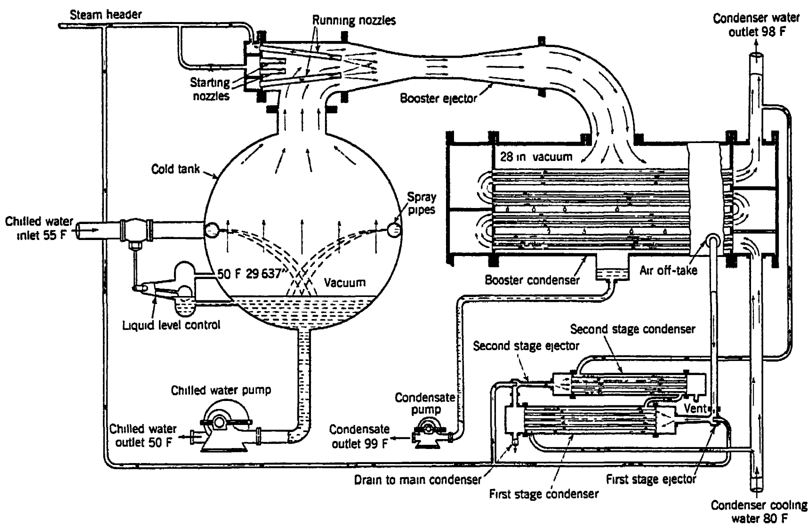

FIG. 3. DIAGRAMMATIC ARRANGEMENT OF STEAM JET VACUUM COOLING UNIT

desired low temperature is pumped out of the evaporator and used at the point where it is required.

The ejector compresses the vapor which has been flashed in the evaporator, plus any entrained air taken from the circulated water, to a somewhat higher absolute pressure and the vapor and air mix with the impelling steam on the discharge side of the jet. The total mixture then passes from the ejector into the condenser.

The slight amount of air which may be entrained in the cooled water is removed by a small secondary ejector which raises the pressure sufficiently so that the air can be discharged to the atmosphere. A secondary condenser is necessary to condense the steam in the secondary jet.

Although steam jet vacuum cooling units have been built for as small as 5 to 6 tons capacity, a single booster of smaller than 15 tons capacity is difficult to build. They can readily be built for steam pressures of from

5 to 200 lb per square inch and condenser water temperatures as high as 90 F. The steam consumption in pounds per hour per ton of refrigeration increases rapidly as the booster steam pressure is lowered. For example, the lowering of the booster steam pressure from 200 to 90 lb per square inch results in an increase in steam consumption of approximately 5 per cent whereas a further decrease in booster steam pressure to 10 lb per square inch increases the steam consumption by approximately 72 per cent over that required at 200 lb per square inch.

The capacity of a steam jet system is usually controlled by controlling the number of boosters in use since the unit usually has several boosters operating on the same evaporator. Usually one booster is automatically

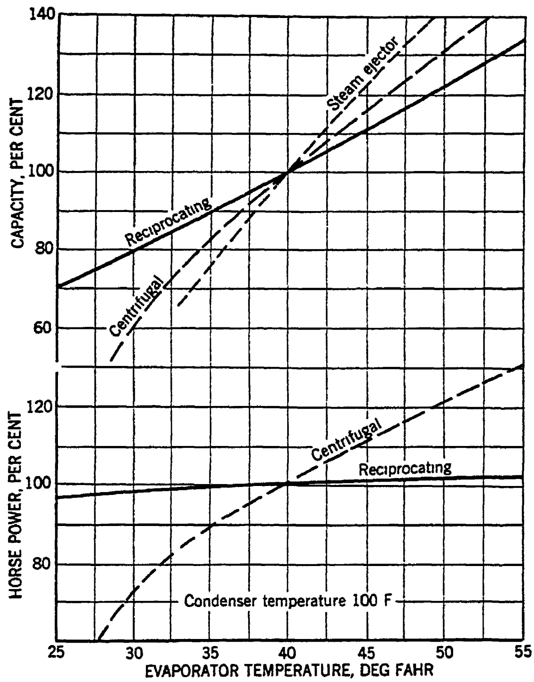

FIG. 4. PERFORMANCE CHARACTERISTICS OF COMPRESSION REFRIGERATION MACHINES AT CONSTANT SPEED

controlled whereas the others are manually operated. The capacity is dependent, as for all compressors, upon the evaporator temperature, or in other words, the suction pressure. For example, the capacity is lowered approximately 17 per cent if the evaporator or chilled water temperature is lowered from 50 to 45 F. The capacity therefore can be controlled to some extent by regulating the evaporator temperature.

CHARACTERISTICS OF COMPRESSION SYSTEMS

The various types of compression systems have quite different characteristics of capacity and power with varying evaporator and condenser temperatures, as may be noted from curves in Figs. 4 and 5.

From Fig. 5 it may be observed that power requirements for the centrifugal compressor increase much more rapidly than for the reciprocating compressor with increase in evaporator temperature. Similarly, the capacities of the steam ejector and centrifugal compressors increase more rapidly than those of the reciprocating compressor with increase in evaporator temperature. Thus, both the steam jet and centrifugal machines tend to be more self-regulating than the reciprocating. It is also evident from Fig. 5 that the steam jet equipment is best suited for operation at high evaporator temperatures.

The effect of condenser temperature upon the power and capacity of the different types of compressors is shown in Fig. 6. It may be noted that the power required by the reciprocating compressor increases rapidly

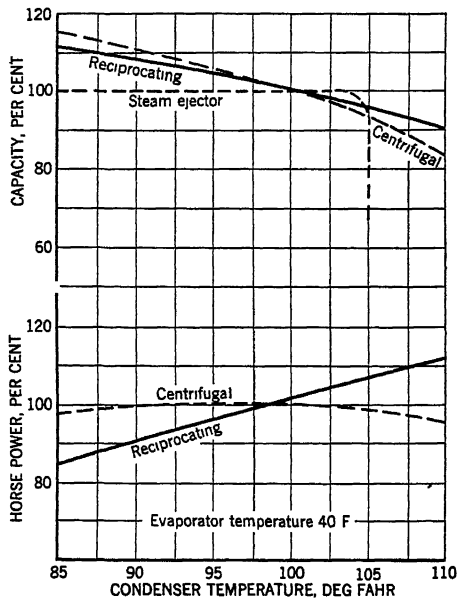

FIG. 5. PERFORMANCE CHARACTERISTICS OF COMPRESSION REFRIGERATION MACHINES AT CONSTANT SPEED

with increase in condenser temperature, while the power curve for the centrifugal compressor is relatively flat. It is also evident that the capacity of the steam jet compressor is independent of condenser temperature until a certain point is reached where it drops to zero. As previously stated, steam jet equipment requires more condensing water than other types of compression systems. Consequently, steam jet systems are well suited to those applications where condensing water is cheap, or where condensing water is rather high in temperature.

ABSORPTION SYSTEMS

The fundamental rule governing the absorption (in a closed system) of a gas by a liquid is Raoult's Law, which states that at any given tem-

perature the ratio of the partial pressure of a volatile component in a solution to the vapor pressure of the pure component at the same temperature is equal to its mol fraction in the solution. The mol fraction in turn is equal to the number of mols of substance divided by the total number of mols present. The number of mols in a given weight of a compound is equal to the weight divided by the molecular weight.

This law applies strictly, only to what is known as an ideal solution, that is, one in which the inter-molecular forces between the substances present in the solution are equal. Actually, no such solutions exist, so that deviations from Raoult's Law are always found in practice. The deviation is called positive when the observed pressure is greater than that calculated from Raoult's Law, while the term negative deviation refers to the opposite case. Negative deviations are found wherever chemical attraction exists between the solvent and the solute. Positive

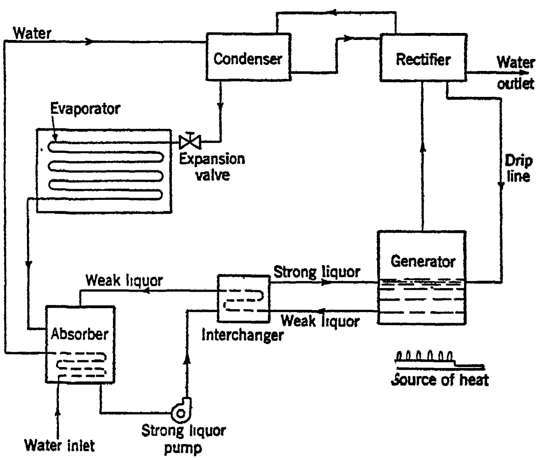

FIG. 6. CLOSED ABSORPTION SYSTEM

deviation occurs when there is a difference in the internal pressure of the components, chemical attraction between them being absent.

In order to make an effective absorption machine, large negative deviations from Raoult's Law must be shown by solutions of the refrigerant in the liquid absorbent, because the larger the negative deviation, the greater is the amount of refrigerant that can be cycled, using a given weight of absorbent. Cycling a large amount of refrigerant for a given weight of absorbent is important because of the heat required to raise the temperature of the mixture and disassociate the refrigerant and the absorbent. Only the latent heat of the refrigerant can be recovered for useful work.

Many refrigerant-absorbent combinations have been proposed and quite a number have been tested. A diagrammatic representation of a typical closed absorption system is outlined in Fig. 6. In this system a mixture of refrigerant and absorbent is evaporated in the generator, passes to an analyzer and rectifier where it is purified, and then to a condenser where the refrigerant and remaining absorbent is condensed. It

then passes through an expansion valve to an evaporator, where heat is absorbed from a cooling load. From the evaporator the vapor and residual absorbent passes to an absorber where it meets absorbent which is initially low (weak) in refrigerant concentration. The absorbent absorbs the vapor and the strong absorbent liquor is transferred to the generator through an interchanger with the weak liquor returning from the generator.

A cooling medium, ordinarily water, is used in the absorber to remove the heat of absorption and maintain the absorptive power of the absorber at a maximum.

Like the steam jet system, the absorption system compares most favorably when a cheap source of cooling water and steam or other heat is available. Unlike the steam jet system, the comparative performance is usually best with a wide range of temperature between the evaporator and absorber, since with a good refrigerant-absorbent combination, the amount of heat and water required for a given refrigerating effect increases slowly with an increase of evaporator-condenser temperature range.

At the present time the most used refrigerant-absorbent combinations are: (1) water and ammonia, and (2) dichloromonofluoromethane and dimethyl ether of tetraethylene glycol. With the latter combination the boiling points of the refrigerant and absorbent are sufficiently wide apart that almost pure refrigerant is obtained without the use of a rectifier.

EXPANSION VALVES

The thermostatic expansion valve is a device to regulate the flow of liquid refrigerant so that the evaporator will always be used to best advantage. The evaporator coil must be kept as full as possible without any chance of liquid refrigerant entering the suction line. The expansion valve accomplishes this by regulating the supply of refrigerant, so that the temperature of the gas leaving the evaporator is always slightly higher than the temperature of the boiling refrigerant inside of it. This difference in temperature between the outgoing (suction) gas and the liquid refrigerant in the evaporator is called the superheat of the gas.

The operation of the thermostatic expansion valve can best be explained by means of a diagram, Fig. 7. A small refrigerant charge in the control bulb exerts a pressure through the tube to the upper side of the diaphragm, which tends to open the valve.

The magnitude of this pressure is determined by the temperature of the suction gas leaving the evaporator, as the control bulb is attached to the suction line at this point and is at approximately the same temperature. The suction pressure in the evaporator is transmitted through the equalizer tap and exerts an opposing force on the other side of the diaphragm in the direction to close the valve. This pressure corresponds to the temperature of the boiling refrigerant. The resulting force on the diaphragm is determined by the differential between the temperature of the suction gas and the boiling point of the refrigerant, which is the amount of superheat in the gas. If this temperature differential becomes greater (superheat increases), the resultant force on the diaphragm opens the valve and admits more refrigerant. The reverse is true if the superheat decreases,

and the valve partly closes, thus admitting less refrigerant. The spring keeps the valve closed until the resultant force on the diaphragm corresponds to the desired superheat. The adjustment of the spring will change the amount of superheat to be maintained in the suction gas.

The selection of the expansion valve is, of course, determined by the capacity of the valve. The capacity of a valve with a given orifice is determined by the refrigerant used, the differential of pressure across the valve and the amount the liquid is sub-cooled as it enters the valve. The expansion valves are usually rated at zero sub-cooling of the liquid, or 100 per cent liquid. Oftentimes special devices are used to properly distribute the refrigerant among the parallel paths of the evaporator. These distributing devices usually have considerable pressure drop. Where they are used, the pressure drop across the expansion valve is not the difference between suction and discharge pressures, as allowance must be made for the pressure drop across the distributing device. An equal-

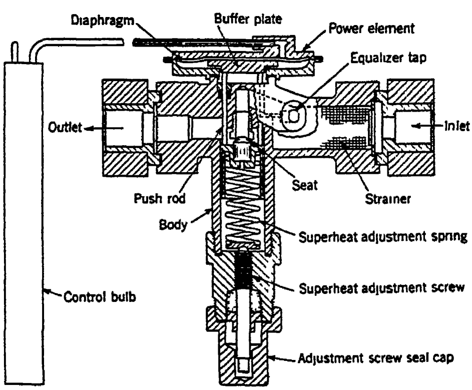

FIG. 7. TYPICAL THERMOSTATIC EXPANSION VALVE

izer connection from the evaporator suction line must be made to the underside of the diaphragm (see Fig. 7) whenever the valve outlet is not at the evaporator pressure so as to insure suction pressure at this point. When distributing devices are used, this equalizer connection is essential for proper operation of the valve. Another pressure drop allowance must be made for the liquid line, particularly when the liquid line has an appreciable vertical rise.

CONDENSERS

Condensers used for liquifying the refrigerant are of three general designs: (1) air, (2) water, and (3) evaporative (combination air and water).

Air Cooled

Air cooled condensers are seldom used for capacities above 3 tons of refrigeration, unless an adequate water supply is extremely difficult to

obtain, as, for instance, in railway air conditioning. Even on fractional tonnage installations, air is used as the condensing medium only where water is expensive or where simplicity of installation warrants the higher condensing pressure, and consequent higher power costs than would be obtained using water as the condensing medium.

The conventional air cooled condenser consists of an extended surface coil across which air is blown by a fan. The hot discharge gas enters the coil at the top and, as it is condensed, flows to a receiver located below the condenser. Air cooled condensers should always be located in a well ventilated space so that the heated air may escape and be replaced by cooled air.

The principal disadvantages of air cooled condensers are the power required to move the air and the reduction of capacity on hot days. This loss of capacity due to high condensing pressures on hot days requires that equipment of increased capacity be selected to meet the peak load. Thus at normal loads the equipment is oversized.

Water Cooled

Water cooled condensers are of the double pipe type, the shell and tube type, or the shell and coil type. Double pipe condensers are arranged so that water passes through the inner of two concentric pipes and refrigerant circulates through the annular space between the pipes. Where possible, there should be counter-flow of the refrigerant and the condensing water to obtain maximum temperature differences. This type is usually used only with small condensing units.

The amount and temperature of the condensing water determine the condensing temperature and pressure, and indirectly the power required for compression. It is therefore necessary to determine a balance so that the quantity of water insures economical compressor operation.

Because there is a decided tendency to conserve the water in city mains and because most large cities are restricting the use of water for air conditioning and refrigeration equipment, it is often necessary to install cooling towers or evaporative condensers. Cooling towers, unfortunately, produce the warmest condensing water at the time when the load on the system is greatest, so that the refrigeration equipment must be designed to meet the maximum load at abnormal condensing water temperatures. If properly designed, this makes little difference in the efficiency of operation throughout the year except at those times when the condensing water temperature is highest. As this occurs only for 5 per cent of the entire cooling period it can be disregarded as a factor in establishing yearly operating costs.

The cooling tower has a certain advantage over the use of water from the city mains. Economies are possible when a cooling tower is used, which cannot be achieved by the use of condensing water from city mains. In certain localities, the lowest city water temperature met during the summer months is from 65 to 70 F. This temperature range takes place for the entire cooling period, regardless of the outdoor temperature. With a cooling tower, the temperature of the condensing water may rise to 80 or 85 F under maximum conditions, but under less than maximum conditions the temperature of the water leaving the cooling tower drops

considerably. It has been established that in these localities during 50 per cent of the time, the outdoor wet-bulb temperature varies from 60 to 70 F and the cooling tower water, for the same periods, varies from 65 to 75 F. When the outdoor wet-bulb temperature drops below 60 F, which occurs approximately 30 per cent of the time, the condensing water temperature is still lower. The cost of water used for condensing is small as the only water required is that used to make up the loss by evaporation in the cooling tower itself. Refer to the section on Cooling Towers in Chapter 27.

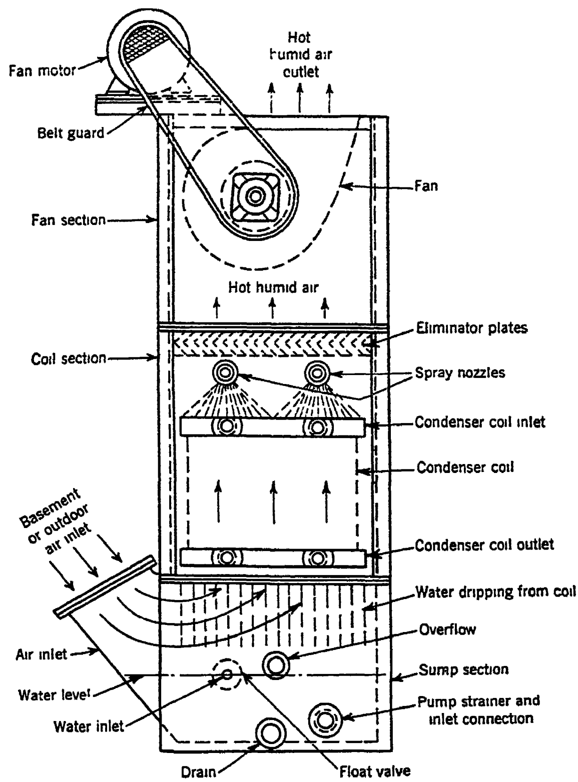

Fig. 8. Schematic View of an Evaporative Condenser

Shell and coil condensers are in general use for medium sized condensing units, and consist of a coil of tubing mounted inside a shell. The cooling water passes through the coil.

Evaporative Condensers

Due to the high cost of city water for condenser purposes, and due to ordinances in some localities prohibiting the discharge of large quantities of such water into the sewage systems, there has been developed a condenser which uses a minimum amount of water on a finned surface, cooling

it to approximately the wet-bulb temperature of the surrounding atmosphere.

The end view of a typical evaporative condenser is shown in Fig. 8. The fan draws the air over a finned tube condenser which is kept wet by a water spray. The discharge refrigerant gas from the compressor enters the top of the condenser coil and the liquid refrigerant is drained from the bottom of the coil into a liquid receiver and then circulates through the remaining portion of the system in the usual way.

The water is circulated through the spray nozzles and the level is maintained in the sump by means of a float valve. The eliminator plates are placed in the path of the water-air mixture so as to remove the entrained water. The air leaving the unit is almost completely saturated, so that care must be taken in locating discharge ducts to prevent condensation.

Evaporative condensers are available in sizes up to 100 tons or more. These units use only a small portion of the water required for a water cooled condenser. The water is vaporized by the heat of the refrigerant so that each pound of water used extracts approximately 1000 Btu from the refrigerant, whereas, under standard rating conditions where the water temperature rise is 20 F, each pound of water extracts only 20 Btu from the refrigerant. Including the water lost by entrainment in the discharge air, by overflow and stand-by evaporation, the water used is about 3 to 5 per cent of the amount that would be required for a water cooled condenser.

The evaporative condenser requires more maintenance, occupies greater space (must be located where air is available), and has a higher first cost than the water cooled condenser, but where the use of water is restricted or expensive, the evaporative condenser has become widely accepted. Compared with a water cooled condenser and cooling tower, which combination uses about the same quantity of water, the evaporative condenser has the advantage of lower cost and smaller space requirements.

EVAPORATORS AND COOLERS

The types of coolers used in connection with air conditioning work fall into three general groups. The *first*, is the direct cooling of water; the *second*, direct cooling of air; and the *third*, cooling of brine for circulation in a closed system, which can cool either water or air. One method of the direct cooling of water is to install direct expansion coils in the spray chamber so that the water sprayed into the air comes in direct contact with the cooling coils. Another common and efficient method of cooling spray water is to use a Baudelot type of heat absorber where the water flows over direct expansion coils at a rate sufficiently high to give efficient heat transfer from water to refrigerant.

Another type of spray water cooler is the shell and tube heat exchanger in which the refrigerant is expanded into a shell enclosing the tubes through which the water flows. The velocity of the water in the tubes affects the rate of heat transfer, and as the refrigerant is in the shell completely surrounding the tubes at all times, good contact and a high rate of heat transfer are insured. The disadvantage of such a system is that with the falling off of load on the compressor the suction temperature or the

temperature in the evaporator drops and there is a possibility of freezing the water in the tubes, which, of course, might split the tubes and allow the refrigerant to escape into the water passage. This danger can be eliminated by automatic safety devices.

Another system of cooling spray water is to submerge coils in the spray collecting tank, or in a separate tank used for storage. The heat transmission through the walls of the coils, however, is low and a great deal more surface is required than for any other type of cooler. However, with large storage tanks this type of cooling can be utilized to advantage.

When direct cooling of air is employed, the refrigerant is inside the coil and the air passes over it. Cooling depends upon convection and conduction for removing the heat from the air. The type of coil used can be either smooth or finned, the finned coil being more economical in space requirement than the smooth coil. The fins, however, must be far enough apart so as not to retain the moisture which condenses out of the air.

The indirect cooler, where brine is cooled by the refrigerant and the resulting cold brine is used to cool either air or water, introduces several other considerations. It is not the most economical from a power consumption standpoint, as it is necessary to cool the brine to a temperature sufficiently low so that there is an appreciable difference between the average brine temperature and that of the substance being cooled. This requires that the temperature of the refrigerant must be still lower, and consequently the amount of power required to produce a given amount of refrigeration increases due to the higher compression ratio. There are other considerations which make such a system desirable. In the first place, where a toxic refrigerant is undesirable or cannot be used because of fire or other risks, especially in densely populated areas, the brine can be cooled in an isolated room or building and can then be circulated through the air conditioning equipment. This arrangement eliminates any possibility of direct contact between the air and refrigerant.

REFRIGERANT PIPE SIZES

The selection of proper pipe sizes and frictional pressure losses varies with the installation and the capacity of the system. Generally the suction piping should be selected so that the pressure loss is between 2 and 3 lb per square inch. The pressure drop in liquid lines should be maintained so as to permit no vaporization in the pipes with limiting pressure drops not to exceed 5 lb per square inch. Hot or discharge gas lines should be limited to approximately 4 lb per square inch pressure drop. All pressure drops mentioned are total system losses and include not only the piping losses, but also the pressure losses in the valves, fittings and coils.

Pressure drops for discharge or hot gas lines may be determined from Table 7. Pressure losses in liquid refrigerant lines of various sizes and capacities are given in Table 8. Pressure drops of suction refrigerant pipe lines at varying capacities and refrigerant temperatures are given in Table 9. Oil circulating with the refrigerant appreciably increases the pressure losses in both suction and discharge lines from that given in these tables. All tables are for 100 ft of pipe, including an average number of fittings, and for other lengths the losses are proportionate. Losses through

Table 7. Pressure Losses in Dichlorodifluoromethane Discharge or Hot Gas Lines[a]

Capacity Btu per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[b]									
	Line Sizes, Inches									
	5/8	3/4	7/8	1 1/8	1 3/8	1 5/8	2 1/8	2 5/8	3 1/8	3 5/8
10,000	2.3	1.0	0.6							
15,000	4.9	2.0	1.0							
20,000	8.5	3.4	1.7	0.6						
25,000		5.3	2.6	0.9						
30,000		7.5	3.6	1.2	0.5					
40,000			6.4	2.1	0.7					
50,000			9.8	3.1	1.0	0.5				
60,000				4.4	1.3	0.7				
70,000				6.0	1.9	0.9				
80,000				8.0	2.5	1.1				
90,000				10.2	3.1	1.4				
100,000					3.8	1.7	0.5			
125,000					6.0	2.6	0.7			
150,000					8.5	3.8	1.0			
175,000					11.6	5.1	1.3			
200,000						6.7	1.7	0.6		
250,000						10.4	2.6	0.9		
300,000							3.7	1.2	0.5	
400,000							6.7	2.2	0.9	
500,000							10.5	3.5	1.5	0.7
600,000								5.0	2.1	1.0
800,000								9.0	3.8	1.8
1,000,000									5.8	2.9
1,250,000									9.5	4.4
1,500,000										6.4
2,000,000										11.3

[a]Soft annealed copper tubing up to and including 3/4 in. outside diameter. Hard copper pipe 7/8 in. outside diameter and larger.

[b]Length of tubing includes the average number of fittings.

control and regulating valves must be added to the other pipe losses to determine the total drop. All copper pipe referred to in these tables are of type L wall thickness and are designated by outside diameter.

The effect of the sizes of refrigerant lines on the system may be studied by referring to the preceding discussion on Characteristics of Compression Systems. It will be noted that any lowering of the suction pressure at the compressor lowers the capacity. Therefore, excessive pressure drop through the suction piping should be avoided. On the other hand, the suction line must not be made too large when using refrigerants which are soluble in oil, because under such circumstances the velocity of the returning refrigerant may become too low to carry back the entrained oil. Pressure drop in the discharge line also lowers the capacity of the system

Table 8. Pressure Losses in Dichlorodifluoromethane Liquid Refrigerant Lines

Capacity Btu per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[a]			
	Pipe Sizes, Inches			
	7/8	1 1/8	1 3/8	1 5/8
100,000	0.6			
125,000	0.9			
150,000	1.3			
175,000	1.8			
200,000	2.3	0.6		
225,000	2.9	0.8		
250,000	3.6	1.0		
275,000	4.3	1.2		
300,000	5.1	1.4		
325,000	5.9	1.6		
350,000	6.9	1.8		
375,000	7.9	2.1		
400,000	9.0	2.3	0.8	
450,000		2.9	1.0	
500,000		3.5	1.3	
550,000		4.3	1.5	0.7
600,000		5.0	1.8	0.8
700,000		6.7	2.4	1.1
800,000		8.7	3.1	1.4
900,000			3.9	1.7
1,000,000			4.7	2.1
1,200,000			6.7	3.0
1,400,000			9.0	4.0
1,600,000				5.1
1,800,000				6.3
2,000,000				7.9
2,200,000				9.2

[a]Length of tubing includes the average number of fittings.

but not to the same extent as does the pressure drop in the suction line. The velocities of the refrigerant in either suction or discharge lines must not be excessive or noise will result. Velocities of 1000 to 2000 fpm are common in suction lines, and from 2000 to 3500 fpm are used in discharge lines. Velocities in the discharge lines as high as 5000 fpm can only be used where the fittings and bends are all stream-lined as noise will otherwise result.

The pressure drop in the liquid line affects the capacity of the expansion valve as the pressure drop across the valve is reduced by the amount of the pipe line drop. If the liquid line drop is sufficient to cause flashing (*i.e.* vaporizing) of some of the liquid refrigerant, a hissing noise in the lines and valves usually develops.

Table 9. Pressure Losses in Dichlorodifluoromethane Suction Refrigerant Lines

Copper Pipe Actual O.D. Inches	Capacity Btu per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[a] — Refrigerant Temperature Deg F: −10	0	10	20	30	40	50
¾	2,000	0.3	0.3	0.2	0.2	0.2	0.1	0.1
	4,000	1.3	1.0	0.8	0.7	0.6	0.5	0.4
	6,000	2.8	2.2	1.8	1.5	1.2	1.0	0.9
	8,000	4.8	3.8	3.1	2.6	2.1	1.8	1.5
	10,000	7.4	5.8	4.8	3.9	3.3	2.8	2.3
	12,000	10.5	8.4	6.8	5.6	4.7	4.0	3.3
	14,000	14.0	11.0	9.1	7.6	6.4	5.4	4.5
	16,000		14.5	12.0	9.8	8.3	7.0	5.8
	18,000			15.0	12.3	10.4	8.7	7.2
	20,000				15.0	12.7	10.7	8.9
1⅛	7,000	0.4	0.3	0.3	0.2	0.2	0.2	0.1
	10,000	1.0	0.7	0.5	0.5	0.4	0.3	0.3
	15,000	1.9	1.5	1.2	1.0	0.8	0.7	0.6
	20,000	3.3	2.6	2.1	1.7	1.4	1.2	1.0
	25,000	5.0	4.0	3.2	2.7	2.2	1.9	1.6
	35,000	9.7	7.7	6.2	5.1	4.3	3.6	3.0
	45,000	15.8	12.6	10.0	8.4	7.0	5.9	4.9
	60,000				14.8	12.2	10.2	8.6
	70,000						14.0	11.7
1⅜	10,000	0.3	0.2	0.2	0.2	0.1	0.1	0.1
	15,000	0.7	0.5	0.4	0.3	0.3	0.2	0.2
	20,000	1.2	0.9	0.7	0.6	0.5	0.4	0.4
	30,000	2.6	2.1	1.6	1.3	1.1	0.9	0.8
	40,000	4.6	3.6	2.8	2.3	1.9	1.6	1.4
	50,000	7.0	5.5	4.4	3.5	2.9	2.5	2.1
	60,000	10.0	7.8	6.2	5.0	4.2	3.5	3.0
	80,000		14.0	11.0	8.7	7.3	6.2	5.2
	100,000				13.5	11.3	9.5	8.2
1⅝	30,000	1.6	1.3	1.0	0.8	0.7	0.6	0.5
	40,000	2.7	2.1	1.7	1.4	1.1	0.9	0.8
	50,000	4.2	3.2	2.5	2.1	1.7	1.4	1.2
	60,000	6.1	4.5	3.6	2.9	2.4	2.0	1.7
	70,000	8.7	6.3	4.8	3.8	3.1	2.6	2.2
	80,000		8.4	6.3	4.9	4.0	3.3	2.8
	90,000			8.0	6.2	4.9	4.1	3.5
	100,000			10.0	7.6	6.1	5.0	4.2
	120,000					8.6	7.0	5.9
	140,000						9.5	7.9

[a]Length of tubing includes the average number of fittings.

TABLE 9. PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE SUCTION REFRIGERANT LINES (CONTINUED)

Copper Pipe Actual O.D Inches	Capacity Btu per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[a] Refrigerant Temperature Deg F						
		−10	0	10	20	30	40	50
2⅛	50,000	0.7	0.5	0.4	0.3	0.3	0.2	0.2
	100,000	2.6	1.8	1.4	1.1	0.9	0.8	0.7
	150,000	5.6	3.9	3.0	2.4	2.0	1.6	1.4
	200,000	9.8	6.7	5.2	4.1	3.4	2.8	2.4
	250,000	14.8	10.3	8.0	6.3	5.1	4.2	3.6
	300,000		14.5	11.3	9.0	7.2	6.0	5.0
	350,000		19.5	15.3	12.0	9.7	7.8	6.7
	400,000			19.6	15.3	12.5	10.0	8.5
2⅝	50,000	0.2	0.2	0.1	0.1	0.1	0.1	0.1
	100,000	0.7	0.6	0.5	0.4	0.3	0.2	0.2
	150,000	1.6	1.2	1.0	0.8	0.6	0.5	0.4
	200,000	2.8	2.1	1.7	1.4	1.1	0.9	0.7
	250,000	4.3	3.4	2.6	2.1	1.7	1.3	1.1
	300,000	6.1	4.5	3.7	3.0	2.4	1.9	1.5
	350,000	8.2	6.0	5.0	4.0	3.2	2.5	2.0
	400,000		7.8	6.5	5.1	4.2	3.3	2.7
	450,000			7.7	6.4	5.3	4.0	3.5
	500,000				7.8	6.4	5.0	4.2
	550,000					7.7	6.2	5.1
	600,000						7.4	6.2
3⅛	200,000	1.2	1.0	0.8	0.6	0.5	0.4	0.4
	300,000	2.6	2.0	1.6	1.3	1.0	0.8	0.7
	400,000	4.5	3.4	2.6	2.1	1.7	1.4	1.3
	500,000	7.3	5.4	4.1	3.3	2.7	2.2	1.9
	600,000		8.1	6.0	4.7	3.8	3.1	2.7
	700,000			8.4	6.5	5.2	4.2	3.5
	800,000				8.6	6.8	5.5	4.6
	900,000					8.7	7.0	5.9
	1,000,000						8.9	7.3
3⅝	300,000	1.2	0.9	0.7	0.6	0.5	0.4	0.3
	400,000	2.0	1.6	1.3	1.0	0.8	0.7	0.6
	500,000	3.2	2.5	1.9	1.6	1.3	1.0	0.9
	600,000	4.6	3.6	2.8	2.2	1.8	1.5	1.3
	700,000	6.4	4.9	3.8	3.0	2.5	2.0	1.7
	800,000	8.7	6.4	4.9	3.9	3.2	2.5	2.2
	900,000		8.2	6.2	4.9	3.9	3.2	2.7
	1,000,000			7.7	6.1	4.9	4.0	3.3
	1,100,000			9.4	7.3	5.8	4.8	4.0
	1,200,000				8.7	6.9	5.6	4.8
	1,300,000					8.0	6.6	5.6
	1,400,000					9.3	7.6	6.4

[a]Length of tubing includes the average number of fittings.

TABLE 9. PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE SUCTION REFRIGERANT LINES (CONCLUDED)

Copper Pipe Actual O.D. Inches	Capacity Btu per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[a]						
		Refrigerant Temperature Deg F						
		−10	0	10	20	30	40	50
4⅛	400,000	1.0	0.8	0.6	0.4	0.4	0.3	0.3
	600,000	2.4	1.8	1.4	1.1	0.9	0.7	0.6
	800,000	4.1	3.1	2.4	2.0	1.6	1.3	1.1
	1,000,000	6.6	4.8	3.7	3.0	2.5	2.0	1.6
	1,200,000	10.0	7.1	5.4	4.4	3.5	2.9	2.4
	1,400,000		10.0	7.5	5.9	4.8	3.9	3.3
	1,600,000			10.0	7.7	6.2	5.1	4.2
	1,800,000				10.0	7.9	6.4	5.3
	2,000,000					9.7	7.9	6.6
	2,200,000						9.5	7.9

[a]Length of tubing includes the average number of fittings.

ICE SYSTEMS

Cold water systems using ice as the cooling agent have been installed in many theaters, restaurants, funeral homes, churches and other places where short hours of operation and high peaks of cooling demand make this type of system desirable. A comparatively small quantity of ice in the water cooling tank of such a system can release refrigeration at a relatively rapid rate. For instance, neighborhood theaters having a peak demand of 1,200,000 Btu per hour (100 tons refrigeration) have found 8 ton capacity ice bunkers satisfactory.

In operation, the water in the air conditioning system is circulated over ice placed in an insulated box and is cooled to the 38 or 40 F range or higher if desired. This cold water is pumped from the ice bunker to air cooling coils or spray type air washers. The blowers, coils, air washer or air handling sections are the same as those parts in any system employing cold water as a refrigerant.

The ice water cooler or ice bunker is usually built on the job in a location where it can easily be iced. It can be built of any desired material such as concrete, steel, or wood with a 4 in. thickness of standard insulation to save the ice from one period of use to the next. The basic requirement is that the tank be durable and water tight. A typical bunker with connections to a coil type air conditioning system is shown in Fig. 9. About 60 cu ft of gross bunker volume is allowed per ton of ice capacity.

The shape of the bunker usually conforms to the available space. The one illustrated has overhead sprays, but if head-room is lacking the ice is placed on the floor of the bunker with the water returned around the lower part of the blocks from a perforated distribution pipe run along one side of the bunker. To secure good circulation the supply water is extracted from a similar perforated pipe on the opposite side of the bunker.

The temperature of the water is controlled at a predetermined point by a thermostat in the supply line. If the temperature drops too low, a part of the return water is by-passed directly to the sump and is not cooled over the ice. In the larger systems it is customary to install an overflow control which, as the ice melts, discards the excess water through an economizer coil. The surface of the economizer is large in relation to the flow so that the water is warmed to 60 F or more as it is discharged from the system.

EQUIPMENT SELECTION

The selection of proper refrigeration equipment for any air conditioning job is of utmost importance for satisfactory results. The most important factors in the selection of the equipment are:

1. Loads (as determined by the conditions of the space to be cooled).
2. Economics (both initial and operating costs).

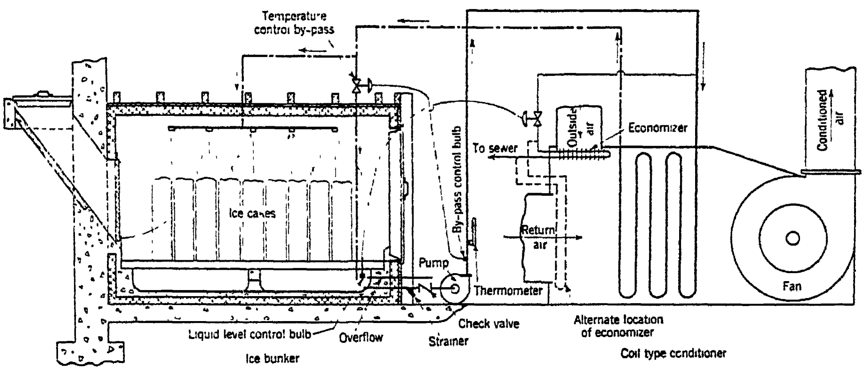

FIG. 9. TYPICAL ICE SYSTEM SHOWING BUNKER DETAILS

3. Codes (local safety codes must be adhered to and influence the type of system to be used).

A broad division of equipment to be used may be made on the basis of the magnitude of the load. Current general practice is outlined in Table 10.

Unit or *packaged* systems, consisting of a reciprocating compressor, condenser, evaporator and fans, are generally used in the smaller sized jobs where electric power is available, as they are manufactured complete ready to install and are the most economical.

The reciprocating compressor in the built-up central system covers the widest range of application since it is applicable to either the direct expansion or indirect systems and can be driven by steam or gas engines, or by electric motors. The quantity of condensing cooling medium required is also less than for any other system with the exception of the centrifugal compressor, which uses the same amount.

Centrifugal compressors are used for large installations, and usually where the indirect system is required. The driving mechanism can be

TABLE 10. BASIS OF EQUIPMENT SELECTION

CAPACITY TONS	MAJORITY USED	SOME USED	FEW USED
0 to 5	Unit systems in conditioned space.	Unit central systems using duct distribution.	Built up central systems.
5 to 25	Built up central systems using reciprocating compressors.	Unit central systems using duct distribution.	Unit systems in conditioned space. Built up systems using absorption and adsorption systems.
25 to 50	Built up central systems using reciprocating compressors.	Built up central systems using centrifugal compressors.	Central systems using adsorption systems.
50 to 400	Built up central systems using reciprocating compressors.	Built up central systems using steam jet and centrifugal compressors.	
400 and Over	Built up central systems using centrifugal compressors.	Built up central systems using steam jet.	

steam turbine or electric motor. The steam jet system is used where steam is available and cooling water can be had in large quantities.

It will be noted by referring to Fig. 4 that all systems using compressors have a common characteristic and that is, that the capacity varies with the evaporating temperature. Not only can the equipment be selected to produce a given result but the performance can be predicted under varying load conditions by the simple expedient of using the variable of evaporating temperature as the abscissa and the load or capacity as the ordinate in a series of curves.

Manufacturers of compressors and cooling coils furnish performance data for apparatus that can be plotted in the form of curves similar to those shown in Fig. 10. The performance of a compressor is plotted as a series of curves, each curve being drawn for a given condensing pressure. The performance of a direct expansion coil at two different air velocities is plotted on the same graph. The operating point will be, of course, where the two curves cross.

TABLE 11. TYPICAL OPERATING CONDITIONS FOR TWO TYPES OF LOAD

TYPE OF ENCLOSURE	LOAD, BTU PER HOUR			RATIO SENSIBLE TO TOTAL	AIR ENTERING COIL		OPERATING BALANCE POINT		
	Sensible	Latent	Total		Deg F	Per Cent R.H.	Evaporator Temp Deg F	Condenser Pressure Lb per Sq In.	Per Cent Sensible Heat
Restaurant	103,000	45,000	148,000	0.695	82	45	34.4	123	69.9
Office	121,000	27,000	148,000	0.820	82	45	42.2	100	82.1

Data given in Table 11 illustrate two types of conditioned enclosures having the same total load of 148,000 Btu per hour, but with two different ratios of sensible to total heat. In the case of the office with a ratio of 82 per cent sensible to total heat, the operating point *A* in Fig. 10 is found to be 42.2 F evaporating temperature with a face velocity of 500 fpm. In the case of the restaurant, with a ratio of 69.5 per cent sensible to total heat, the air velocity is lowered to 300 fpm and the evaporating temperature is lowered to 34.4 F as shown in point *B* of Fig. 10. In order to obtain the same capacity, a larger condensing unit is used. This illustration assumes zero pressure drop through the suction line. The pressure drop can be taken into account by shifting the compressor performance curves by the amount of pressure drop expressed in degrees Fahrenheit.

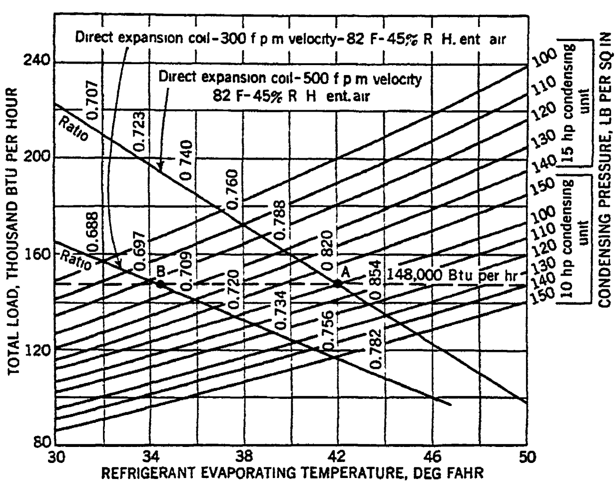

FIG. 10. COMPRESSOR AND COIL PERFORMANCE

THE REVERSE CYCLE

In heating by the reverse refrigeration cycle energy is absorbed in an evaporator from some available source of heat, pumped to a higher temperature and delivered to a condenser. The heat from the condenser is used for heating purposes. The compressor acts as a heat pump whose fundamental function is to raise the potential of the heat. The theoretical ratio of the heat delivered to the work of compression is given in Equation 1.

$$\frac{T_2}{T_2 - T_1} \tag{1}$$

where

T_1 = absolute temperature of evaporator.
T_2 = absolute temperature of condenser.

Thus, with a small spread of temperature between the evaporator and the condenser, 6 or 8 times as much heat may be obtained theoretically,

and 3 to 5 times practically, as the work introduced[1]. There are a number of limitations, however, the most serious of which is the lack of ready availability of a practical source of heat.

1. Well water is the most desirable since its temperature is higher than other sources even in the winter, and thus a large amount of heat may be removed in relation to the weight of water handled.
2. Air may be used but its specific heat is low and its temperature uncertain. When the most heat is needed, the temperature of the air is lowest, thus resulting in the least favorable temperature combination.
3. It has been proposed to obtain heat by freezing water but this is still in the experimental stage.

Some of the other factors which act as limitations are: the large temperature spread when using air as a source of heat and when attempting to cool with even moderately low outside temperatures, the frequent disparity between the size of the cooling load and heating load requiring

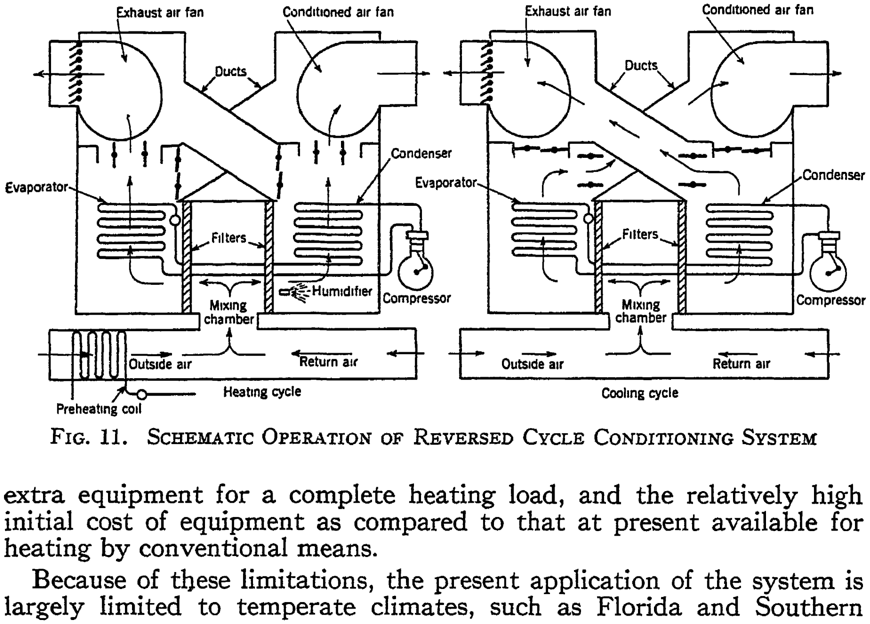

FIG. 11. SCHEMATIC OPERATION OF REVERSED CYCLE CONDITIONING SYSTEM

extra equipment for a complete heating load, and the relatively high initial cost of equipment as compared to that at present available for heating by conventional means.

Because of these limitations, the present application of the system is largely limited to temperate climates, such as Florida and Southern

[1]Cooling Homes, A Field for Refrigeration, by A. R. Stevenson, presented at the symposium of the Refrigeration with Gas Committee of the *American Gas Association*, April 20, 1926.

The Heat Pump, An Economical Method of Producing Low-grade Heat from Electricity, by T. G. N. Haldane (*Electric Review*, Vol. 105, p. 1161-1162, December 27, 1929, and *I. E. E. Journal*, Vol. 68, p. 666-675, June, 1930).

Edison Building Heated and Cooled by Electricity, by H. L. Doolittle (*Power*, Vol. 74, p. 384, September 8, 1931).

House Heating by Pump with 5 to 1 Pick-up Ratio, by Gilbert Wilkes and R. E. Marbury (*Electrical World*, Vol. 100, p. 828, December 17, 1932).

An All Electric Heating, Cooling and Air Conditioning System, by Philip Sporn and D. W. McLenegan (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 307).

Using the Reversed Cycle Refrigerating Principle for a Self-Contained Heating and Cooling Unit, by Henry L. Galson (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1935, p. 497).

Heating by Reversed Refrigeration, by A. J. Lawless (*Heating, Piping and Air Conditioning*, August, p. 473, September, p. 519, 1940).

California, or to heating only for intermediate seasons, or to other localities which have peculiar advantages as, for instance, the ready availability of well water. In these locations it is frequently possible to do all of the heating necessary with the refrigeration equipment so that the extra cost is only that of reversing the functions of the condenser and evaporator.

There are a number of reversed systems now in operation, particularly among utility companies, using well water as the source of heat. These systems range in size up to 320 hp. In the case of the largest system in operation at present, the cost of the electrical energy would have to be approximately 0.7 cents per kilowatthour in order to compete with oil at 6 cents per gallon.

A typical arrangement of a reversed cycle conditioning system where air is used as a source of heat is shown in Fig. 11. If the air seldom drops below freezing, heat is often required in the morning and cooling during the afternoon in order to maintain comfortable conditions in such a system. The arrangement as shown lends itself to automatically changing over as required.

Example 1. Electrically driven dichlorodifluoromethane condensing units are to be used in an air conditioning system, requiring 20 tons refrigerating capacity for conditions of maximum load. An overall analysis of the seasons operating conditions shows an average load factor of 62.5 per cent, and allowing for variable time intervals of operation of refrigeration units installed, three-quarters of the operating season, or 750 hr, would require operation of the equipment at one-half load, and one-quarter of the operating season or 250 hr full load capacity of the refrigeration equipment would be required.

The increased first cost of 2-10 hp, 10 ton condensing units over 1-20 hp, 20 ton condensing unit is, $830.00 installed price, to the customer.

The increased first cost of a 2-speed compressor motor of 20 hp size over a constant speed of 20 hp size motor including increased starter cost is $210.00. The efficiency of the 2-speed motor above is 83 per cent at full load speed, and 79 per cent for full load at ½ speed. At ½ speed, full load is ½ total bhp of full load speed.

Discuss the consideration involved in making a decision as to whether a single unit with a 20 hp motor of the 2 speed type would be used in preference to 2-10 hp constant speed units.

Solution. The cost of 2-10 hp 10 ton units in excess of 1-20 hp, 20 ton unit with 2-speed motor, is $830.00 − $210.00 or $620.00, increased first cost. At 15 per cent fixed charges, this represents an increased annual cost of $93.00 for 2 compressors over one compressor. The advantage of 2 compressors instead of one compressor on an installation of this type, is in the breakdown service provided in the event one compressor is shut down for repairs the system could be operated at one-half capacity utilizing the duplicate machine. The motor efficiency of the constant speed unit would be higher at full load than would be the efficiency of the 2-speed motor at low speed. Offsetting this latter advantage however, is the fact that the condenser on the condensing unit would provide a lower refrigerant condensing temperature for ½ load operation with the same final condensing water temperature than would be the case with duplicate units each furnished with its own compressor and condenser. Operation at a lower condensing temperature would provide for a power saving compensating for the lower efficiency of the 2-speed motor when operated at slow speeds. It is, in a case of this kind, purely a question as to whether or not the purchaser would deem an investment of $620.00 more and an increased fixed charge of $93.00 a year, advisable to get breakdown service through the installation of duplicate units. In most cases, this increased first cost would not be warranted because of the fact that satisfactory indoor conditions could not be obtained at full load if only one-half the refrigeration capacity were available.

Example 2. For condensing purposes, an air conditioning system uses city water which has an average 70 F supply temperature. The following table lists the number

of hours per year during which definite wet-bulb temperatures and corresponding refrigeration rates pertain.

Wet-Bulb Temperature F	No. of Hours per Year	Refrigeration Required Tons
80	6	284
79 – 75	100	233
74 – 70	277	183
69 – 65	330	157
64 – 60	277	144
59 – 55	158	79
54 – 50	52	37
	Total 1200 hours	

If the power requirements of a dichlorodifluoromethane refrigeration system are in accordance with the following data on partial load operation, determine the seasonal power cost at 2 cents per kwhr:

Tons of Refrigeration	284	233	183	157	144	79	37
Kw per ton	0.89	0.89	0.87	0.86	0.86	0.93	0.97

Solution. Seasonal power cost:

Wet Bulb Temperature F	Ton-Hours	Kwhr
80	6 × 284 = 1,704	1,704 × 0.89 = 1,517
79 – 75	100 × 233 = 23,300	23,300 × 0.89 = 20,750
74 – 70	277 × 183 = 50,700	50,700 × 0.87 = 44,100
69 – 65	330 × 157 = 51,800	51,800 × 0.86 = 44,500
64 – 60	277 × 144 = 39,900	39,900 × 0.86 = 34,300
59 – 55	158 × 79 = 12,500	12,500 × 0.93 = 11,600
54 – 50	52 × 37 = 1,920	1,920 × 0.97 = 1,860
Totals	181,824 ton-hours	158,627 kwhr

The 158,627 kwhr at 2 cents per kwhr will cost $3,173.

The average consumption will be $\frac{158{,}627 \text{ kwhr}}{181{,}824 \text{ ton-hours}} = 0.873$ kw per ton.

Example 3. Using the data from Example 2, if city water costs 20 cents per thousand gallons, and if 1.25 gal are used per minute per ton, estimate the annual water cost.

Solution.

$60 \times 1.25 = 75$ gal per ton-hour.

181,824 ton-hours $\times$ 75 = 13,620,000 gal per year.

$$\frac{13{,}620{,}000 \times \$0.20}{1000} = \$2{,}724 \text{ the yearly cooling water cost.}$$

Chapter 26

HEAT TRANSFER SURFACE COILS

Coil Applications, Construction and Arrangement, Steam Coils, Water Coils, Direct-Expansion Coils, Flow Arrangements, Applications, Calculation of Heat Transfer, Air Flow Resistance, Coil Performance, Selection

THE coils described in this chapter are used in air conditioning systems for heating or cooling an air stream under forced convection. The surface coil equipment may be made up of a number of banks assembled in the field, or the entire assembly may be factory constructed. The applications of each type of coil are limited to the field within which it is rated. Other limitations are imposed by code regulations, by proper choice of materials for the refrigerants used and the condition of the air handled, or by an economic analysis of the possible alternates on each installation.

For heating service, these coils are used as preheaters, reheaters or booster heaters, (see Chapters 21 and 22). The function of the coils is air heating only, but the apparatus assembly may include means for humidification and air cleaning. Steam or hot water are the usual heating media, although others are used in special cases, such as reheating by means of discharge gas from a refrigerating system.

Coils are used for air cooling with or without accompanying dehumidification. Examples of cooling applications without dehumidification are precooling coils using well water or other relatively high temperature water to reduce the load on the refrigerating machinery, or water cooled coils to remove sensible heat in connection with chemical moisture-absorption apparatus. By proper coil selection it is possible to handle both sensible cooling and dehumidification together as further explained later. The apparatus assembly usually includes an air cleaning means to protect the coil from accumulation of dirt and to keep dust and foreign matter out of the conditioned space. Although cooling and dehumidification are the usual functions, there are cases of cooling coils purposely wetted as an aid to air cleaning and odor absorption.

The usual cooling media used in surface coils are cold water and volatile refrigerants such as dichlorodifluoromethane and methyl chloride, but others are used in special cases. Brines are seldom required for the range of applications covered by this chapter, although there are cases where low entering air temperatures with large latent heat loads require a refrigerant temperature so low that water becomes impractical. Some-

times, also, brine from an industrial system already installed is the only convenient source of refrigeration.

For combined cooling and dehumidifying, surface coils present an alternate to spray dehumidifiers. For many applications it is possible, by proper selection of apparatus, choice of air velocities, refrigerant temperatures, etc., to perform the same duty with either. In a few cases both sprays and coils are used. The coils may then be installed within the spray chamber, either in series with the sprays or below them. In making the selection between spray and surface dehumidifiers, certain advantages of each should be considered. The fact that a spray dehumidifier is usually designed to deliver nearly saturated air tends to simplify the control problem. In this case the dry-bulb temperature is also the dew-point, and hence a dew-point control can be arranged by using a simple duct thermostat. Spray dehumidifiers have the advantage over unwetted coils of a certain degree of air cleaning and odor absorption. On the other hand, coils make possible a closed and balanced cooling water circuit, obviating the unbalanced pumping head, the complication of water level control, and danger from possible floods incidental to multiple-spray dehumidifiers, especially if located on different levels. The use of coils often makes it possible for the same surface to serve for summer cooling and winter heating by circulating cold water in the one season and hot water in the other, with consequent saving in apparatus and piping. Surface-coil dehumidifiers seldom deliver saturated air, and wet-bulb depression of 0.5 to 4 F (or more) is usual. Another advantage is that where the surface coil system can be used with direct expansion of refrigerant, it is comparatively low in initial and operating costs. Of course the safety of the occupant must be kept in mind in comfort conditioning applications. Some localities have refrigeration codes which restrict the use of direct-expansion coils in the air stream, and hence local codes should be consulted by the engineer before a system employing direct expansion methods is designed. The choice between spray dehumidifiers and coils depends upon the necessities and the economic aspects of each case and no general rule can be given. There are many installations in which either can be used.

COIL CONSTRUCTION AND ARRANGEMENT

Coils are basically of two types, those consisting of bare tubes or pipe and those of *extended* surface construction. The former are little used for the applications covered by this chapter, but are often employed where conditions cause frost accumulation, and for cooling surface within spray dehumidifiers.

The heat transmission from air passing over a tube to a refrigerant flowing within it is impeded by three resistances. The same is true when the air is being heated by steam or hot water in the tube. The first resistance is from the air to the surface of the tube, usually called the outside surface resistance or air-film resistance. Second is the resistance to the flow of heat by conduction through the metal itself. Finally there is another surface or film resistance to the flow of heat between the inside surface of the metal and the fluid in the tube. For the applications under consideration both the resistance of the metal wall to heat conduction, and the inside surface or film resistance are usually low as compared with the air-side surface resistance. This is especially the case where sensible

heating or cooling only is accomplished. Where dehumidification accompanies sensible cooling, or where the external surface of the tube is sprayed with large quantities of water, the resistance to heat flow between the tube and the air flowing over it is much decreased. In the case of the water spray, the surface resistance depends on the amount and the method of application of the water. Economy in space, weight and cost make it advantageous to decrease the external surface resistance, where it is proportionately large, to approach that of the tube wall, and that from tube to refrigerant. This is accomplished by increasing the external surface by means of fins. With water spray the external resistance is already low, and the fins are less useful for increasing the overall heat transfer. Sometimes water spray is applied to the same type surface as would have been used without it. The overall heat transfer is not necessarily increased much by such an arrangement, but the water spray may serve other purposes than to increase the flow of heat, such as air and coil cleaning.

In fin or *extended* surface coils the external surface of the tubes is known as *primary* and the fin surface is called *secondary*. The primary

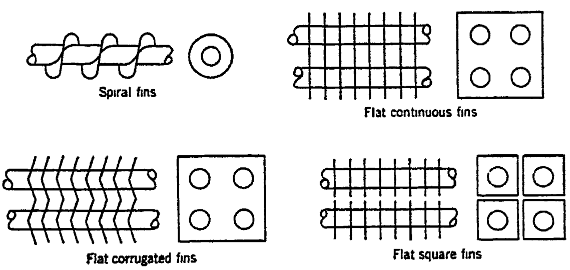

FIG. 1. TYPES OF FIN COIL ARRANGEMENT

surface consists generally of round tubes or pipes. In some cases these are staggered and in others in line with respect to the air flow. The staggered arrangement gives a somewhat higher heat transfer value but also a higher resistance to air flow and in some cases makes the header and return bend arrangement more complicated. A number of types of fin arrangement are used, the most common of which are spiral, flat and flat-crinkled or corrugated, all as shown in Fig. 1. While the spiral fin surrounds each tube individually in all cases, the flat types may be continuous (including several rows of tubes), or they may be round or square, with individual fins for each tube. All of these, as well as other less common types, are in use, the selection for a particular installation being based on economic considerations, space requirements and resistances of individual designs of coils. A most important factor in the performance of extended surface coils is the bond between the fin and the tube. An intimate contact is assured in a number of ways. The assembled coil may be coated with tin, zinc, etc., after fabrication. The spiral type fin may be knurled into a shallow groove on the exterior of the tube. The tube may be expanded after the fins are assembled, or the tube hole flanges of a flat or corrugated fin may be made to override those in the preceding

fin and so compress them upon the tube. There are also types of construction where the fin is formed out of the material of the tube itself. In any case the successful performance of a fin surface depends upon the bond between fin and tube being secure and remaining so in service.

For heating coils the materials most generally used are copper, steel and aluminum. Sometimes aluminum or brass fins are used on copper tubes. Steel is uncommon except in special cases. Some types of heating coils are made of cast-iron. There are sufficient practical installations of each of these to demonstrate that they can all give good service. However for equal performances brass and aluminum fins must be of greater thickness than copper fins on account of their lower coefficients of conduction. The copper coils are frequently tin-dipped and steel coils galvanized to protect them from corrosion and to assure a bond between fin and tube.

Cooling coils for water or for volatile refrigerants are most frequently of copper, both fin and tube. Aluminum fins on copper tubes are also used. For brines such as sodium or calcium chloride and for ammonia, steel fins and tubes are common.

Although there are many variations for special cases, tube and fin sizes and spacings for air conditioning coils, both heating and cooling, fall within fairly narrow limits. The tubes are usually 3/8, 1/2, 5/8, or 3/4 in. OD, and the fins spaced from 4 to 8 per inch, 6 per inch being a common design. The tube spacing generally varies from about 1 1/8 to 2 in. on centers. Small tube size and close fin spacing give large capacity with small space demand, but the resistance, both over the surface and through the tubes, is higher than with larger tubes and more widely spaced fins. Moreover, too close a fin spacing may result in trouble from dirt accumulation, especially on dehumidifying coils, and may also cause trouble from water *hold-up* between the fins, particularly with air flow vertically upward. This condition increases the air resistance and decreases the capacity of the coil. Water hold-up sometimes causes flooding trouble in vertical air flow units by accumulating too much water for the drain to handle all at once when the fan is stopped.

Steam Coils

For proper performance of steam heating coils, condensate and air must be continually eliminated and the steam must be evenly distributed to the individual tubes. This distribution is usually accomplished by individual orifices in the tubes, by distributing plates and orifice in the steam header, or by perforated internal steam-distributing pipes extending into the individual tubes. The latter arrangement has the advantage of distributing the steam throughout the length of each tube, and is conducive to uniform delivered air temperatures. The tendency for freezing of condensate at the bottom of the coil with cold entering air and light heating loads is also minimized. This is especially valuable for outside air preheaters. Methods of air and condensate elimination are discussed in detail in Chapters 14, 15 and 22.

Water Coils

The performance of water coils, for heating or cooling, depends on the elimination of air from the system and proper distribution of water. Air elimination is taken care of in the system piping as described in Chapter

16. To assure a pressure drop sufficient for adequate distribution but at the same time to provide against excessive pumping head where large water quantities are handled, water coils are provided with various water circuit arrangements. For instance, a typical coil 18 tubes high and 6 tubes deep in the direction of air flow can be arranged for 6, 9, 18 or 36 parallel water circuits as conditions may require. Orifices in individual tubes are occasionally employed but are usually unnecessary as the resistance of individual water circuits is generally sufficient to effect a satisfactory distribution. In cases such as well water precooling coils, where there may be considerable sand and other foreign matter in the water, provision for cleaning of individual tubes is of advantage. It is important to arrange water coils for drainage if located where they will be

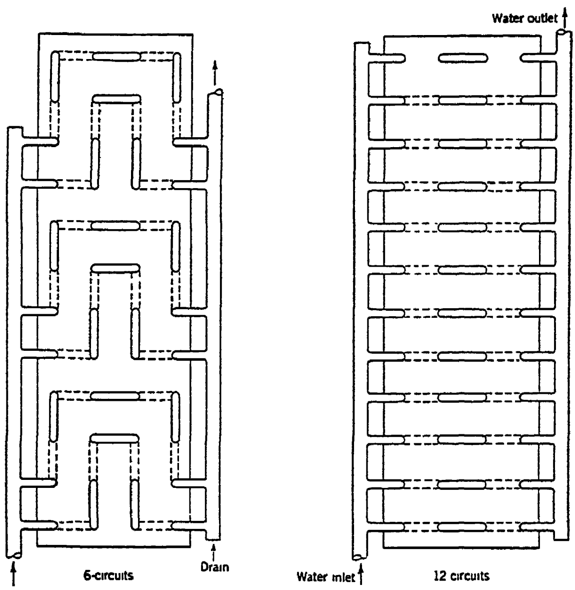

FIG. 2. VARIOUS WATER CIRCUIT ARRANGEMENTS

exposed to freezing. For this reason the circuits should be so laid out that there are no pockets to hold water. Fig. 2 shows such construction. The drains may be provided in the water piping although they are often arranged in the coil headers.

Direct-Expansion Coils

Coils for volatile refrigerants present more complex problems of fluid distribution than do water, brine or steam. It is desirable that the coil be effectively and uniformly cooled throughout, and necessary that the compressor be protected from entrained, unevaporated refrigerant. There are two types; namely, flooded systems, and thermal expansion valve systems, as shown in Figs. 3 and 4. With flooded control the coils are supplied with liquid by the same type of circulation that exists in a water tube boiler, while the level in the surge drum is maintained by the action

of the float regulator, or by properly charging the plant in the case of the high pressure float drainer. The thermal expansion valve system depends upon the thermal valve automatically feeding just as much liquid to the coils as is required to maintain the superheat at the coil suction outlet within predetermined limits which vary from about 6 to 10 F. The

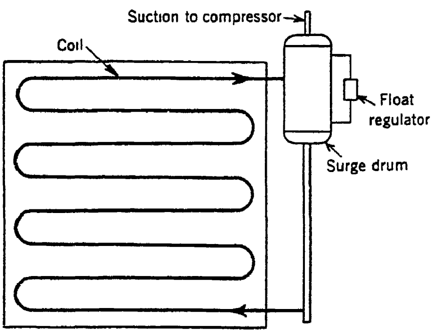

Fig. 3. Direct-Expansion Coil with Flooded System

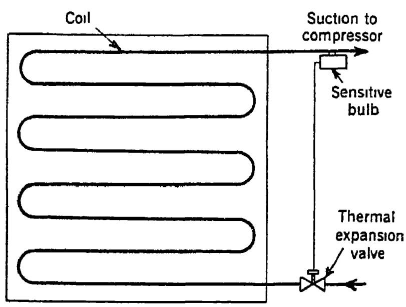

Fig. 4. Direct-Expansion Coil with Thermal Valve System

thermal valve arrangement is in common use for the type of coils covered by this chapter, while the flooded system is comparatively rare.

With the flooded system the refrigerant distribution through the tubes depends on properly selecting the length of the feeds and the head of liquid imposed upon the liquid inlets. No auxiliary distributing devices

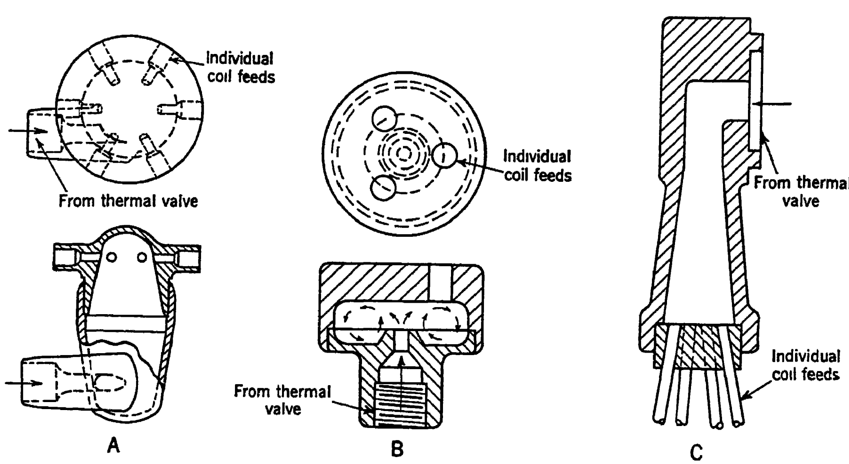

Fig. 5. Types of Refrigerant Feed Distributing Heads

are required. With the thermal valve system there are two factors to consider. There must be, generally, more than one refrigerant feed through the coil per thermal valve to keep the pressure drop through the refrigerant circuit within practical limits and to reduce the corresponding penalty in increased evaporating temperature. At the same time the

coil must be so arranged that the required suction superheat can be attained with a minimum sacrifice in the performance of the coil as a whole. It is general practice to attain this superheat within the coil itself and not by the use of external heat exchangers or other auxiliary devices.

With thermal expansion valves it is advantageous to keep the pressure drop through the refrigerant feeds as low as possible. The feeds are laid out to expose each to the same mean temperature difference so that it handles the same refrigerating load. A distributing means is imposed between valve and coil liquid inlets to divide the refrigerant equally among the feeds. Such a distributor shall be effective for distributing both liquid and vapor, since the entering refrigerant is a mixture of the

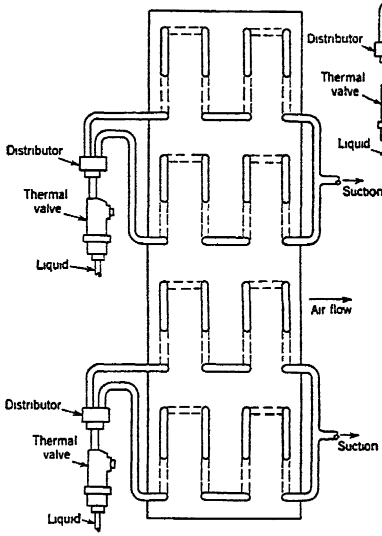

FIG. 6. ARRANGEMENT FOR FACE CONTROL

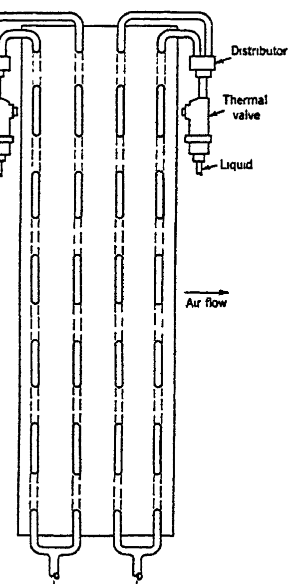

FIG. 7. ARRANGEMENT FOR DEPTH CONTROL

two. Fig. 5 shows three typical types of distributors. In distributor *A* the liquid and gas mixture from the thermal valve is led tangentially into a chamber. The coil feed connections extend outward radially at the top of this chamber. In distributor *B* the refrigerant is discharged at a high velocity through a central jet against the end plate, forming a uniform mixture of gas and liquid within the distributor, from which individual connections are led as shown. In type *C* the refrigerant enters at high velocity from the thermal valve and is discharged against the end plug in which the individual liquid feeds are closely arranged. These distributors can be used in either vertical or horizontal position. Although there are other forms of distributors the above are typical examples. The individual liquid connections from the distributor to the coil inlet are commonly made of small diameter tubing and are all of the same length

and diameter in order to impose the same friction between the distributor and the coil. Since the thermal valves act in response to the superheat at the coil outlet, this superheat should be produced with the least possible sacrifice of active evaporating surface. Sometimes a single thermal valve is used per coil. In other cases multiple valves are used, with the coil divided across the air flow or parallel to the air flow as shown in Fig. 6. The arrangement of Fig. 7 should be avoided since it offers the disadvantage of unequal load on the two parallel circuits.

Flow Arrangement

The relative direction of flow of the air outside the tubes and the medium within them influences the performance of the surface. There are three types of relative flow in common use. Fig. 8A shows parallel-flow in which the air and the medium in the tubes proceed through the coil in the same direction. Fig. 8B shows counter-flow in which the

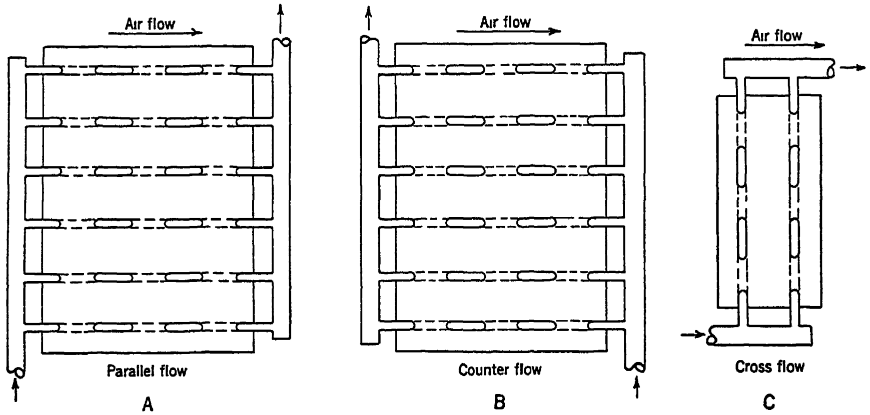

FIG. 8. FLOW OF MEDIA IN TUBES IN RELATION TO AIR FLOW

medium in the tubes proceeds in a direction opposite to the flow of air. Fig. 8C shows cross-flow in which the air and the medium in the tubes pass at right angles to each other. Parallel flow is seldom used for the reason that a lesser mean temperature difference results than with counter-flow. The counter-flow arrangement is almost universally used in brine or water coils to take advantage of the highest possible mean temperature difference for given entering water and air temperatures. It is also invariably used in coils fed with volatile refrigerant to take advantage of the higher air temperature for superheating the leaving gas. This arrangement assists complete evaporation and superheating of the refrigerant which is essential to proper operation of the thermal expansion valve. Cross-flow is common in steam heating coils, the temperature within the tubes being substantially uniform and the mean temperature difference the same whatever the direction of flow, relative to the air. Cross-flow is to be avoided in coils with volatile refrigerants on account of unequal loading of parallel circuits and danger of short circuiting of liquid refrigerant which will disturb proper functioning of the thermal expansion valve.

Applications

Heating coils in field assembled banks are used for a number of purposes as described in Chapter 21. They may be arranged with the air flow vertical or horizontal, although the latter is more common. For steam heating the coils may be set with the tubes vertical or horizontal. In the latter case the coil should be sloped to provide for condensate drainage. Because of the multi-circuit feed arrangement and the necessity for avoiding air and water pockets, water heating coils are generally arranged with the tubes horizontal. Certain precautions must be taken against freezing. Where steam coils are used with entering air below freezing temperature, throttling the steam supply may result in freezing the condensate in the bottom of the coil if the tubes are of the variety not provided with internal distributing pipes, or an equivalent arrangement.

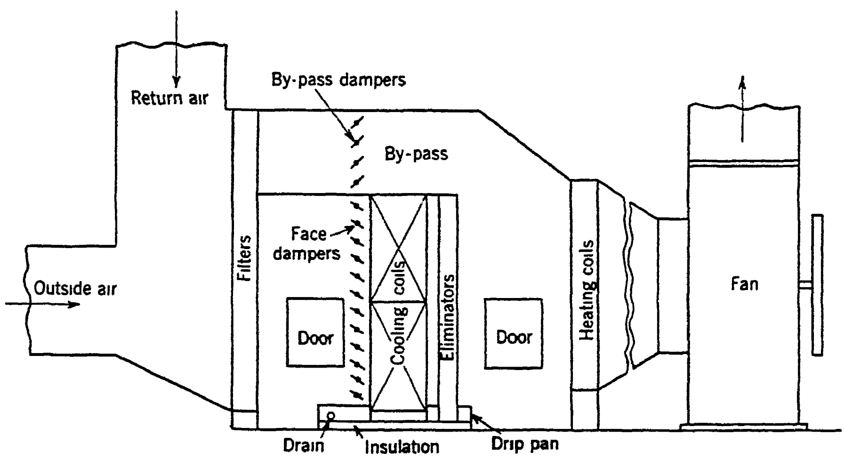

FIG. 9. TYPICAL ARRANGEMENT OF COOLING COILS IN A CENTRAL SYSTEM

If these are used, there is little danger of freezing the condensate as long as the leaving air temperature is not allowed to fall below about 40 F. As an added precaution with both steam and water coils the outside air inlet dampers are often closed automatically when the fan is stopped to avoid trouble caused by very cold outside air drifting in during *off* periods.

A typical arrangement of water cooling coils is shown in Fig. 9. Some means should be provided to filter all the entering air to keep dirt and foreign matter from accumulating on the coils. The assembly is provided with a drip-pan to catch the condensate during summer dehumidifying duty and to collect the non-evaporated water from the humidifying sprays in winter. The drip connection should be made ample in size and liberally provided with plugged tees and crosses for cleaning. It should not be exposed to freezing temperatures in winter if the apparatus is used on winter humidifying duty. Access doors should be provided for servicing filters, humidifying nozzles, and fan bearings and for cleaning the coils. With certain designs of coils when used for dehumidifying, eliminators must

be used beyond the coil to catch any water which may be blown into the air stream. It is customary to include these eliminators when the air velocity exceeds about 450 fpm with the individual fins and about 600 fpm for the continuous flat fin type. Where a number of coil sections are stacked one upon another, and where the velocities are low, so that eliminators need not be used, occasional trouble results when water splashes down from one coil to the next and blows out into the air stream. In such cases drip troughs as shown in Fig. 10 are used to collect this water and conduct it to the condensate pan.

Sometimes finned surface coils on summer cooling and dehumidifying duty are provided with water sprays. These sprays are of two types. In the first type a set of spray nozzles is arranged for intermittent cleaning. The operator can wash the coils off as frequently as necessary. These

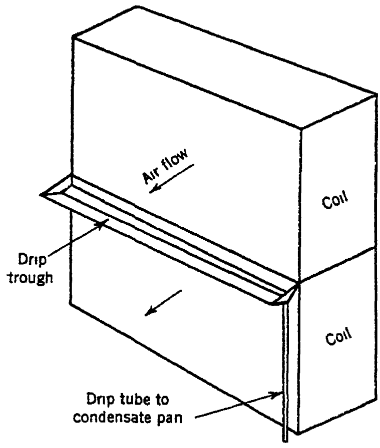

FIG. 10. COIL ARRANGED WITH DRIP TROUGH

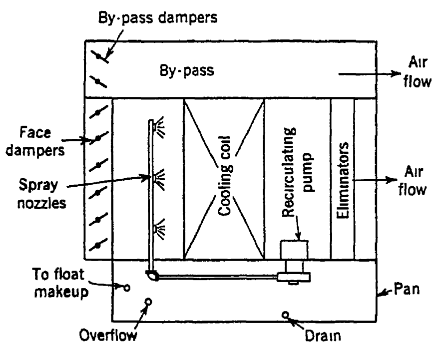

FIG. 11. RECIRCULATING SPRAY SYSTEM FOR CLEANING COILS

sprays are not operative when the system is in use and no recirculating pump is provided. The second arrangement requires a collecting tank and a recirculating pump. The water is in circulation whenever the apparatus is in operation, and assists in keeping the coil clean and in absorbing odors. Fig. 11 illustrates such an arrangement. Wherever air by-passes are used around a coil on summer duty for control purposes, it is of advantage to direct only return air through the by-pass rather than a mixture of return and outside air. The casing should be arranged accordingly. To maintain the air quantity handled by the fan reasonably constant, and to assure the required design quantity of by-passed air when the by-pass damper is open, cooling coil banks are frequently furnished with both face and by-pass dampers as shown in Fig. 9.

Although both heating and cooling coils are made of sufficient strength to take up expansion and contraction arising within themselves, care should be taken to avoid imposing strains from the piping on to the coil connections. (See Chapters 15 and 16).

HEAT TRANSFER AND AIR FLOW RESISTANCE

The transfer of heat between the heating or cooling medium and the air stream is influenced by several variables:

1. The temperature difference.
2. The design and surface arrangement of the coil.
3. The velocity and character of the air stream.
4. The velocity and character of the medium in the tubes.

The driving force is usually taken as the logarithmic mean temperature difference for heating or cooling without dehumidification. For combined cooling and dehumidification, a special measure of the propelling force is used as described later. Logarithmic differences are generally employed in practice although there are special flow relationships used, such as cross-flow, where they do not strictly apply. With volatile refrigerants there is often an appreciable pressure drop and corresponding change in evaporating temperature through the refrigerant circuit. The problem is further complicated by the fact that the refrigerant is evaporating in part of the circuit and superheating in the remainder. In spite of this, heat transfers and ratings for coils using volatile refrigerants are usually based in practice on a refrigerant temperature corresponding to the average pressure in the coil.

The design and surface arrangement of the coil includes such items as materials, type, thickness, height and spacing of the fins, and the ratio of this surface to that of the tube, the use of the staggered or in-line tube arrangement, and provisions to increase the air turbulence such as the use of corrugated as against flat fins. Staggered tubes increase the total heat transfer as against the in-line arrangement and corrugated fins are more effective than flat. Of especial importance is the bond between fin and tube.

The velocity of the air usually considered is the coil face velocity. This bears a varied relation to the actual velocity over the surface, depending upon the individual coil design. As long as a fixed design of coil is under consideration face velocities may be used, but they may be unsatisfactory in comparing different designs, as it is the actual surface velocity that is significant. The air volume is often based on standard air at 70 F and a barometric pressure of 29.92 in. Hg. The use of air volume in coil rating information may be misleading. The significant value is mass velocity in pounds per minute and not cubic feet per minute, because for a fixed volume the corresponding weight may vary widely, depending upon the temperature and barometric pressure under consideration.

At the same mass air velocity, varying performance can be obtained depending upon the turbulence of the air flow into the coil and upon the uniformity of distribution of air over the coil face. The latter is very important in obtaining reliable test ratings and in realizing rated performance in practical installations. The resistance through the coils will assist in properly distributing the air, but where the inlet duct connections are brought in at sharp angles to the coil face, the effect is frequently bad and there may even be reverse air currents through the coils. This

reduces the capacity, but can be largely avoided by proper layout or by the use of directing baffles.

The heat transfer depends also upon the velocity of the medium in the tubes and upon its character, whether flowing water, condensing steam or evaporating volatile refrigerant. Heat transfer rates expressed as Btu per square foot of internal surface per degree logarithmic mean effective temperature difference between the fluid and tube wall are, for example, about 150 to 300 for evaporating dichlorodifluoromethane, about 350 to 1200 for water at 2 and 6 fps and about 1200 for condensing steam. The influence of the medium in the tubes on the overall heat transfer rate is, therefore, apparent.

Because of these variables, reliable rating and performance information for any design of coil must be based on actual tests on that coil under the expected conditions of operation. A comparison between the performance of two designs, unless based on such tests on each, may lead to entirely erroneous conclusions.

PERFORMANCE OF HEATING AND COOLING COILS

Heating and cooling coils are essentially heat exchangers and as such their performance depends in general upon:

1. The overall coefficient of heat transfer from the fluid within the coil to the air it heats or cools.
2. The mean temperature difference between the fluid within the coil and the air flowing over the coil.
3. The physical dimensions of the coil.

Thus, for any one definite operating condition, the heating or cooling capacity of a given coil is expressed by the following basic formula:

$$Q = U \times MTD \times A \qquad (1)$$

where

Q = total heat transferred by the coil, Btu per hour.

U = overall coefficient of heat transfer, Btu per hour per square foot of external coil surface per degree Fahrenheit temperature difference between the fluid within the coil and the air flowing over the coil.

MTD = mean temperature difference, degrees Fahrenheit between the fluid within the coil and the air passing over it. (This is commonly taken as the logarithmic mean temperature difference. See Chapter 47).

A = external surface area of the given coil, square feet.

The performances of heating and cooling coils are influenced by the same factors in all but one very important exception, that is, when cooling coils operate wet or act as dehumidifying coils. For this reason, in the discussion which follows, heating and dry cooling coils are treated as one group and dehumidifying coils as another.

OVERALL COEFFICIENT OF HEAT TRANSFER

Of all factors affecting the performance of heating or cooling coils, the overall coefficient of heat transfer is the most difficult to determine as it in itself is influenced by several factors depending upon coil design and conditions of operation.

Considering any coil, whether of bare pipe or of finned type, the overall heat transfer coefficient for a given size and design of coil can always be considered as a combined effect of three individual heat transfer coefficients, namely:

1. The film coefficient of heat transfer between air and the external surface of the coil, usually given in Btu per hour per square foot external surface per degree Fahrenheit mean temperature difference.
2. The coefficient of heat transfer through the coil material—tube wall, fins, ribs, etc.
3. The film coefficient of heat transfer between the internal surface of the coil and the fluid flowing within the coil, usually given in Btu per hour per square foot internal surface per degree Fahrenheit mean temperature difference.

These three individual coefficients acting in series result in an overall coefficient of heat transfer in accordance with the basic laws. For a *bare pipe* coil the overall coefficient of heat transfer, whether for heating or for cooling (dry), can be expressed by a simplified basic formula as follows:

$$U = \frac{1}{\frac{R}{f_r} + \frac{X}{k} + \frac{1}{f_a}} \tag{2}$$

where

U = overall coefficient of heat transfer, Btu per hour per square foot external surface per degree Fahrenheit mean temperature difference between air and fluid within the coil.

f_r = film coefficient of heat transfer between the internal surface of the coil and the fluid flowing within the coil, Btu per hour per square foot internal surface per degree Fahrenheit mean temperature difference between that surface and the average fluid temperature.

f_a = film coefficient of heat transfer between air and the external surface of the coil, Btu per hour per square foot external surface per degree Fahrenheit mean temperature difference between the mass of air and the external surface.

k = conductivity of material from which the bare pipe is constructed, Btu per hour per square foot per degree Fahrenheit per inch thickness.

X = thickness of tube wall, inches.

R = ratio between external and internal surface of the bare tube, usually varying from 1.03 to 1.15 for the tube used in typical heating or cooling coils. This ratio R is inserted in the formula in order to place internal fluid coefficient of heat transfer on the basis of external surface.

Frequently, when pipe or tube walls are thin and of material having high conductivity (as is the case in construction of typical heating and cooling coils) the term X in Equation 2 becomes negligible and is generally disregarded. (The effect of the term X in typical bare pipe heating or cooling coils seldom exceeds 1 to 2 per cent of the overall coefficient). Thus, in its simplest form, for bare pipe:

$$U = \frac{1}{\frac{R}{f_r} + \frac{1}{f_a}} \tag{3}$$

For finned coils the formula[1] for the overall coefficient of heat transfer can be conveniently written:

$$U = \frac{1}{\frac{R}{f_r} + \frac{1}{zf_a}} \tag{4}$$

[1]Rational Development and Rating of Extended Air Cooling Surface, by H. B. Pownall (*Refrigerating Engineering*, October, 1935, p. 211).

in which the term z, called the *fin efficiency*, is introduced to allow for the resistance to heat flow encountered in the fins.

The term R, in this case, is the ratio of *total* external surface to internal surface. For typical designs of finned coils for heating or cooling, this ratio varies from 10 to 30. Term R is again introduced to place the internal surface coefficient of heat transfer on a basis of external surface. In the discussions which follow, coefficients f_r and zf_a will be considered separately, and also various ways of combining them will be outlined.

External Film Coefficient

While formulas have been developed expressing the film coefficient f_a for air passing parallel to a plane surface, they cannot be used directly for fins on tubes because of air turbulence and because of the temperature gradient prevalent from the edge of a fin to its center. It is therefore necessary to make tests to evaluate the combined term zf_a. The term, zf_a, will be written merely f_a in this discussion as there is no necessity for separately evaluating z and because values of f_a are usually applied only to the particular coils for which tests are made.

Transfer of heat from a fluid to a solid is accomplished by the contacting of the molecules of the fluid with the solid. When a molecule strikes a solid, its energy level equalizes with the energy level of the solid. The total amount of heat exchanged between the molecules of a fluid and a solid is determined by the number of contacts per unit of surface per unit of time, and by the energy change of the fluid.[2] The energy change, in the case of air, is measured by the temperature change times the specific heat of the air. The number of contacts is measured by a percentage of the weight of air flowing per unit of time.

In the case where water vapor is mixed with air, and the water vapor is cooled but not condensed, the amount of heat transferred is increased by the energy change of the vapor particles. The additional energy is measured by the temperature change, by the specific heat of the water vapor, and by the weight of vapor contacting the surface per unit of time. In a mixture of air and vapor there is a definite ratio between the weight of the vapor and of the air per cubic foot of the mixture. Therefore, as the temperature of the mixture is lowered, the amount of heat lost by the vapor always bears a definite ratio to the amount of heat lost by the air. The amount of energy involved in the temperature change of the vapor is small, however, and it is usually included with that of the air by using a value of 0.245 for the specific heat of humid air.

Dehumidification of air by a cooling coil occurs whenever the surface temperature of the fins and tubes is below the dew-point temperature of the air. Enough molecules of water vapor are condensed on the coil to create a state of equilibrium between the vapor pressure of the moisture on the coil surface and the vapor pressure of the moisture in that part of the air stream which is in immediate contact with the coil surface. Because of the good contact between the condensed film of water and the coil surface, the water film attains a temperature approaching that of the coil surface. Therefore, those particles of air which actually contact the

[2]Graphical Method of Determining Finned Coil Capacities Described, by E. P. Wells (*Heating, Piping and Air Conditioning*, December, 1936, p. 665).

water film leave the film with a dew-point temperature equal to the outer surface film temperature. However, many air particles, with their attendant water vapor particles, never contact the coil surface, but are by-passed between the fins. These air particles have the same dew-point temperature when they leave the coil as they had when they entered, but after leaving the coil they mix with the air particles which did contact the surface, producing a mixture of air which has a dew-point temperature that lies between the original dew-point temperature and the film surface temperature. This process explains why air seldom leaves a coil in a saturated condition.

The foregoing contact-mixture concept of heat transfer has been found by several independent investigators to be consistent with experimental data. The concept has been used successfully in analyzing the performance of evaporative condensers, cooling towers, condensers and evaporators. A relation has been found between heat transfer and pressure drop of flowing fluids, by assuming that molecules of a fluid lose their momentum upon contact with a solid.[3]

The fact that a coil starts to condense moisture when the surface temperature drops below the dew-point temperature of the entering air makes it possible to measure the surface temperature of a coil, an otherwise practically impossible task. After the surface temperature has been determined, it is possible to analyze completely the surface film coefficient of both the air side and refrigerant side of a coil.

The air side coefficient, f_a, of a dry coil of particular dimensions is an exponential function of the mass velocity of the air:

$$f_a = C\,w^n \tag{5}$$

where

f_a = film coefficient of heat transfer, Btu per hour per square foot external surface per degree Fahrenheit mean temperature difference between air and average surface temperature.

w = air mass velocity, pounds per hour per square foot of coil face area.

C and n = constants which depend upon air turbulence, the number of square feet of external surface per square foot of coil face area, and the depth of the coil.

The difficulty of obtaining sufficient tests to evaluate the constants C and n for all conditions of coil design and operation makes it desirable to use Equation 6 for determining the air side coefficient:

$$f_a = 0.245 \times \frac{w}{a} \times 2.3 \times \log_{10}\left(\frac{1}{1\text{-}E}\right) \tag{6}$$

where

0.245 = specific heat of humid air, Btu per pound per degree Fahrenheit.

2.3 = the constant which converts logarithms from base e to base 10.

a = external surface area, square feet per square foot of coil face area.

E = coil efficiency, a decimal less than 1.0.

This formula gives values of f_a after tests have been made to evaluate the coil efficiency. Equation 6 can be derived[4] by combining the basic

[3]The Contact-Mixture Analogy Applied to Heat Transfer with Mixtures of Air and Water Vapor, by W. H. Carrier (*A.S.M.E. Transactions*, January, 1937, Vol. 59, No. 1, p. 49).

[4]Loc. Cit. Note 2.

equations of heat transfer, mean temperature difference and coil efficiency:

$$Q_s = f_a \times a \times MTD_a \tag{7}$$

$$MTD_a = \frac{t_1 - t_2}{2.3 \log_{10}\left(\frac{t_1 - t_s}{t_2 - t_s}\right)} \tag{8}$$

$$E = \frac{t_1 - t_2}{t_1 - t_s} \text{ (by definition)} \tag{9}$$

$$Q_s = 0.245 \times w \times (t_1 - t_2) \tag{10}$$

where

Q_s = sensible heat transferred, Btu per hour per square foot of coil face area.
t_1 = temperature of air entering coil, degrees Fahrenheit.
t_2 = temperature of air leaving coil, degrees Fahrenheit.
t_s = average temperature of coil external surface, degrees Fahrenheit.
MTD_a = logarithmic mean temperature difference between air and coil surface.

Coil Efficiency

One method of expressing air-coil contact efficiency is the ratio between the weight of air that actually contacts the coil surface and the total weight of air passing through the coil. Due to the fact that the specific heat of air is fairly constant over a wide range of temperature, coil efficiency[5] can be expressed as equal to the number of degrees that the entire amount of air is cooled, divided by the number of degrees between the entering air temperature and the coil surface temperature.

For a particular heat transfer surface, coil efficiency is only a function of the mass velocity of the air, which may be observed by equating Formulae 5 and 6 and combining all constants into D and u:

$$\log_{10}\left(\frac{1}{1-E}\right) = \frac{D}{w^u} \tag{11}$$

This equation can be used in graphical form by plotting coil efficiency against mass velocity as shown in Fig. 12. The significance of coil efficiency can be visualized in Fig. 13, where the length of the line *C-D*, divided by the length of line *C-E*, measures the coil efficiency. The relation between coil capacity and coil efficiency is given by:

$$Q = Ew\,(h_1 - h_s) \tag{12}$$

where

h_1 = specific enthalpy of air entering coil, Btu per pound.
h_s = specific enthalpy of saturated air at surface temperature, Btu per pound.

When no latent heat is being removed from air, the change in enthalpy is equal to the change in temperature times the specific heat, so Equation 12 can be changed to:

$$Q = Ew\,(t_1 - t_s)\,0.245 \tag{13}$$

Dehumidification of Air

When moisture is being condensed on the coil surface Equation 12 can be used. If a coil has an efficiency of 0.8 (80 per cent) for the removal

[5]When coil efficiency is used herein it is intended to express air-coil contact efficiency and does not express total performance efficiency.

of sensible heat, it will at the same time remove 80 per cent of the difference in moisture content between the entering air and saturated air at the surface temperature. This is due to the fact that 80 per cent of the air particles contact the surface and attain a dew-point temperature equal to the surface temperature. This condition is expressed graphically in Fig. 13.

This psychrometric chart is constructed so that equal increments along the horizontal axis represent equal changes in sensible heat content, and

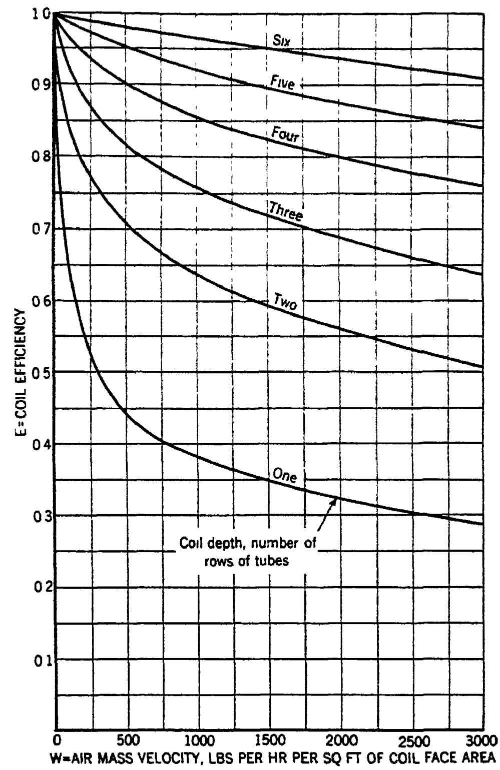

Fig. 12. Relation of Coil Efficiency to Mass Velocity

equal increments along the vertical axis represent equal changes in latent heat content of air. Point *A* represents the condition of return or recirculated air, point *B* that of outside air, point *C* the mixture of two-thirds recirculated air and one-third outside air, and point *E* the average surface temperature. Point *D*, which represents the air leaving the coil, lies on a line which connects points *C* and *E*, and its distance from point *C* is equal to the length of the line *C-E* times the coil efficiency. The ratio between the vertical distance from *C* to *D* and the horizontal distance from *C* to *D*, expressed in heat units, is the ratio between latent heat and sensible heat removed. It can be shown by trigonometric relations that the slope of

the line *C-D* is a measure of the ratio of latent to total heat removed, and that any line parallel to *C-D* gives the same heat ratio.

To enhance the practical usefulness of the psychrometric chart illustrated in Fig. 13, a set of marked master slope lines is included. The value of this arrangement is easily illustrated by the graphical example shown.

Example 1. To determine the required average effective external coil surface temperature. *Given:* (1) Air entering cooling coil at temperature of 83 F dry-bulb and 69 F wet-bulb. (2) Ratio of latent to total heat that must be removed from air is 35 per cent. *Required:* To find the average external coil surface temperature.

Solution. (1) Draw through point *N*, at the origin of the heat load ratio lines, a line *N-O* with a slope of 35 per cent in accordance with scale *S*. (2) Mark in the body of the

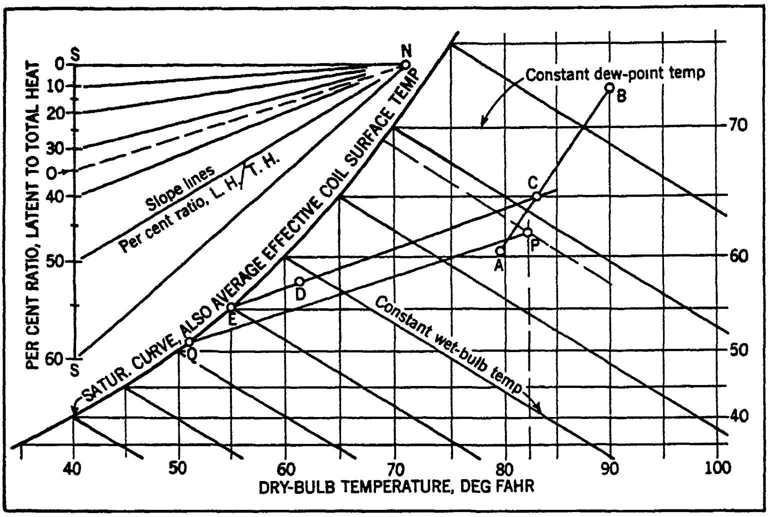

FIG. 13. PSYCHROMETRIC CHART SHOWING STRAIGHT-LINE METHOD FOR REPRESENTING COIL PERFORMANCE

chart, point *P* representing the condition of air entering the cooling coil at 83 F dry-bulb and 69 F wet-bulb. (3) Through point *P* draw a line *P-Q* parallel to line *N-O*. (4) The line *P-Q* intersects the saturation curve at 51 F, which means that the effective external coil surface temperature must be maintained at 51 F in order to obtain the desired 35 per cent latent to total ratio of heat removal from the air passing over the given cooling coil.

Inspection of Equation 12 reveals that the total capacity of a coil is dependent on the entering and leaving wet-bulb temperatures. The entering dry-bulb temperature is unimportant.

The amount of latent heat of condensation of a coil can be calculated from:

$$Q_l = 1060\,Ew\,(W_1 - W_s) \tag{14}$$

where

Q_l = latent heat removed, Btu per hour per square foot of coil face area.
W_1 = pounds of moisture per pound of dry air entering the coil.
W_2 = pounds of moisture per pound of dry air saturated at the average surface temperature.
1060 = average value of latent heat of water vapor, Btu per pound of vapor.

The amount of sensible heat removed can be obtained by subtracting the value of Q_l from the value of Q in Equation 12.

Equation 12 gives accurate results when it is used for coils having a small change of temperature of the fluid in the tubes, as for example with evaporating refrigerants and with water having a small temperature rise. In cases where water in the tubes has a large temperature rise, the effective surface temperature changes throughout the depth of the coil, and in extreme cases moisture may be condensed on only a portion of the coil. In such cases it is possible to estimate the wet and dry portions of the coil separately, using cut-and-try methods.[6]

Internal Film Coefficient

The internal film coefficient, f_r, which appears in Equation 3, is evaluated in various ways, depending upon the nature of the fluid, and whether the fluid is changing state.

When evaporating refrigerants are being used in tubes, the temperature of the fluid is fairly constant, being affected principally by pressure drop through the tubes, by superheat of the evaporated refrigerant, and by the presence of oil in solution. To obtain maximum coil capacity it is necessary to keep the pressure drop through the tubes at a minimum (1/4 lb per square inch), to keep the superheat as low as possible without carrying liquid back to the compressor, and to arrange for good separation and return of oil to the compressor. An additional important factor is the removal of gas so that the tube surface may be flooded with liquid as much as possible. The internal film coefficient is markedly increased by heavy heat loads, because the increased turbulence and gas velocity cause good contact of the liquid with the tubes. Values of f_r usually lie between 150 and 450. For accurate rating of dehumidifying coils, good results are obtainable by first determining the average external surface temperature as previously described, and then using the difference between the external film temperature and the refrigerant for evaluating f_r in Equation 15.

$$f_r = \frac{Q}{\frac{a}{R}\,(t_s - t_r)} \tag{15}$$

The term $(t_s - t_r)$ is commonly written Δt. The usefulness of the foregoing equation is impaired by the fact that both f_r and Δt must be evaluated experimentally. More direct results can be obtained by ignoring f_r and determining the relation between Δt and total coil capacity:

$$\Delta t = t_s - t_r = mQ^n \tag{16}$$

where

m and n = constants determined by tests.

When water is used as a cooling medium in tubes, the rate of heat transfer is a function of water velocity, because this results in an increase in the number of contacts of the water molecules with the tube surface, per unit of time. Thus increased water velocity and reduced tube diameter cause increased heat transfer. Heat transfer is also greater at

[6]Calculation of Coil Surface Areas for Air Cooling and Dehumidification, by J. McElgin and D. C. Wiley (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940).

higher temperatures of the water. The basic formula for the film coefficient of heat transfer for flow of water is as follows:

$$f_w = 1.5\ (t - 100)\ \frac{V^{0.8}}{D^{0.2}} \tag{17}$$

where

f_w = internal film coefficient of heat transfer, Btu per hour per square foot of internal tube surface per degree Fahrenheit.
V = water velocity, feet per second.
D = internal diameter of tube, inches.
t = average water temperature, degrees Fahrenheit.

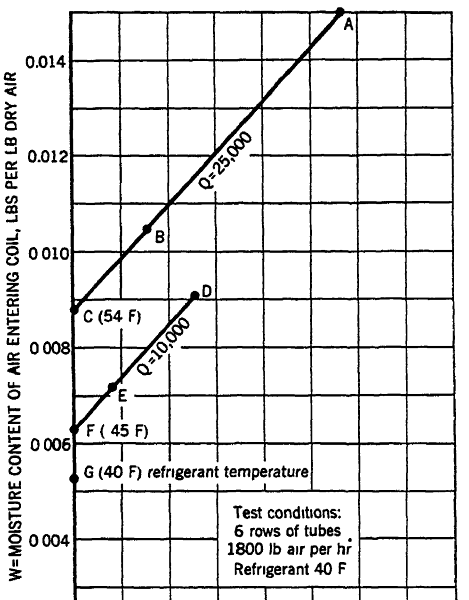

FIG. 14. DETERMINATION OF SURFACE TEMPERATURE

In the case of finned tubes, values of f_w may be lower than those obtained by use of Equation 17. Accurate results can be obtained by using Equation 15, if the logarithmic mean temperature difference between surface and water is used in place of Δt.

When saturated steam is condensed in the tubes of coils, the film coefficient f_r varies from 1000 to 2000, depending on freedom from air in the steam, and upon good drainage of the tubes. The coefficient is fairly constant for a particular coil, giving values of Δt that are directly proportional to Q.

GRAPHICAL ANALYSIS OF COIL PERFORMANCE

In testing coils, determination of surface temperatures is most important. A convenient way of determining surface temperatures is

illustrated in Fig. 14. Test points A and B are made without varying the wet-bulb temperature of entering air, the air velocity, the refrigerant temperature, and the total capacity of the coil. Only the dry-bulb and dew-point temperatures of the entering air are varied. A straight line is drawn between points A and B, and is extended to the ordinate of zero moisture removal, giving point C which represents the moisture content of saturated air that corresponds to the surface temperature. Points D and E are similarly plotted, the only difference being that another total coil capacity and entering air wet-bulb temperature are chosen.

The saturation temperatures of points C and F are then used in Equation 16, in conjunction with the test values of t_r and Q, so as to evaluate the constants m and n by solving two simultaneous equations. The

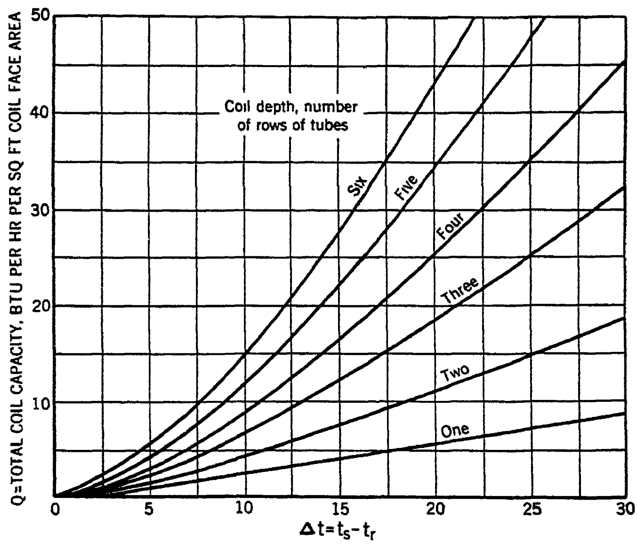

FIG. 15. TYPICAL CURVES SHOWING RELATION BETWEEN TOTAL CAPACITY AND TEMPERATURE DIFFERENCE FOR REFRIGERANTS

resulting equation is plotted as shown in Fig. 15, or can be plotted as a straight line on logarithmic paper.

Having determined the surface temperature, the test data can be used to evaluate coil efficiency, from the ratio $(t_1 - t_2) \div (t_1 - t_s)$. Then constants of Equation 11 can be evaluated and a group of curves constructed as in Fig. 12.

Use of Graphs for Predicting Performance

Coil performance under any dehumidifying condition can be predicted as shown in the following example, using Figs. 12, 13 and 15.

Example 2. Given: Total heat to be removed, 18,000 Btu per hour per square foot of coil face area; ratio of latent to total heat, 35 per cent; dry-bulb temperature of air entering coil, 83 F; dew-point temperature of air entering coil, 65 F. *Required:* Coil depth, air velocity and refrigerant temperature.

Solution. (1) Plot the entering air conditions at point C on Fig. 13. (2) Draw line C-E, parallel to the 35 per cent line N-O of the index chart, and obtain the required surface temperature, 55 F. (3) In Fig. 15, assume a coil depth of 4 rows, and obtain

Δt = 16 F. Subtracting 16 deg from 55 deg gives a required refrigerant temperature of 39 F. (4) In Fig. 12, assume an air velocity of 2000 lb per hour and obtain a coil efficiency of 0.8. (5) Solving for Q in Equation 12, a capacity of 17,400 Btu per hour is obtained, which is not the required capacity. It is necessary to try a higher air velocity until a balanced condition is found at an air velocity of 2080 pounds per hour. (6) By assuming a coil depth of 6 rows and repeating the same procedure, another solution can be obtained at a refrigerant temperature of 43.5 F and an air velocity of 1760.

The foregoing cut-and-try calculations can be eliminated by the use of the type of graph shown in Fig. 16, which may be constructed as outlined herewith:

1. The three axes of the nomogram on the left side of the chart are drawn in such a manner that the C axis represents the differences in total heat content between the air at wet-bulb temperature along B axis and air at wet-bulb temperature along A axis. Thus, the C axis represents the total heat (Btu per pound of air, sensible and latent) which could be removed from the air at some inlet wet-bulb temperature on B axis if

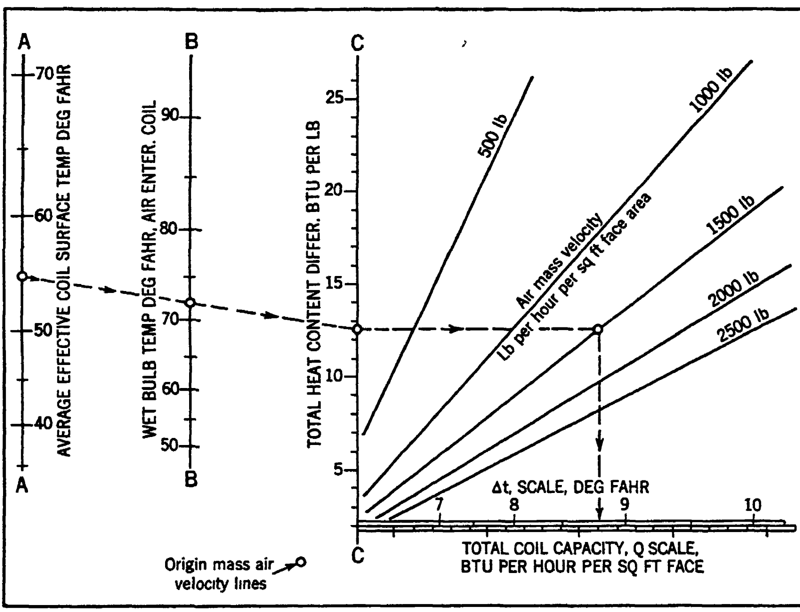

FIG. 16. TYPICAL COIL PERFORMANCE CHART

the coil heat transfer efficiency were 100 per cent and the wet-bulb temperature of the air could be reduced to some average (effective) external coil temperature on A axis. For example if a straight line is drawn through 72 F wet-bulb temperature of entering air on axis B and the 55 F average effective coil (external surface) temperature on axis B, then this straight line will intersect the C axis at 12.6, which figure represents the difference in total heat content of air between 72 and 55 F wet-bulb temperature.

2. Next scale Q is drawn to cover the range of the likely practical loading for the given coil in Btu per hour per square foot coil face area.

3. Lastly, the diagonal mass air velocity lines are drawn in at the intersection of various values on C axis and the corresponding values on the Q scale. The values on the Q scale corresponding to various values on C axis are obtained by multiplying the values on C axis by mass air velocity and coil efficiency. In this way the calculations required by Equation 12 are performed.

4. Parallel to the Q scale is drawn the Δt scale, so that the difference between average surface temperature and refrigerant temperature can be read directly, eliminating the use of Equation 16.

For coils using water as a cooling medium, the chart shown in Fig. 17 can be used for the purpose of eliminating calculations. Such a chart can embrace all sizes of coils of a particular design, but requires an index which gives the *coil design factor* for each size. The coil design factor is the number of square feet of internal tube surface of the entire coil. The curves shown in the lower right hand quarter of the chart perform the calculations of Equation 17, by using an average water temperature and the actual tube diameter.

Performance of Coils and Refrigeration Compressor

Practically all data published by various makers of direct expansion cooling coils are based upon maintaining a predetermined refrigerant

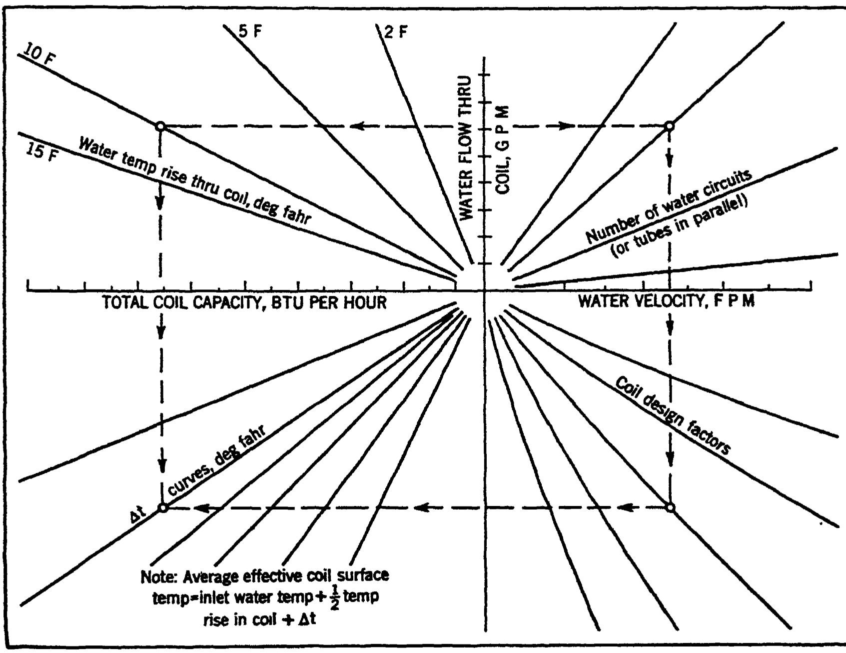

FIG. 17. WATER COIL PERFORMANCE CHART

temperature within the coils. While it is often possible to maintain a definite refrigerant temperature within a given cooling coil, for the greater part it is either impossible or impractical. This is due to the fact that the capacity of standard refrigeration compressors is usually fixed and in matching a given cooling coil with a standard compressor the capacity of the latter is often somewhat smaller or greater than that of the former. Consequently, very often the refrigerant temperature resulting within a cooling coil and correspondingly the capacity of the coil-compressor combination are not what they were originally calculated to be.

In order to determine the actual performance of a given coil-compressor combination under varying conditions of operation, a graphical solution of the balance point is highly desirable. A typical method of graphical

analysis of a coil-compressor combination performance is shown in Fig. 18, which is constructed in a manner described herewith:

1. On a piece of graph paper (with a uniform scale), the equipment capacity scale, total Btu per hour, is laid out along the vertical axis while the refrigerant suction temperature scale is laid out along the horizontal axis.

2. The performance curve of a given compressor with a definite condenser (combination usually called a *condensing unit*) is plotted as a function of suction temperature corresponding to the saturation suction pressure at the compressor suction service valve for a given inlet water temperature and quantity supplied to the condenser.

3. The performance curve of the given cooling coil is next plotted as a function of mean suction temperature within the coil, the mass air velocity over the coil and the wet-bulb temperature of air entering the coil.

4. The refrigerant pressure drop between the center of the cooling coil and the compressor suction service valve is computed and converted into the terms of temperature-

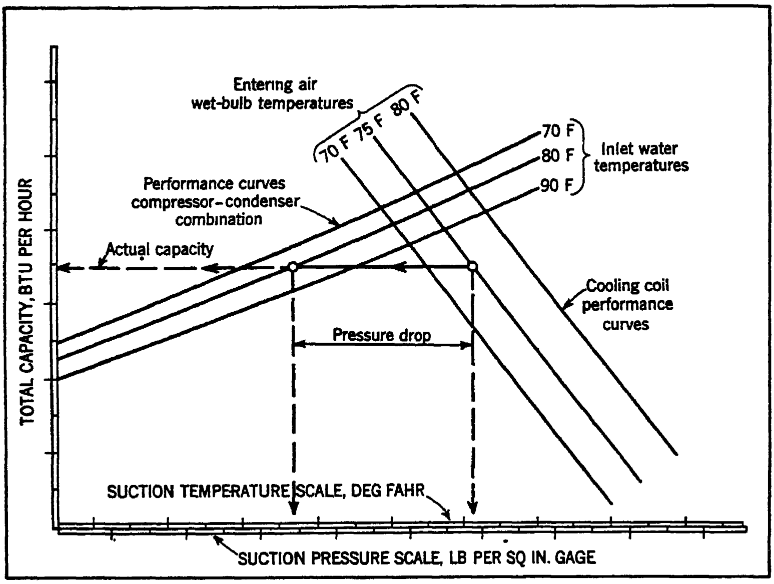

FIG. 18. GRAPHICAL ANALYSIS OF COIL-COMPRESSOR PERFORMANCE

difference. This temperature difference is then fitted in horizontally between the performance curves of the cooling coil and the compressor, as shown, and the total capacity of the coil-compressor combination is read along the horizontal line upon which the above mentioned *temperature-difference* segment falls.

COIL SELECTION

In the selection of a coil it is necessary to consider several factors:

1. The duty required—heating, cooling, dehumidifying.
2. Temperature of entering air—dry-bulb only if there is no dehumidification, dry- and wet-bulb if moisture is to be removed.
3. Available heating and cooling media.
4. Space and dimensional limitations.
5. Air quantity limitations.
6. Allowable resistances in air circuit and through tubes.
7. Peculiarities of individual designs of coils.

8. Individual installation requirements, such, for example, as type of automatic control to be used.

The duties required may be determined from information in Chapters 4, 5, 6 and 7. There may or may not be a choice of cooling and heating media, as well as temperatures available, depending upon whether the installation is new or is in combination with present sources of heating or cooling. Space limitations are dictated by the requirements of individual cases. The air quantity is influenced by a number of considerations. The air quantity through heating coils is often made the same as that necessary to handle the summer cooling load. The air handled may be fixed by the use of old ventilating ducts as an air distribution system for new air conditioning apparatus, or may be dictated by requirements of satisfactory air distribution or ventilation. The resistance through the air circuit influences the fan horsepower and speed. This resistance may be limited to allow the use of a given size of fan motor, or to keep the operating expense low, or it may be limited by the maximum fan peripheral velocity which requirements of quietness may permit. The friction through the water or brine circuit may be dictated by the head available from a given size of pump and pump motor. As the fan and pump motor inputs represent a refrigerating load on cooling installations, it is economical to keep them low.

Proper performance of a surface heating or cooling coil depends upon correct choice of the original equipment and upon certain other factors. The usual coil ratings are based on a uniform face velocity of air. If the air is brought in at odd angles or if the fan is located so as to block part of the air flow, the performance as given in the manufacturer's ratings cannot usually be obtained. To obtain this performance it is necessary also that the air quantity be adjusted on the job to that used in determining the coil selection, and must also be kept at this value. The most common causes for a reduction of air quantity are the fouling of the filters and collection of dirt in the coils. These difficulties can be avoided by proper design and proper servicing. There are a number of ways in which coils may be cleaned. A common method is to wash them off with water. They can sometimes be brushed and cleaned with a vacuum cleaner. In bad cases of neglect, especially on restaurant jobs where grease and dirt have accumulated, it is sometimes necessary to remove the coils and wash off the accumulation with steam, compressed air and water, or hot water. The most satisfactory solution, however, is to keep the filters serviced, and thus make the cleaning of the coils unnecessary.

The proper selection of coils requires an understanding of the necessities of each case and should be based on an economic analysis of the plant design as a whole. No general rule can, therefore, be laid down for the selection of heating or cooling coils. It is possible, however, to point out the limits of usual practice and to indicate the influence of the variables involved in the coil selection.

Heating Coils

Steam and hot water heating coils are usually rated within these limits:

Air Face Velocity—200 to 1200 fpm, sometimes up to 1500 fpm.
Steam Pressure—2 to 200 lb, sometimes up to 350 lb per square inch.
Hot Water Temperature—150 to 225 F.
Water Velocity—2 to 6 fps.

Individual cases may deviate widely, but the tabulation given herewith will serve as a guide to usual heating practice:

Air Face Velocity—500 to 800 fpm face, 500 being a common figure.

Delivered Air Temperature—varies from about 72 F for ventilation only to about 150 F for complete heating.

Steam Pressure—2 to 10 lb, 5 lb being common.

Hot Water Temperature—150 to 225 F.

Water Velocity—2 to 6 fps.

Water Quantity—Based on about 20 F temperature drop through a hot-water coil.

Air Resistance—The total resistance through heating coils is usually limited to from ⅜ to ⅝ in. of water gage for public buildings, to about 1 in. for factories.

The selection of heating coils is relatively simple as it involves dry-bulb temperatures and sensible heat only, without the complication of simultaneous latent heat loads, as in cooling coils. For a given duty, entering air temperature, and steam pressure, it is possible to select several arrangements of the same design of coil depending upon the relative importance of space, cross-sectional area, and air resistance.

Cooling Coils

The usual range of ratings for cooling and dehumidifying coils are enumerated herewith:

Entering Air Dry-Bulb—60 to 100 F.

Entering Air Wet-Bulb—50 to 80 F.

Air Face Velocities—300 to 800 fpm, (sometimes as low as 200 and as high as 1200).

Volatile Refrigerant Temperatures—25 to 55 F, at coil suction outlet.

Water Temperatures—40 to 65 F.

Water Quantities—2 to 6 gpm per ton, or equivalent to a water temperature range of from 4 to 12 F.

Water Velocity—2 to 6 fps.

The ratio of total to sensible heat removed varies in practice from 1.00 to about 1.65, i.e., sensible heat is from 60 to 100 per cent of total, depending on the application. (See Chapter 21). Required ratios may demand wide variations in air velocities, refrigerant temperatures, and coil depth, so that general rules as to these values may be misleading. On usual comfort installations air face velocities between 400 and 600 fpm are frequent, 500 being a common value. Refrigerant temperatures will ordinarily vary between 40 and 50 F where cooling is accompanied with dehumidification. Water velocities will range from 2 to about 6 fps.

When no dehumidification is desired, for which condition the dew-point of the entering air will be equal to or lower than the cooling coil temperature, the coil selection is made on the basis of dry-bulb temperatures and sensible heat transfers only, the same as with heating coils. It is possible also to choose various arrangements of face area, depth, air velocity, etc., for the same duty.

Dehumidifying Coils

The selection of coils for combined cooling and dehumidifying duty is more involved than for heating or sensible cooling and requires consideration of both dry- and wet-bulb air temperatures. It is further complicated by the fact that the proportional amount of dehumidification

TABLE 1. VARIOUS COOLING COIL ARRANGEMENTS

SELECTION	1	2	3	4
Total cooling capacity, tons	100	100	100	100
Sensible cooling capacity, tons	69	69	69	69
Latent cooling capacity, tons	31	31	31	31
Ratio total to sensible heat	1.45	1.45	1.45	1.45
Air quantity, cfm	47,800	41,700	37,100	46,800
Cfm per total ton	478	417	371	468
Face velocity, fpm	325	423	500	600
Resistance, in. water	0.11	0.27	0.51	0.37
Coil face area, sq ft	147	99.0	74.2	78.1
Coil rows deep	4	6	8	4
Coil evaporator temp. deg F	45	45	45	38

required is also highly variable. The methods outlined previously under Heat Transfer and Resistance may be used to determine whether it is possible for a coil to perform the duty required. If entering and leaving air conditions are arbitrarily specified, the corresponding duty sometimes cannot be obtained at all without the use of reheat. As with heating and sensible cooling coils, there are combinations of face areas, depth, air velocity and refrigerant temperatures which will give the required performance. This is illustrated in Table 1.

It is possible as shown in Table 1 to perform approximately the same duty at a given refrigerant temperature with small face area and large thickness or vice versa. The large face area coil will give low air velocity and resistance but high air quantities per ton. The coil of small face area and great depth will require small air quantities per ton of refrigeration, high resistance and high air velocities. As shown also in Table 1 the same sensible, latent and total cooling capacity may be obtained with various refrigerant temperatures by proper choice of coil. This makes it possible to keep the evaporating temperature high enough to carry the load with a chosen size of condensing unit. High evaporating temperatures with correspondingly small compressor operating expense can be attained but at the expense of coil surface, air quantity or both. The choice will be determined by the necessities of individual installations.

For a given quantity and condition of entering air the evaporating temperature of a volatile refrigerant coil will be determined by a balance between the condensing unit and the coil. The total, sensible and latent cooling capacity can then be determined from the coil rating information.

TABLE 2. CAPACITY BALANCES FOR MAXIMUM AND MINIMUM LOAD CONDITIONS

CONDITIONS	CAPACITY IN TONS			RATIO $\frac{\text{TOTAL}}{\text{SENSIBLE}}$
	Total	Sensible	Latent	
Required at peak load conditions	10.90	7.90	3.00	1.38
Required at minimum load conditions	6.62	3.36	3.26	1.98
Peak load equipment balance	10.90	7.90	3.00	1.38
Same equipment balanced at minimum load conditions	9.85	6.58	3.26	1.50
Same equipment balanced at maximum load conditions with 40 per cent by-pass	8.38	5.05	3.33	1.66
Same equipment balanced at minimum load conditions with 38,800 Btu per hour reheat	6.62	3.36	3.26	1.98

If the condensing unit and cooling coil have been properly balanced for the required load and, due to miscalculated duct resistance or improper choice of fan speed, the air quantity is reduced, the total cooling capacity will also be reduced. The decrease is generally in the sensible capacity. This is the effect also when the air by-pass or volume control is used.

It is necessary that not only the total capacity but also the sensible and latent cooling requirements both be met. The installation of an excess of coil will result in an increase in total capacity, but not a proportional gain in latent heat capacity. On installations controlled from dry-bulb temperature the operating time will be shortened because of the added sensible cooling capacity. The result will be less moisture pick-up than calculated, and higher relative humidity. If an oversize condensing unit is installed the opposite situation will take place. The relative humidity will be lower than estimated. This is not generally a disadvantage except that it results in a greater load from outside air than calculated, as well as in increased power consumption. If oversize equipment is furnished, a balance should be made to assure that the ratio of total to sensible capacity is the same as in the estimated load.

Sometimes arbitrary air quantities are specified for ventilation or other reasons independent of the selection of the cooling coil. As shown in Table 1, the coil selection can be altered to take care of various air quantities for the same duty.

Where coil and condensing unit are selected for the peak load condition, and the sensible load partially disappears due to fall of outside temperature or other cause, the condensing unit and coil rebalance. This may result in more sensible capacity than required at the light load condition and less latent in proportion, with an increased relative humidity in the conditioned space. Such a condition, for a typical installation, is shown in Table 2. If approximately 40 per cent of the total air is by-passed, the condition will be improved as indicated. The situation could be entirely avoided by using reheat. With sufficient reheat, it is possible to handle any ratio of sensible and latent loads and maintain the design temperature and humidity.

Care should be taken to avoid freezing at light loads. In general, freezing occurs when the coil surface temperature falls to 32 F. With usual coils for comfort installations, this will not occur unless the evaporating temperature at the coil outlet is about 20 to 25 F. The exact value depends on the design of coil and the amount of loading. Although it is not customary to choose coil and condensing units to balance at low temperatures at peak loads, there is danger of this occurring when the load decreases. This is further aggravated if a by-pass is used so that less air is passed through the coil at light loads. It may be even worse if the control is arranged for decrease of inside temperature with fall of that outside. Freezing can be avoided by making the full load balance a high evaporating temperature and checking the balance at the minimum load condition.

Care should be exercised in the design of humidity control to minimize the cycling of the refrigerating compressor because of re-evaporation of moisture from the fins. It is sometimes necessary to by-pass air around a coil when the compressor is not operating.

Chapter 27

SPRAY EQUIPMENT

Air Washers, Apparatus for Direct Humidification, Spray Generation and Distribution, Air Dehumidification with Washers, Water Main Temperatures, Atmospheric Water Cooling Equipment, Design Wet-bulb Temperatures for Water Cooling, Cooling Ponds, Winter Freezing

AIR humidification is effected by the vaporization of water and always requires heat from some source. This heat may be added to the water prior to the time vaporization occurs or it may be secured by a transformation of sensible heat of the air being humidified to latent heat as the vapor is added to the air. The thermodynamics of the process are discussed in Chapter 1. The removal of moisture from air may or may not involve the removal of heat from the air-vapor mixture. With spray equipment dehumidification of air always necessitates the removal of heat.

AIR WASHERS

Air washers may be used as either humidifiers or dehumidifiers depending upon the method of operation and the temperature of the spray water. The functions of an air washer are to regulate the moisture and heat content of air passing through it and to remove dust and dirt from the air. Air washers are not as effective as air filters in the removal of dust and dirt.

The construction of commercial air washers is indicated in Figs. 1 and 2. Any air washer consists essentially of a chamber through which the air passes in intimate contact with water. The lower portion of the washer chamber serves as a sump for the spray water.

Contact between the air and the washer water is secured: (1) by breaking the water into a very fine mist, (2) by passing the air over surfaces which are continuously wetted by water, or (3) by a combination of water sprays and wetted plates. Scrubber-plate types of washers are used largely to wash heavy reclaimable products from the air, and are generally composed of one to three eliminator-type baffle scrubber plates across the air stream. Water is supplied at the tops of the scrubber plates by flooding nozzles placed across the top of the washer. Spray washers have one or more banks of water atomizing nozzles placed in the air stream above the level of the water in the sump. The direction of the water sprays may be against the air stream, with the air stream, or with

one bank spraying with the air stream and one against it. The number of nozzles required depends upon their design, the quantity of air handled, and the arrangement of the nozzles.

Scrubbers generally consist of eliminator-type baffle plates placed in the air stream to cause several reversals of the direction of air flow. The scrubber plates are more effective as air cleaners than as humidifiers. All washer chambers should have inlet diffuser plates to aid in producing more uniform air flow through the washer spray chamber. These inlet vanes also aid in preventing spray water from being thrown into the air duct ahead of the washer. However, if the water spray is against the air flow, the ordinary perforated diffuser plate is not sufficient, and specially designed eliminator baffles must be used to prevent spray from passing

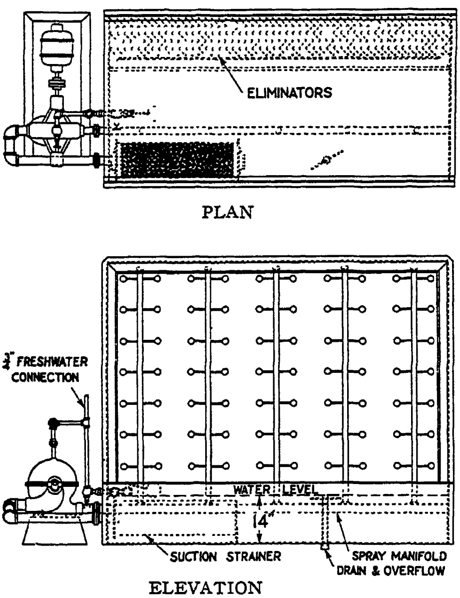

FIG. 1. TYPICAL SINGLE BANK AIR WASHER

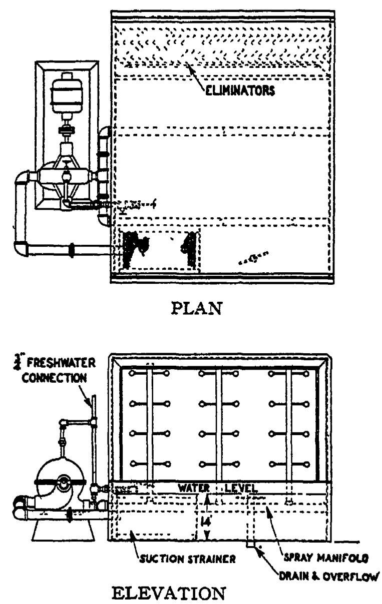

FIG. 2. TYPICAL TWO BANK AIR WASHER

into the air inlet duct. At the outlet end of the washer suitable flooded eliminator plates, which will cause from 4 to 6 reversals of the direction of air flow, should be installed for the purpose of removing drops of unvaporized water from the leaving air. When the air carries certain substances mixed with it, the spray water may become acidulated and special consideration must be given to the materials used, to reduce the corrosive action.

Essential items in air washer operation are: uniform distribution of the air across the chamber section above the level of the water in the sump; moderate velocities of air flow, 300 to 600 fpm in the spray chamber; an adequate amount of spray water broken up into a fine mist throughout the air stream; sufficient length of air travel through the water spray and over thoroughly wetted surfaces; and the elimination of free moisture from the air as it leaves the unit.

Washers are sometimes arranged in two or more stages to cool through long ranges or to increase the overall efficiency of heat transfer between the air and the heating or cooling medium. A multi-stage washer is equivalent to a number of washers in a series arrangement. Each stage is in effect a separate washer.

Usually the catalog capacity of a washer is expressed in cubic feet of air per minute and is based upon an air velocity of 500 fpm through the gross inlet area of the unit. At this rating spray type washers handle about 2½ gpm of water per bank per square foot of area, that is, about 5 gpm per bank per 1000 cfm. These proportions of air, water, area, and velocity may be departed from to meet the needs of some particular job, but certain limiting relationships should be observed.

For a single stage air washer, a 15 F drop in dry-bulb temperature of the air passing through the washer is about the maximum that should be anticipated. For greater decrease in dry-bulb temperature, multi-stage washers should be utilized. A rise of 6 F should be the calculated maximum for the spray water.

The width and height of a washer may be dictated by space limitations outside the washer, such as headroom, or by the inside space requirements, such as face area needed by a bank of cooling coils. The length of a washer is determined by the number of spray banks, or scrubber plates, and if cooling coils are installed in the unit, by the number of banks of coils. Roughly, a spray space of about 2 ft 6 in. in length is required for each bank of sprays; *leaving* eliminators require about 1 ft 6 in., and *entering* eliminators about 1 ft.

The resistance to air flow through an air washer varies with the type of eliminators, number of banks of sprays, direction of spray, air velocity, type of scrubber plates, and size and type of cooling coils if located in the washer. Manufacturers should be consulted to obtain the resistance for a particular installation.

HUMIDIFICATION WITH AIR WASHER

Air humidification can be accomplished in three ways with an air washer. These are: (1) use of recirculated spray water without prior treatment of the air, (2) preheating the air and washing it with recirculated spray water, and (3) using heated spray water. In any problem of air washing the air should not enter the washer with a dry-bulb temperature less than 35 F so that there will be no danger of freezing the spray water.

Method 1. Except for the small amount of energy added from outside by the recirculating pump in the form of shaft work, and for the small amount of heat leak from outside into the apparatus, including the pump and its connecting piping, the process would be strictly adiabatic. Evaporation from the liquid spray would therefore be expected to bring the air immediately in contact with it to saturation adiabatically; and, since the liquid is recirculated, its temperature would be expected to adjust to the thermodynamic wet-bulb temperature of the entering air.

It does not follow from the above reasoning that the whole air stream is brought to complete saturation, but merely that its state point should move along a line of constant thermodynamic wet-bulb temperature as

explained in Chapter 1. The extent to which the final temperature approaches the thermodynamic wet-bulb temperature of the entering air, or the extent to which complete saturation is approached is conveniently expressed by a ratio known as *humidifying efficiency* or saturating efficiency and defined as follows:

$$e_h = \frac{t_1 - t_2}{t_1 - t'} \tag{1}$$

where

e_h = humidifying efficiency, per cent.
t_1 = dry-bulb temperature of the entering air, degrees Fahrenheit.
t_2 = dry-bulb temperature of the leaving air, degrees Fahrenheit.
t' = thermodynamic wet-bulb temperature of the entering air, degrees Fahrenheit.

The humidifying or saturating efficiency of a washer is dependent upon the number of spray banks and nozzles, the effectiveness of the nozzles in breaking an adequate quantity of water into a fine spray, the velocity of air flow through the water sprays, and the time of the contact of the air with the spray water. Other conditions being the same, low velocities of air flow are more conducive to higher humidifying efficiencies. The following may be taken as representative humidifying or saturating efficiencies of air washers for the conditions stated:

1 bank—downstream	60–70 per cent
1 bank—upstream	65–75 per cent
2 banks—downstream	85–90 per cent
2 banks—1 upstream and 1 downstream	90–95 per cent
2 banks—upstream	90–95 per cent

The air leaving the washer may require reheating to produce the required dry-bulb temperature and relative humidity.

Method 2. The preheating of the air increases both the dry and wet-bulb temperatures, lowers the relative humidity, but does not alter the humidity ratio (pound water vapor per pound dry air). At a higher wet-bulb temperature but the same humidity ratio, more water can be absorbed per pound of dry air in passing through the washer, assuming that the humidifying efficiency of the washer is not adversely affected by operation at the higher wet-bulb temperature. The analysis of the process occurring in the washer itself is the same as that explained under Method 1. The final desired conditions are secured by adjusting the amount of preheating to give the required wet-bulb temperature at entrance to the washer and then reheating when necessary after passage through the washer.

Method 3. Even if heat is added to the spray water, the mixing occurring in the washer itself may still be regarded as adiabatic. The state point of the mixture should move in a direction determined by the specific enthalpy of the heated spray as explained in Chapter 1. By sufficiently elevating the spray water temperature it should be possible to completely saturate the air and even raise its temperature above the dry-bulb temperature of the entering air.

APPARATUS FOR DIRECT HUMIDIFICATION

Humidifiers may be divided into the following general types, according to the method of operation: (1) indirect, such as the air washer, which

introduces moistened air; and (2) direct, which sprays moisture into the room or introduces moisture by means of steam jets.

As in the cases of humidification by use of an air washer, the heat necessary for the vaporization of the moisture added to the air by direct humidification is secured either from heat stored in the spray water or by a transformation of sensible to latent heat in the air humidified. In the latter case the enthalpy of the air remains constant but the dry-bulb temperature of the air is reduced.

Direct humidification is usually preferable where high relative humidities must be maintained, but where there is little cooling or ventilation required. In comfort air conditioning, where both humidification and ventilation are required, the indirect humidifier is preferable. In industrial applications, where the cooling or ventilation load is large and where very high relative humidities must be maintained, a combined system employing both direct and indirect humidifiers is sometimes used.

Spray Generation

Spray generation is obtained by (1) atomization, (2) impact, (3) hydraulic separation, and (4) mechanical separation.

Atomization involves the use of a compressed air jet to reduce the water particles to a fine spray. With the *impact* method, a jet of water under pressure impinges directly on the end of a small round wire. Where *hydraulic separation* is employed, a jet of water enters a cylindrical chamber and escapes through an axial port with a rapid rotation which causes it immediately to separate in a fine cone-shaped spray. In the *mechanical separation* process, water is thrown by centrifugal force from the surface of a rapidly revolving disc and separates into particles sufficiently small to be utilized in certain types of mechanical humidifiers.

Spray Distribution

Spray distribution is obtained by (1) air jet, (2) induction, and (3) fan propulsion.

The air jet which generates the spray in atomizers also carries the spray through a space sufficient for its distribution and evaporation, and this method of distribution is termed *air jet*. Where distribution is obtained by *induction*, the aspirating effect of an impact or centrifugal spray jet is utilized to induce a current of air to flow through a duct or casing, and this air current distributes the spray. *Fan propulsion* obviously consists of the utilization of fans to entrain and distribute the spray.

Industrial type direct humidifiers are commonly classified as (1) atomizing, (2) high-duty, (3) spray and (4) self-contained or centrifugal.

Atomizing Humidifiers

There are several types of atomizing humidifiers, all of which rely upon compressed air as the atomizing and distributing agency, similar to the familiar method used in ordinary nasal atomizers. Compressed air (ordinarily about 30 lb per square inch) is supplied from a centrally-located air compressor through pipe lines to the atomizing units. The air lines are usually horizontal and parallel to water lines which supply water by gravity from a float tank. The water in the tank is maintained

at a constant level slightly lower than the outlets of the atomizers themselves and is drawn constantly to the atomizer by aspiration when compressed air is supplied. This aspiration ceases and the flow of water stops when the air supply is cut off. The water should not be supplied under pressure to atomizers because of the possibility of leakage, drip, or coarse spray. These cannot occur when water is supplied by aspiration.

High-Duty Humidifiers

Water is supplied to high-duty humidifiers under high pressure (usually about 150 lb per square inch) through pipe lines from a centrally-located pumping unit. The spray-generating nozzle which is of the impact type is located in a cylindrical casing. A drainage pan provides for the collection and return of unevaporated water which flows through a return pipe to a filter tank, from which it is recirculated. A powerful air current is forced through the humidifier by means of a fan mounted above the unit.

The air enters from above, is drawn through the head, charged with moisture, and cooled. It then escapes from the opening below at a high velocity in a complete and nearly horizontal circle. The spray is evaporated and the resulting vapor diffused. This distribution of fine spray over the maximum possible area promotes complete and rapid vaporization even at high humidities.

Spray Humidifiers

This type of humidifier consists of an impact spray nozzle in a cylindrical casing with a drainage pan below it. The aspirating effect of the spray nozzle induces a moderate air current through the casing which distributes the entrained spray. The general method of circulating and returning the water is similar to that employed for high-duty humidifiers. A suitable pump and centrally-located filter tank are required.

Self-Contained Humidifiers

The self-contained or centrifugal humidifier has the ability to generate and distribute spray without the use of air compressors, pumps, or other auxiliaries. These may be used either singly or in groups. In large installations, where suitable connections are provided to permit the cleaning and servicing of individual units without affecting the room as a whole, group control of the water and power may be employed.

AIR DEHUMIDIFICATION WITH WASHERS

Moisture removal from an air-vapor mixture can be accomplished by use of an air washer so long as the temperature of the spray medium is lower than the dew-point of the air passing through the unit. The final dry-bulb temperature and the relative humidity of the air leaving a dehumidifier washer are dependent upon: the air velocity, the length of air travel through the sprays, the dry- and wet-bulb temperatures of the entering air, the spray temperature, the number of spray banks and nozzles, the quantity of spray medium handled, and the effectiveness of the nozzles in breaking the spray into a fine mist.

Both sensible and latent heat are removed in the process of dehumidification by cooling. Abstraction of sensible heat occurs during the entire time that the air is in contact with the spray medium. Latent heat

removal takes place as condensation occurs. Therefore, the lower the spray temperature the greater the amount of moisture removal per pound of dry air, all other conditions remaining the same. Washers with two or more banks of sprays are usually selected for comfort air conditioning installations. Such washers will cool the air to within 1 or 2 F of the leaving spray water temperature.

Where a limited supply of cold water is available multiple stage washers may be used to an advantage. The cool water is pumped through the multiple spray systems in series. By this arrangement the entering air is cooled first by the warmer water and finally by the cooler water which gives the maximum amount of cooling with the minimum amount of water. The approximate temperatures of water from wells at depths of 30 to 60 ft are given in Fig 3[1]. Frequently the temperature of the city water main supply is low enough during the summer to permit an appreciable cooling effect. Table 1 lists the maximum city water main temperatures for various localities in the United States and Canada.

Air washers using refrigerated spray generally have their own recirculating pumps. These pumps deliver to the sprays a mixture of water from the washer sump, which has not been re-cooled, and refrigerated water. The quantities of each are controlled by a three-way or mixing valve actuated by a dew-point thermostat located in the washer air outlet or by humidity controllers located in the conditioned space.

ATMOSPHERIC WATER COOLING EQUIPMENT

In the operation of a refrigerating plant or a condensing turbine, one of the main problems is the removal and dissipation of heat from the compressed refrigerant or the discharged steam. This is accomplished ordinarily by first transferring the heat of the gas to water in a heat exchanger, from which water it may then be dissipated in a number of ways. If the plant is situated on the banks of a river or lake, an intake may be taken up-stream or at a considerable distance from the discharge, to prevent mixing of the heated discharged water with the inlet water. If the source of cooling water is a city supply or a well, the discharge water may be run into the nearest sewer or open waterway. Lacking an unlimited water supply, or in cases where city water is too expensive or where the water available contains dissolved salts which would form scale on the heat-exchanging apparatus, it is necessary to recirculate the water, and to cool it after each passage through the heat-exchanger by exposure to air in an atmospheric water cooling apparatus.

Air has a capacity for absorbing heat from water when the wet-bulb temperature of the air is lower than the temperature of the water with which it is in contact. The rapidity with which this transfer of heat occurs depends upon (1) the area of water in contact with the air, (2) the relative velocity of the air and water, and (3) the difference between the wet-bulb temperature of the air and the temperature of the water. Because the changes in rate do not occur in direct proportion to changes in the governing factors, data on the performance of atmospheric water cooling equipment are largely empirical.

[1]Temperature of Water Available for Industrial Use in the United States, by W. D. Collins (*U. S Geological Survey*, Water Supply Paper No. 520 F).

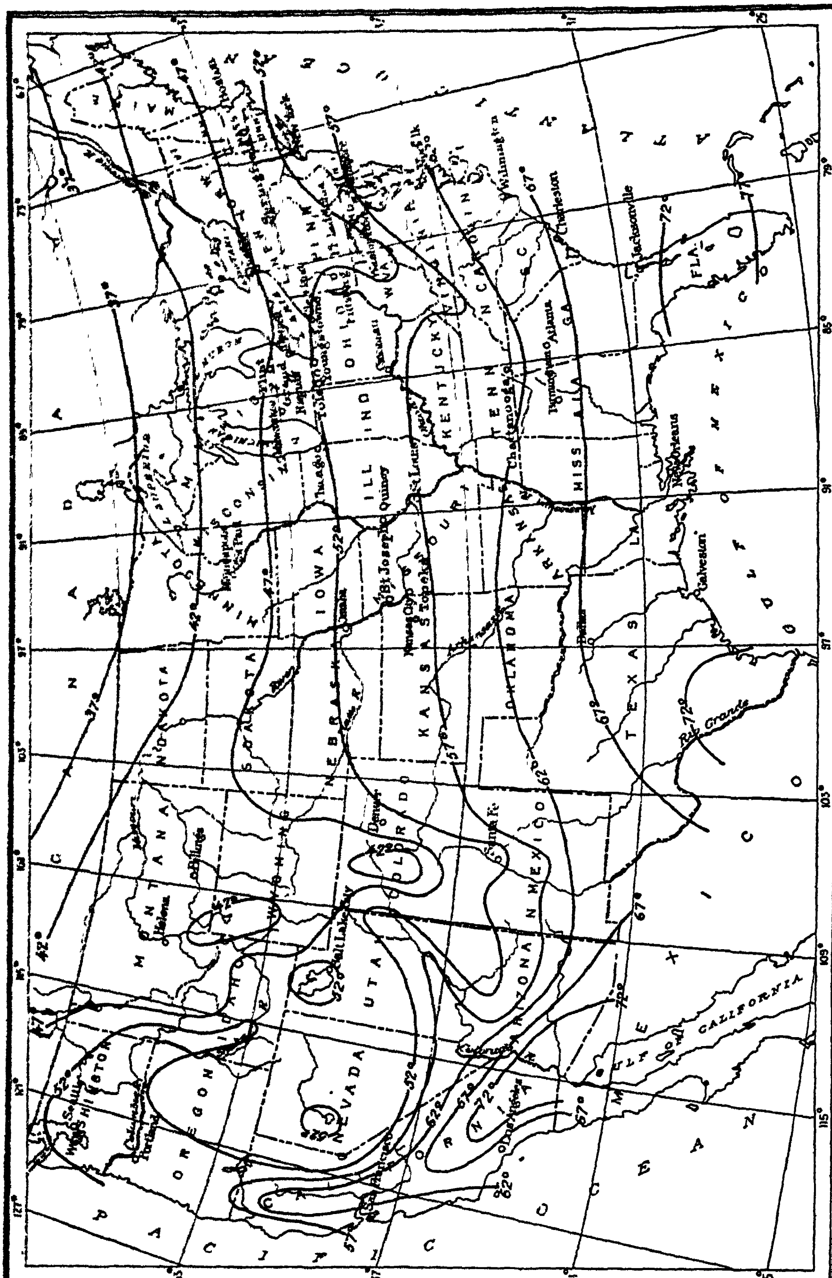

FIG. 3. APPROXIMATE WELL WATER TEMPERATURES AT DEPTHS OF 30 TO 60 FT

TABLE 1. AVERAGE MAXIMUM WATER MAIN TEMPERATURES[a]

State	City	Temp. F	State	City	Temp. F
Ala.	Birmingham	84	Mass.	Boston	80
	Mobile	73		Cambridge	70
Ariz.	Phoenix	81		Fall River	76
	Tucson	80		Lowell	50
Calif.	Anaheim	60		Lynn	68
	Berkeley	69		New Bedford	70
	Fresno	72		Salem	68
	Fullerton	75		Worcester	76
	Glendale	68	Mich.	Detroit	77
	Los Angeles	75		Flint	70
	Oakland	69		Grand Rapids	84
	Ontario	70		Highland Park	77
	Pasadena	82		Jackson	56
	Pomona	75		Kalamazoo	53
	Riverside	78		Lansing	64
	Sacramento	72		Saginaw	82
	San Bernardino	65	Minn.	Duluth	55
	San Diego	82		Minneapolis	80
	San Francisco	62		St. Paul	77
	Whittier	75	Mo.	Jefferson City	82
Colo.	Denver	75		Kansas City	84
Conn.	Bridgeport	66		St. Joseph	84
	Hartford	73		St. Louis	85
	New Haven	76		Springfield	70
	Waterbury	72	Nebr.	Lincoln	87
D. C.	Washington	84		Omaha	87
Del.	Wilmington	83	Nev.	Reno	61
Fla.	Jacksonville	80	N. H.	Manchester	76
	Miami	80	N. J.	Jersey City	63
	Tampa	77		Newark	74
Ga.	Atlanta	87		Paterson	78
	Macon	80		Trenton	79
Ill.	Chicago	76	N. Y.	Albany	68
	Cicero	76		Buffalo	75
	Evanston	73		Jamaica	56
	Peoria	67		Mt. Vernon	74
	Rockford	59		New Rochelle	75
	Springfield	82		New York	72
Ind.	Evansville	86		Rochester	70
	Gary	75		Schenectady	60
	Indianapolis	80		Syracuse	74
	South Bend	61		Utica	69
	Terre Haute	82		Yonkers	70
Iowa	Cedar Rapids	78	N. C.	Asheville	74
	Des Moines	77		Charlotte	85
	Sioux City	62		Winston-Salem	82
Kans.	Concordia	57	N. M.	Albuquerque	65
	Kansas City	86	Ohio	Akron	76
	Topeka	88		Canton	50
	Wichita	72		Cincinnati	84
Ky.	Louisville	85		Cleveland	74
La.	Baton Rouge	85		Columbus	82
	New Orleans	85		Dayton	60
Me.	Augusta	60		Lakewood	82
Md.	Baltimore	75		Springfield	72
				Toledo	83

[a]These averages taken from various city water main locations, with some actual values slightly higher and some lower than values shown.

Table 1. Average Maximum Water Main Temperature[a] (Concluded)

State	City	Temp F	State	City	Temp. F
Okla.	Oklahoma City	82	Utah	Logan	44
	Tulsa	85		Salt Lake City	60
Ore.	Eugene	69	Va.	Fredericksburg	75
	Portland	64		Lynchburg	73
Pa.	Altoona	74		Norfolk	80
	Erie	75	Wash.	Olympia	58
	Johnstown	74		Seattle	62
	McKeesport	82		Spokane	51
	Philadelphia	83		Tacoma	57
	Pittsburgh	81	W. Va.	Charleston	85
R. I.	Providence	68		Huntington	78
S. C.	Charleston	80		Wheeling	78
	Greenville	81	Wis.	LaCrosse	54
	Spartanburg	78		Madison	58
S. Dak.	Rapid City	55		Milwaukee	70
Tenn.	Chattanooga	84		Racine	68
	Knoxville	89			
	Memphis	70			
	Nashville	90			
Texas	Amarillo	65	Province		
	Austin	90			
	Beaumont	86			
	Dallas	86	Alta.	Calgary	64
	Fort Worth	84	B. C.	Vancouver	60
	Galveston	90	Ont.	London	50
	Houston	84		Toronto	63
	Port Arthur	83	P. E. I.	Charlottetown	48
	San Antonio	76	Que.	Montreal	78
	Wichita Falls	85		Quebec	68

[a]These averages taken from various city water main locations, with some actual values slightly higher and some lower than values shown.

As the heat content of the air increases, its wet-bulb temperature rises. (See Chapter 1.) Because it is impractical to leave the air in contact with water for a long enough time to permit the wet-bulb temperature of the air and the temperature of the water to reach equilibrium, atmospheric water cooling equipment aims to circulate only enough air to cool the water to the desired temperature with the least possible expenditure of power.

In an air washer, humidifier or dehumidifier, the air is first conditioned by water to change its moisture and temperature, and it is then sent to the place where it is to be used. In water cooling equipment the temperature of the water is reduced by air, and the cooled water is carried to its point of usage. In the air washer, an excess of water is used to condition a fixed quantity of air, while in water cooling equipment, an excess of air is used to cool a fixed quantity of water.

Both types of equipment have a common basis of design, however, in that the size of the equipment is determined by the quantity of air that must be handled. With the air washer, the size of the equipment is fixed by the quantity of air to be conditioned, and the amount of conditioning is controlled by the quantity and temperature of the water supplied and its method of application. With water cooling apparatus, its size and the

quantity of air required bear no direct relation to the quantity of water being cooled, but vary through a wide range for different services and conditions.

Sizes of Equipment

Assuming a definite quantity of water to be cooled, the size and design of atmospheric cooling equipment are affected by the following factors:

1. Temperature range through which the water must be cooled.
2. Number of degrees above the wet-bulb temperature of the entering air to which the water temperature must be reduced.
3. Temperature of the atmospheric wet-bulb at which the required cooling must be performed.
4. Time of contact of the air with the water. (This involves height or length of the apparatus and velocity of air.)
5. Surface of water exposed to each unit quantity of air.
6. Relative velocity of air and water.

Items 1, 2, and 3 are established by the type of service and geographical location, while items 4, 5, and 6 depend upon the design of the equipment.

The establishment of a proper cooling range depends upon:

1. Type of service (refrigerating, internal combustion engine and steam condensing).
2. Wet-bulb temperature at which the equipment must operate satisfactorily.
3. Type of condenser or heat-exchanger used.

Because the design of an entire plant is usually affected by the quantity and temperature of the cooling water supply, plants should be designed for cooling water conditions which can be most efficiently attained. The first consideration is usually the limiting temperature of the plant. For example, if an ammonia compressor refrigerating plant is to be designed for 185 lb head pressure as a normal maximum, the limiting temperature of the ammonia in the condenser is 96 F. Should the ammonia temperature go above this figure the head pressure will exceed 185 lb and power consumption increases. To obtain this head pressure, the temperature of the circulating water leaving the condenser must always be less than 96 F by an amount depending upon the size and design of the condenser, the

Table 2. Condenser Design Data

Gas	Maximum Pressure Desired in Condenser	Gas Temperature in Condenser Deg F	Leaving Hot Water Temperature Deg F	
			Best Condenser Design	Average Condenser Design
Steam	28 in. vacuum	101.2	97	93
Steam	27 in. vacuum	115.1	110	105
Steam	26 in. vacuum	125.9	120	114
Ammonia	185 lb gage head pressure	96.0	92	88
Carbon dioxide	1030 lb gage head pressure	86.0	83	81
Methyl chloride	102 lb gage head pressure	100.0	96	92
Dichlorodi-fluoromethane	117 lb gage head pressure	100.0	96	93

quantity of water being circulated, and the refrigerating tonnage being produced. A condenser having a large surface per ton of refrigeration may be designed to operate satisfactorily with the leaving hot water temperature within 3 or 4 F of the ammonia temperature corresponding to the head pressure, while a small condenser might require a 10 F difference.

Table 2 lists several gases with data as to the temperatures and pressures for which commercial condensers are designed. Internal combustion engines have limiting hot water temperatures of 125 F to 140 F. The cooling of such fluids as milk or wort has variable requirements and is usually done in counter-flow heat-exchangers in which the leaving circulating water is at a much higher temperature than is the leaving fluid.

The temperature range, once the hot water temperature is approximately known, depends upon:

1. Maximum wet-bulb temperature at which the full quantity of heat must be dissipated.

2. Efficiency of the atmospheric cooling equipment considered.

Design Wet-Bulb Temperatures

The maximum wet-bulb temperature at which the full quantity of water must be cooled through the entire range is never, in commercial design, the *maximum* wet-bulb temperature ever known to exist at the location nor the *average* wet-bulb temperature over any period. The former basis would require atmospheric cooling equipment several times greater than normal size, and the latter would result during a large part of the time, in higher condenser water temperatures than those for which the plant was designed. For instance, the maximum wet-bulb temperature recorded in New York City is 88 F, and the July noon average for 64 years is close to 68 F. Yet in the years 1925 to 1934, inclusive, there were but 8 hours per year when the wet-bulb temperature reached 80 F or more, and there were 975 hours in the average summer (June to September inclusive) when the wet-bulb temperature was 68 F or above. As these 975 hours represent a third of the summer period, cooling equipment based upon the noon average July wet-bulb of 68 F would be inadequate. Commercial practice is to choose a wet-bulb temperature for refrigeration design purposes which is not exceeded during more than 5 to 8 per cent of the summer hours (75 F for New York City) with somewhat lower requirements for steam turbines and internal combustion engines. This difference is made because the heaviest load on a refrigerating plant is coincident with high wet-bulb temperatures, whereas the heaviest electric power demand occurs either in the winter or after nightfall in summer, when the wet-bulb temperature is low. Table 1, Chapter 7, shows design wet-bulb temperatures which will not be exceeded more than 8 per cent of the time in an average summer.

Knowing the hot water temperature and the wet-bulb temperature for which the equipment must be designed, the cold water temperature must be chosen to place the requirement within the efficiency range of the type of atmospheric water cooling apparatus to be used. Efficiency of atmospheric water cooling apparatus is expressed as the percentage ratio of the actual cooling range to the possible cooling range. Since the wet-bulb

temperature of the entering air is the lowest temperature to which the water could possibly be cooled this is:

Percentage cooling efficiency of atmospheric water cooling equipment =

$$\frac{(\text{hot water temperature} - \text{cold water temperature}) \times 100}{\text{hot water temperature} - \text{wet-bulb temperature of entering air}}.$$

Efficiencies of various types of atmospheric water cooling apparatus vary through wide limits, depending upon air velocity, concentration of water per square foot of area, and the type of equipment. The commercial range of efficiencies is given in Table 3 although unusual designs may operate outside these ranges.

From consideration of the factors which include the cooling range and design wet-bulb temperature, the quantity of water required can be calculated from the amount of heat to be dissipated. The normal amounts of heat to be removed from various processes of the cooling equipment are:

Compressor refrigeration	220 to 270 Btu per minute per ton.
Condenser turbine	950 to 980 Btu per pound of steam.
Steam jet refrigerating apparatus	1030 to 1150 Btu per pound of steam.
Diesel engine	2800 to 4500 Btu per horsepower.

Cooling Ponds

A natural pond is often used as a source of condensing water. The hot water should be discharged close to the surface at the shore line. Natural air movement over the surface of the water will cause evaporation and carry away heat. Because increased density due to the loss of heat causes the cooled water to sink to the bottom of the pond, the suction connection for intake water should be placed as far below the surface as possible, and at as great a distance from the discharge as practicable.

Spray Cooling Ponds

The spray pond consists of a basin, above which nozzles are located to spray water up into the air. Properly designed spray nozzles break up the water into small drops, but not into a mist because the individual drops must be heavy enough to fall back into the basin and not drift away with the air movement. The water surface exposed to the air for cooling is the combined area of all the small drops. Since the rate of heat removal by atmospheric water cooling is a function of the area of water exposed to the air, the difference in temperature between the water and the wet-

TABLE 3. EFFICIENCY OF ATMOSPHERIC WATER COOLING EQUIPMENT

Equipment	Cooling Efficiency—Per Cent		
	Minimum	Usual	Maximum
Spray Ponds	30	40 to 50	60
Spray Towers	40	45 to 55	60
Natural Draft Deck or Atmospheric Towers	35	50 to 70	90
Mechanical Draft	35	55 to 75	90

bulb temperature of the air, the relative velocity of air and water, and the duration of contact of the air with the water, a much larger quantity of heat may be dissipated in a given area with the spray pond than with the cooling pond, because of (1) the speed with which the drops travel as they are propelled into the air and fall back into the water basin, (2) the increased wind velocity at a point above the surrounding structures or terrain, (3) the increased volume of air used, and (4) the vastly increased area of contact between air and water.

Spray pond efficiencies are increased by (1) elevating the nozzles to a higher point above the surface of the water in the basin, (2) increasing the spacing between nozzles of any one capacity, (3) using smaller capacity nozzles to decrease the concentration of water per unit area, and (4) using smaller nozzles and increasing the pressure to maintain the same concentration of water per unit area. Usual practice is to locate the nozzles from 3 to 7 ft above the edge of the basin, to supply from 5 to 12 lb pressure at the nozzles, using nozzles spraying from 20 gpm to 60 gpm each and spacing them so the average water delivered to the surface of the pond is from 0.1 gpm per square foot in a small pond to 0.8 gpm per square foot in a large pond.

Increasing the pressure, spacing the nozzles farther apart, or increasing the elevation of the nozzles will increase the cross-section of spray cloud exposed to the air, and therefore increase the quantity of air coming in contact with the water. Best results are obtained by placing the nozzles in a long relatively narrow area located broadside to the wind.

Spray ponds may be located on the ground, or they may be placed on roofs. To prevent excessive drift loss, or the carrying of entrained water beyond the edge of the pond by the air on the leeward side, louver fences are required for roof locations and for those ground locations where space is so restricted that the outer nozzles cannot be located at least 20 ft to 25 ft from the edge of the basin. Such fences usually are constructed of horizontal louvers overlapping so the air is forced to turn a corner in passing through the fence, and the heavier drops of water are thrown back, owing to their inertia. The louvers also restrict the flow of air, particularly at the higher wind velocities, and thus further reduce the possibility of water being carried off. The height of an effective fence should be equal to the height of the spray cloud. Louver boards are preferably of red gulf cypress or California redwood supported on cast-iron, steel or wood posts. Where building ordinances forbid the use of combustible materials, sheet metal is customarily used.

Algae growths, during warm weather, in cooling towers and spray ponds may be eliminated while the plant is in operation by the use of potassium permanganate. This chemical can be dissolved at the rate of 1 lb in 1¼ to 1½ gal of hot water. About 10 parts of permanganate should be used per million parts of cooling water.

The permanganate attacks the algae, forms a brown covering over it, and causes it to settle. Enough of the permanganate solution should be added periodically to cause the water to have a pink color for a period of from 15 to 20 min. Small additions of the permanganate daily do not give concentrations which are effective. The best results are obtained when sufficient quantities are added periodically at intervals of several

weeks, the time intervals being dependent upon local operating conditions. The chemical is non-poisonous and is non-corrosive when used as directed.

Spray Cooling Towers

Where not more than 30,000 Btu per minute are to be dissipated, the spray cooling tower is a satisfactory apparatus. The word *tower* in this connection is somewhat of a misnomer as the apparatus is essentially a narrow spray pond with a high louver fence. As usually built, the nozzles spray down from the top of the structure and the distance from the center of the nozzle system to the fence on either side is not more than half the distance that the nozzles are elevated above the water basin. Heights range from 6 ft to 15 ft and the total width of a structure is not usually greater than its height. Spray cooling towers occupy less space on small jobs than spray ponds of equivalent capacities because the towers have a capacity of from 0.6 gpm to 1.5 gpm per square foot of tower area. The louvers are continually wet, and so add to the surface of water exposed to the cooling air.

Natural Draft Deck Type Towers

In past years most of the atmospheric water cooling on refrigeration work has been done with natural draft deck type towers, which are also referred to as *wind* or *atmospheric* towers. These towers consist of heavy wooden or steel framework from 15 to 80 ft high and from 6 to 30 ft wide, having open horizontal lattice-work platforms or decks at regular intervals from top to bottom, and a catch basin at the foot. The hot water is distributed over the upper part of the structure by means of troughs, splash heads, or nozzles, and it drips from deck to deck down to the basin. The object of the decks is to arrest the fall of the water so as to present efficient cooling surfaces to the air, which passes through the tower parallel to the decks. The decks also add to the area of water surface exposed to the air, but since they furnish a resistance to air flow, too many decks are a detriment.

To prevent the loss of water on the leeward side of the tower, wide splash boards are attached at regular intervals from top to bottom. These boards or louvers extend outward and upward, and in most designs the top edge of each louver extends above the bottom edge of the one above it.

Efficiency of a deck tower is improved, within limits, by increased height, increased length, or increased width. The first two increase the area of water exposed to the wind, and the latter increases the time of contact of the air with the water.

Wind Velocities on Natural Draft Equipment

Since natural air movement is the prime requirement for a deck type tower, spray cooling tower, or spray pond, the apparatus must be designed to produce the desired cooling on days when the wind velocity is below average when the wet-bulb temperature is at the maximum chosen for design, and when the plant is operating at full load. The apparatus must also, for best results, be located with its longest axis at right angles to the direction of the prevailing hot weather breeze. Table 1, Chapter 7, gives the average summer wind velocities and directions in representative cities. Natural draft cooling equipment should be designed to operate properly with *not more than one-half* of the *average* wind velocity, and in

no case for a wind velocity of more than 5 mph. It is obvious that natural draft towers and other natural draft equipment must be so located that they are not obstructed by trees, buildings, or other wind deflectors.

Mechanical Draft Towers

Mechanical draft towers usually consist of vertical shells, constructed of wood, metal, or masonry, in which water is distributed uniformly at the top and falls to a collecting basin at the bottom. The inside of the tower may be filled with wood checker-work over which the water drips, or the water surface may be presented to the air by filling the entire inside of the structure with spray from nozzles. Air is circulated through the tower from bottom to top by forced or induced draft fans. Since the air flows counter to the water, the air is in contact with the hottest of the water just before leaving the top of the tower, and each unit of air picks up more heat than a similar unit would on natural draft equipment, so the mechanical draft tower cools water by using less air than the other types of equipment need. As movement of the air through the towers is obtained by power-consuming fans, it is essential that the air used be reduced to a minimum so as to secure the lowest possible operating cost.

The efficiency of a mechanical draft tower is increased by increasing height, area, or air quantity. Increasing the height increases the length of time the air is in contact with the water without affecting seriously the fan power required, but it increases the pumping power needed. Increasing the area while maintaining constant fan power increases the air quantity somewhat and because of lowered velocities it increases the time this air is in contact with the water. The surface area of water in contact with the air is increased in both cases. Increasing the air quantity decreases the time the air is in contact with the water, but, since a greater quantity is passing through, the average differential between the water temperature and the wet-bulb temperature of the air is increased, and this speeds up the heat transfer rate. Increased air quantities are obtained only at the expense of increased fan power, which increases approximately as the cube of the air quantity. Air velocities through mechanical draft towers vary from 250 to 600 fpm over the gross area of the structure.

Mechanical draft water cooling equipment may be set up inside buildings, where it usually draws its air supply from the general space in which it is installed, and discharges its exhaust air through a duct to the outside. Indoor cooling towers may be either of the wood-filled or the spray-filled type. In many cases where little height but considerable area is available, water is cooled in a spray-filled structure similar to an air washer, with the air passing horizontally through the apparatus and being discharged through a duct to the outside. Such apparatus does not have the counterflow advantage of the vertical mechanical draft water cooling equipment, and therefore requires a much larger excess of air for proper operation. Air velocities and operating powers are considerably above those required by vertical mechanical draft water cooling equipment.

Cooling Tower Design

The method of design of equipment for energy transfer from water to an air-water vapor mixture is similar to that used for absorption equip-

ment. Details of this procedure are available[2, 3, 4] and its application to the problem of the cooling tower operating at atmospheric pressure is illustrated by the following development. The nomenclature used is as follows:

a = overall average wetted area, square feet per cubic foot of tower volume.
c = specific heat of liquid water, Btu per (pound) (degree Fahrenheit).
e = effectiveness.
ε = natural base.
G = weight rate of flow of air, pounds of dry air per hour.
h = enthalpy, Btu per pound of dry air.
h_a = enthalpy of air-vapor mixture, Btu per pound of dry air.
h'' = enthalpy of saturated air-vapor mixture at water temperature, Btu per pound of dry air.
K = overall energy unit conductance, Btu per (hour) (square feet) (Btu per pound).
Ka = overall rate coefficient, Btu per (hour) (cubic feet) (Btu per pound).
L = water rate, pounds per hour.
lm = logarithmic mean.
S = average cross-sectional area of cooling tower for air flow, square feet.
t = water-main body temperature, degrees Fahrenheit.
t_{wb} = wet-bulb temperature, degrees Fahrenheit.
V = tower volume, cubic foot.

Subscripts 1 and 2 refer to water entrance and exit sections respectively, for the counter-flow tower.

A section of a typical counter-flow tower is shown in Fig. 4. If the reduction in water rate due to evaporation within the volume is neglected, the energy balance for this differential section of the exchanger volume may be written as:

$$L\,c\,dt = G\,dh \tag{2}$$

The potential for net energy transfer due to heat and mass transfer from the water to the mixture in contact with it may be expressed with reasonable accuracy as the difference between the enthalpy of saturated air at the water temperature, h'', and the enthalpy of the main stream air vapor mixture[5], h_a. The rate of energy transfer is given by the expression:

$$Ka\,(h'' - h_a)\,dV \tag{3}$$

which equation defines the overall rate coefficient, Ka; the latter being the product of the overall energy unit conductance, K, and the ratio of the transfer surface to the exchanger volume, a.

Equations 2 and 3 are conveniently illustrated by means of the temperature-enthalpy diagram of Fig. 5. Equation 2 indicates that the succession of air and water states existing in the exchanger sections must combine to form a straight line (for L = constant) on the temperature enthalpy diagram. The slope of this operating line is $\frac{dh}{dt} = \frac{Lc}{G}$. Since

[2]Principles of Chemical Engineering, by W. H. Walker, W. K. Lewis, W. H. McAdams and E. R. Gilliland (McGraw-Hill Co., New York City, 1937, p. 480).

[3]Absorption and Extraction, by T. K. Sherwood (McGraw-Hill Co., New York City, 1937, p. 91).

[4]Performance Characteristics of a Mechanically Induced Draft, Counterflow, Packed Cooling Tower, by A. L. London, W. E. Mason and L. M. K. Boelter (*A.S.M.E. Transactions*, January, 1940, Vol. 62, p. 41).

[5]Determination of Unit Conductances for Heat and Mass Transfer by the Transient Method, by A. L. London, H. B. Nottage and L. M. K. Boelter (*Industrial and Engineering Chemistry*, April, 1941, Vol. 33, p. 467).

the heat capacity of water is approximately unity, this slope is the ratio of the water to the air rate. Equation 3 indicates that the potential for energy transfer at any section is the difference between the enthalpy of saturated air at the main-body water temperature at that section and the enthalpy of the air stream in contact with that water. This potential is the difference in the ordinates of the saturation and operating lines for the water temperature at the plane in the tower which is under consideration.

Combination of Equations 2 and 3 results in the expression:

$$Gdh = Ka\,(h'' - h_a)\,dV \tag{4}$$

Integrating this equation over the length of the exchanger:

$$\int_2^1 \frac{G}{Ka}\,\frac{dh}{h'' - h_a} = V \tag{5}$$

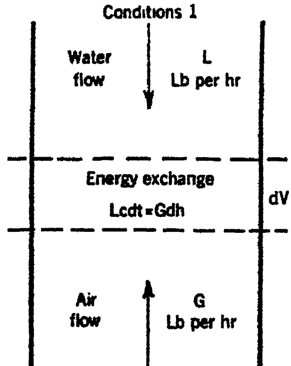

FIG. 4. SECTION OF TYPICAL COUNTER-FLOW TOWER

The integration of the left side of Equation 5 determines the tower volume required to achieve the desired energy exchange. This summation is readily accomplished for counter and parallel flow arrangements. *G* and *Ka* are usually independent of the tower volume and Equation 5 then becomes:

$$NTU = \int_2^1 \frac{dh}{h'' - h_a} = \frac{KaV}{G} \tag{6}$$

Where *NTU* is defined as the *Number of Transfer Units* and is a measure of the difficulty of the cooling process.

The integration is made numerically or graphically. In the graphical integration $\frac{1}{h'' - h_a}$ is evaluated as a function of h_a. This determination involves the use of the energy balance equation integrated from one section to the section in question. The area under the curve between any two abscissae is the number of transfer units required to change the air state from h_2 to h_1.

An approximate value for the number of transfer units can also be determined by a simple graphical method of direct construction on the

temperature enthalpy diagram[6]. This method cannot be applied very satisfactorily to cooling towers as the operating range is small and the value of the NTU is near unity.

When the relationship between the enthalpy of the saturated air and the temperature is linear over the range of water temperatures involved, it can be shown[7] that the logarithmic mean of the terminal potentials, h_{lm}, is the correct driving force. This is true to a good approximation when the water cooling does not exceed 15 F. The approximation to linearity may be determined by inspection of Table 6, Chapter 1, or the temperature enthalpy diagram of Fig. 5. If the logarithmic mean is a valid potential, Equation 6 may be written:

$$\frac{h_1 - h_2}{\Delta h_{lm}} = \frac{KaV}{G} \tag{7}$$

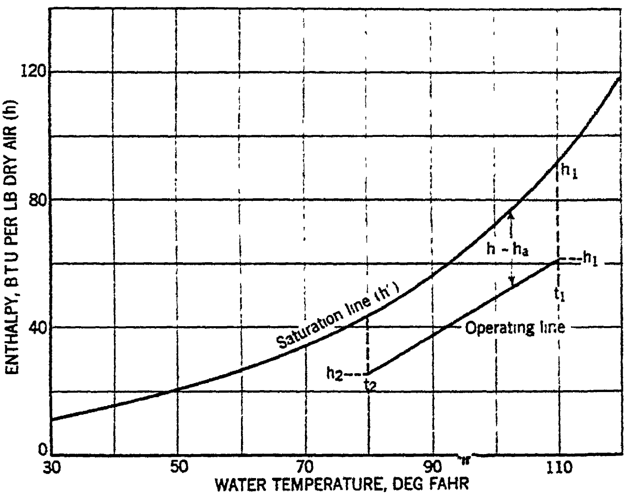

FIG. 5. TEMPERATURE ENTHALPY DIAGRAM FOR AIR WATER VAPOR MIXTURE SHOWING THE OPERATING LINE FOR EXAMPLE 1

and the need for the numerical integration for the determination of the tower volume is eliminated.

The overall rate coefficient, Ka, must be known if the tower volume is to be determined. Experiments conducted on towers containing different packing construction have yielded some magnitudes of this coefficient, evaluated on an overall basis. These data are presented in Fig. 6 as a function of the gas mass velocity through the packing, and apply only to the particular packing structure for which they were obtained. The overall rate coefficient (Ka) may also be a function of the water rate, since a reduction of the water rate may reduce the wetted area within the exchanger[8]. The results included in Fig. 6 probably represent magnitudes

[6]Graphical Method of Determining Number Transfer Units, by T. Baker (*Industrial and Engineering Chemistry*, August, 1935, Vol. 27, p. 977).

[7]Loc. Cit. Note 3, p. 79.

[8]Loc. Cit. Note 4.

of Ka which were obtained for complete wetting. Within the cooling tower operating range the overall rate coefficient for energy transfer is nearly the same numerically as the overall rate coefficient for mass transfer. The conditions of test corresponding to the data presented in Fig. 6 are not well enough known in most cases to warrant recomputation of Ka. Therefore the magnitudes of the overall rate coefficient for mass transfer presented in Fig. 6 may be used directly in Equations 5 and 6, the units of Ka in these equations being Btu per (hour) (cubic foot) (Btu per pound).

A typical design procedure is illustrated in the example following;

Example 1. The rate of air flow, arbitrarily assumed in the data given, is related to the tower volume by economic considerations. A balance between air rate and tower volume rests on consideration of the costs of producing air flow and of the tower construction[9]. A counter-flow forced draft cooling tower is to cool 36,000 lb of water per hour from an initial temperature of 110 F to a final temperature of 80 F. Air having an initial condition of 65 F dry-bulb and 58 F wet-bulb temperature will be forced through the tower counter to the direction of water flow at the rate of 30,000 lb of dry air per hour.

The cross-section of the tower is to be 8 ft x 8 ft and the packing is to be of the type producing a rate coefficient as indicated in curve No. 2 of Fig. 6. For this type of packing the average cross-sectional area for air flow will be 36 sq ft.

Solution:

Initial air enthalpy = 25.1 Btu per pound.

Final air enthalpy:

$$(h_1 - h_2) = \frac{Lc}{G}(t_1 - t_2)$$

$$(h_1 - 25.1) = \frac{36{,}000 \times 1}{30{,}000}(110 - 80)$$

$$h_1 = 61.1 \text{ Btu per pound dry air.}$$

TABLE 4. NUMERICAL INTEGRATION FOR THE NUMBER OF TRANSFER UNITS

Water Temperature Interval Deg F	Mean Water Temperature Deg F	Mean Air Enthalpy, h_a Btu per Lb Dry Air	Saturated Air Enthalpy, h'' Btu per Lb Dry Air	Enthalpy Potential $h'' - h_a$	$\frac{\Delta h}{h'' - h_a}$
80–82	81	26.3	44.6	18.3	0.131
82–84	83	28.7	46.9	18.2	0.132
84–86	85	31.1	49.2	18.1	0.133
86–88	87	33.5	51.7	18.2	0.132
88–90	89	35.9	54.4	18.5	0.130
90–92	91	38.3	57.1	18.8	0.128
92–94	93	40.7	60.0	19.3	0.124
94–96	95	43.1	63.0	19.9	0.121
96–98	97	45.5	66.2	20.7	0.116
98–100	99	47.9	69.6	21.7	0.111
100–102	101	50.3	73.2	22.9	0.105
102–104	103	52.7	77.0	24.3	0.099
104–106	105	55.1	80.9	25.8	0.093
106–108	107	57.5	85.1	27.6	0.087
108–110	109	59.9	89.5	29.6	0.081
					1.723

[9]Loc. Cit. Note 2, p. 142.

A numerical integration (Table 4) is employed to determine the Number of Transfer Units (NTU) required. Temperature increments of 2 F are used between successive determinations of the quantity $\frac{1}{h'' - h_a}$. The energy balance indicates that the enthalpy increments corresponding to these temperature increments are:

$$\Delta h = \frac{Lc}{G}\,\Delta t = \frac{36{,}000}{30{,}000} \times 2 = 2.4 \text{ Btu per pound.}$$

The result of the integration is that the Number of Transfer Units required (NTU) = 1.72.

Unit gas mass velocity, $\frac{G}{S} = \frac{30{,}000}{36} = 830$ lb per (hour) (square foot).

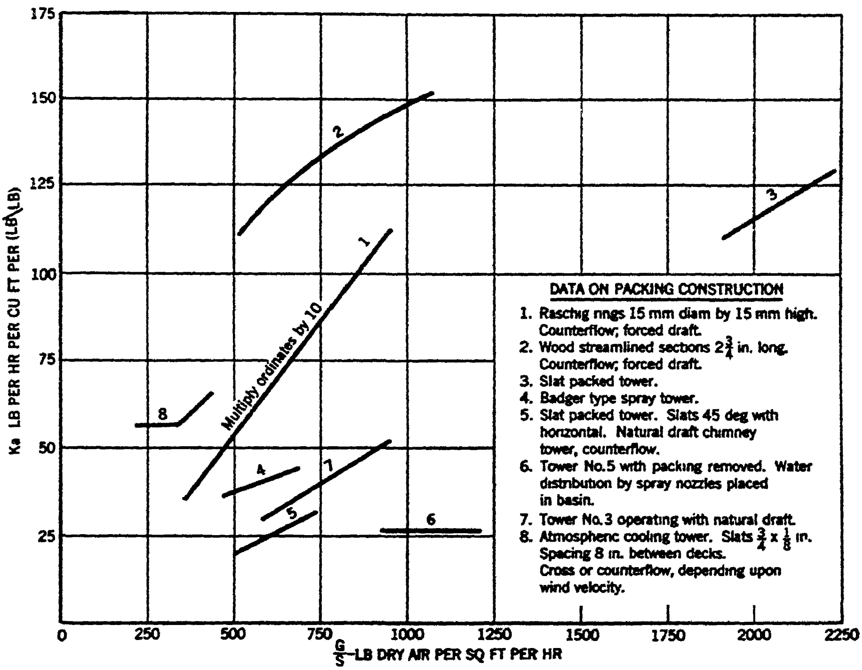

FIG. 6. UNIT CONDUCTANCES FOR VARIOUS TYPES OF PACKING CONSTRUCTION

From Fig. 6, Ka = 138 Btu per (hour) (cubic foot) (Btu per pound).
The tower volume required is:

$$V = \frac{G}{Ka}\,(NTU) = \frac{30{,}000}{138} \times 1.72 = 217 \times 1.72 = 374 \text{ cu ft.}$$

Height of the packed section is:

$$\frac{374}{8 \times 8} = 5.9 \text{ ft.}$$

The graphical solution for the Number of Transfer Units required for the desired performance is plotted in Fig. 7. $\frac{1}{h'' - h_a}$ is plotted as a function of h, and the area under the curve from the initial enthalpy of the air, 25.1 Btu per pound, to the final enthalpy of the air, 61.1 Btu per pound is 1.72, the Number of Transfer Units required. The effect of the rapid decrease in potential due to the cooling of the water is indicated

by comparison of areas A_1, A_2, and A_3 of Fig. 7. Each represents the Number of Transfer Units required to achieve a water temperature reduction of about 10 F.

$$A_1 = 0.471\ NTU\ (110 \text{ to } 100\text{ F})$$
$$A_2 = 0.595\ NTU\ (100 \text{ to } 90\text{ F})$$
$$A_3 = 0.657\ NTU\ (90 \text{ to } 80\text{ F})$$

The use of the logarithmic mean driving potential is illustrated by applying Equation 7 to example 1:

$$NTU = \frac{(Ka)\ V}{G} = \frac{h_2 - h_1}{\Delta h_{lm}}$$

$$\Delta h_{lm} = \frac{(h_1' - h_1) - (h_2'' - h_2)}{ln_e \dfrac{h_1' - h_1}{h_2'' - h_2}} = \frac{30.7 - 18.4}{ln_e \dfrac{30.7}{18.4}} = 24$$

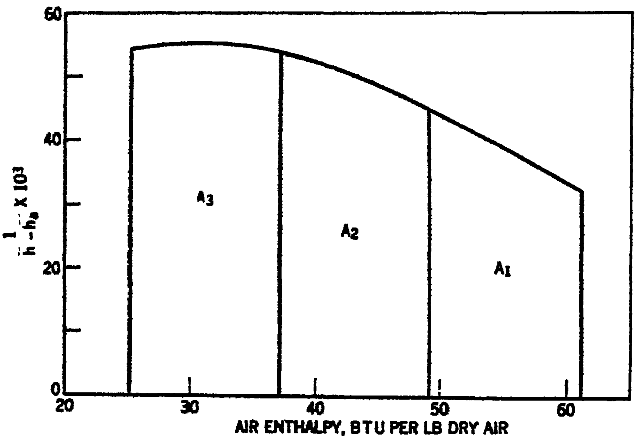

FIG. 7. GRAPHICAL INTEGRATION TO DETERMINE NUMBER OF TRANSFER UNITS REQUIRED FOR DESIRED OPERATING CONDITIONS OF EXAMPLE 1

$$NTU = \frac{61.1 - 25.1}{24} = 1.5$$

The Number of Transfer Units required as determined by use of the logarithmic mean driving potential equals 1.5 which compares favorably with the correct magnitude, 1.72.

The application of the foregoing design method to atmospheric towers is difficult because rate coefficients and flow conditions are not yet well defined for such equipment. If these are known, application of Equation 7 to sections of the tower small enough to justify use of the logarithmic mean potential will yield the tower volume required for each section. The sections must be taken perpendicular to the path of water flow. A correction to adjust the logarithmic mean potential, evaluated as for counter-flow, to the reduced effectiveness of cross-flow, has been derived for heat transfer and may be applied to this case[10].

[10]Heat Transmission, by W. H. McAdams (McGraw-Hill Co., New York City, 1933, p. 157).

Atmospheric towers operate with natural draft, produced in a vertical direction by the stack action of the tower structure, at zero velocity of the approach wind. Approach wind of sufficient magnitude (the magnitude depending on the baffle arrangement which is designed to reduce drift) will cause cross-flow augmenting the natural draft. An adequate design requires the consideration of both flow conditions.

Expression for Cooling Tower Performance

The performance of a cooling tower is described in terms of its effectiveness as an energy exchanger. The effectiveness is defined as the ratio of the energy actually exchanged to the energy available for exchange.

Effectiveness expressions:

Case 1. The slope of the operating line on the $t - h$ diagram exceeds the slope of the saturation line in the region of water temperatures considered.

$$e = \frac{h_1 - h_2}{h_1^{s} - h_2} \tag{8}$$

Case 2. The slope of the saturation line exceeds that of the operating line.

$$e = \frac{h_1 - h_2}{\frac{Lc}{G}(t_1 - t_{wb})} = \frac{t_1 - t_2}{t_1 - t_{wb}} \tag{9}$$

This equation represents the ***approach to wet-bulb.***

Usual tower operating conditions conform to Case 1. Because of the curvature of the saturation line, operating conditions may present themselves to which neither Case 1 nor 2 applies. Since a simple expression for the intermediate case is not available, the expression of Case 1 may be utilized for the small number of operating conditions falling into the intermediate classification.

Make-Up Water

Since the atmospheric water cooling equipment performs its functions chiefly by evaporating a portion of the water in order to cool the re-

TABLE 5. COMPARISON OF VARIOUS TYPES OF ATMOSPHERIC WATER COOLING EQUIPMENT

Figures indicate order of desirability

	Cooling Pond	Spray Pond	Spray Tower	Deck Tower	Mechanical Draft	Indoor Tower
Cost	x	2	1	3	4	5
Area	5	4	3	2	1	x
Height	1	2	3	4–5	4–5	x
Weight per square foot	x	x	1	3	4	2
Independence of wind velocity	6	3	4	5	1–2	1–2
Drift nuisance	1	6	5	4	2–3	2–3
Make-up water required	1	6	5	4	2–3	2–3
Pumping head	1	2	4	5	3	6
Maintenance	2	1	3	4	5	6
Suitability for congested districts	x	5	4	3	1	2
Water quantity required for definite result	6	5	4	1–2	1–2	3

xNot comparable.

mainder, there is a continual drain on the quantity of water in the system, and this loss must be replaced. Approximately 1 gal of water is lost for every 1000 gal of water cooled per degree of cooling range; so if 1000 gpm of water are cooled through a 10 F range, 10 gpm of water will be required to replace evaporated water. Replacement supply is usually regulated by a float control valve. Because the evaporation of the water leaves behind the salts which the water contained, high concentration of salts may make chemical treatment of the make-up water necessary to avoid excessive deposits in the condensers. An additional amount of make-up water must be added to replace windage, or *drift loss*. This additional amount of water varies from 0.1 to 3 per cent of the quantity of water being circulated this percentage depending upon the type of equipment and the wind velocity.

Winter Freezing

If atmospheric water cooling equipment is operated in freezing weather, the water may be cooled below freezing temperature so ice forms and collects until its weight causes damage. To obviate freezing during continued operation, the efficiency of the apparatus may be lowered. This is done on the spray pond and the spray cooling tower by reducing the quantity of water fed to the apparatus, thereby lowering the pressure at the nozzles and increasing the size of the drops produced. On the deck tower the upper system may be shut off and a secondary distribution system put in service midway down the height of the tower. The water will be kept above freezing because it will have shorter contact with the air. The mechanical draft tower can be protected by reducing the air flow through the tower, by stopping or reducing the speed of the fans, or by partially closing dampers.

If the system is operated intermittently in freezing weather, water in the basin may freeze and the expansion of the ice may do harm. Freezing during intermittent operation can be prevented only by draining the water basin when it is out of service. On small roof installations, a tank large enough to hold all the water in the system is often installed inside the building and the basin is drained into this by gravity, the pump suction being taken from this inside tank.

A comparison of various types of water cooling equipment is given in Table 5.

Chapter 28

AIR POLLUTION

Classification of Air Impurities, Dust Concentrations, Air Pollution and Health, Occlusion of Solar Radiation, Smoke and Air Pollution Abatement, Dust and Cinders, Nature's Dust Catcher

THE particulate impurities which contribute to atmospheric pollution include carbon from the combustion of fuels, particles of earth, sand, ash, rubber tires, leather, animal excretion, stone, wood, rust, paper, threads of cotton, wool, and silks, bits of animal and vegetable matter, and pollen. Microscopic examination of the impurities in city air shows that a large percentage of the particles are carbon.

CLASSIFICATION OF AIR IMPURITIES

The most conspicuous sources of atmospheric pollution may be classified in various ways, as dusts, fumes and smoke. In Fig. 1, the classification is by particle size, but recent practice favors differentiation by method of formation. Thus, *dusts* are composed of particles produced by disintegration of larger material, as by crushing or grinding, whereas *fumes* are produced by condensation, and *smoke* consists of the finer carbon particles resulting from incomplete combustion. Similarly, *mists* are formed by the breaking up of liquids and *fogs* by condensation of vapors. There is as yet, however, no general agreement on these terms.

Dusts tend to settle without agglomeration, fumes to aggregate and smoke to diffuse. Particles which approach the common bacteria in size—from 1 to 10 microns—are difficult to remove from air and are apt to remain in suspension unless they can be agglomerated by artificial means. The term fly-ash is applied to solid ashy material, usually finely divided, that is a constituent of the effluent gases from coal-fired furnaces. Cinders denote the larger solid constituents which may be entrained by furnace gases.

Particles larger than 10 microns are unlikely to remain suspended in air currents of moderate strength, but settle out by gravity at speeds dependent upon the shape, size and specific gravity of the particle and upon the wind velocity. These larger particles are of major interest to the engineer in the solution of nuisance problems; on the other hand, it is mainly the smaller particles that are of hygienic significance. A notable exception to this size limitation in the latter case is the common hay-fever producing pollen such as that from ragweed. Pollen grains may be anything from fragments 15 microns or less in diameter to whole pollens 25 microns or more in size.

The lower limit of size of particle visible to the naked eye cannot be stated definitely. It depends not only upon the individual, but also upon the shape and color of the particle and upon the intensity of light. Under

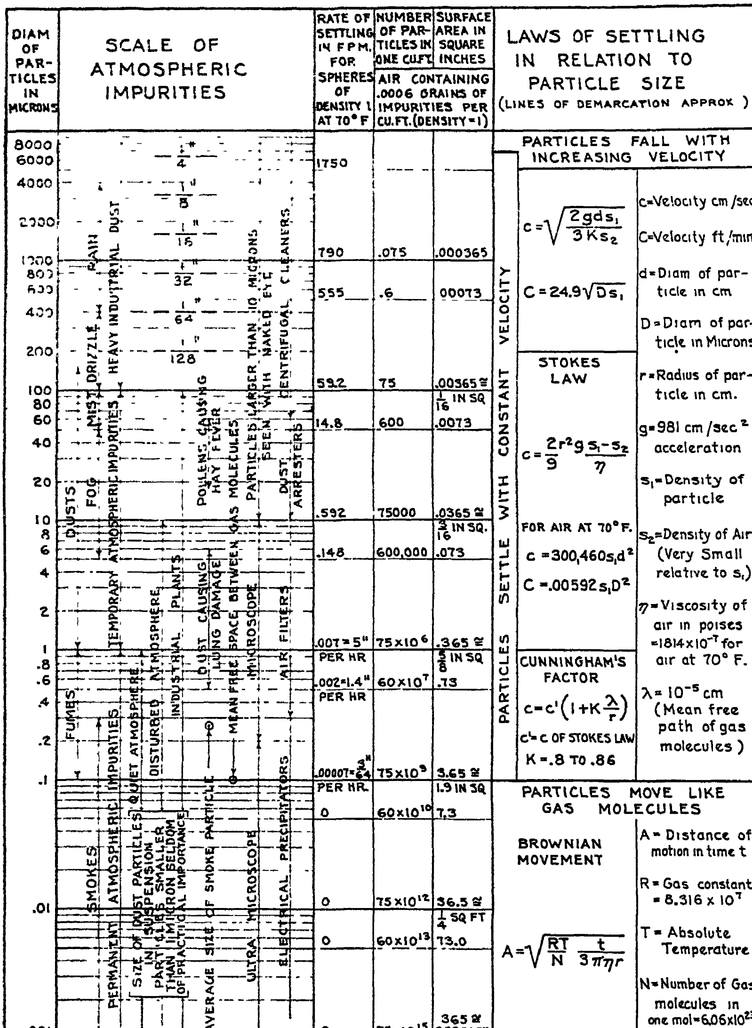

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FIG. 1. SIZES AND CHARACTERISTICS OF AIR-BORNE SOLIDS

ideal conditions a particle 10 microns, or even less, may be recognized, while under less favorable conditions it may be difficult or even impossible to recognize a particle of 50 microns. The lower limit of visibility should, therefore, be considered as within a range (as above) or optimum conditions stated.

Mineral particles, such as grains of sand, bits of rock, volcanic ash, or fly-ash, can be transported long distances under unusual circumstances. Thus, the dust storms of 1935 in the Kansas district resulted in vast amounts of fine top soil being thrown high into the air. Solar illumination

TABLE 1. APPROXIMATE LIMITS OF INFLAMMABILITY OF SINGLE GASES AND VAPORS IN AIR AT ORDINARY TEMPERATURES AND PRESSURES[a]

Gas or Vapor	Lower Limit Volume in Per Cent	Higher Limit Volume in Per Cent	Gas or Vapor	Lower Limit Volume in Per Cent	Higher Limit Volume in Per Cent
Acetaldehyde	4.0	57.0	Hexane	1.2	6.9
Acetone	3.0	13.0	Hydrocyanic acid	5.6	40.0
Acetone[b]	2.5	-----	Hydrogen	4.1	74.0
Acetylene	2.5	80.0	Hydrogen sulphide	4.3	45.5
Acetylene[b]	2.3	-----	Illuminating gas	5.3	31.0
Allyl alcohol	2.4	-----	Iso-amyl alcohol	1.2	-----
Ammonia	16.0	27.0	Iso-butane	1.8	8.4
Amyl alcohol	1.2	-----	Iso-butyl alcohol	1.8	-----
Amyl chloride	1.4	-----	Iso-pentane	1.3	-----
Amylene	1.6	-----	Iso-propyl acetate	1.8	7.8
Benzene	1.4	6.7	Iso-propyl alcohol	2.6	-----
Benzine	1.1	-----	Methane	5.3	14.0
Blast-furnace gas	35.0	74.0	Methane[b]	5.0	15.0
Butane	1.9	8.4	Methyl acetate	3.1	15.5
Butyl acetate (30 C)	1.7	7.6	Methyl alcohol	6.7	36.0
Butyl alcohol	1.7	-----	Methyl bromide	13.5	14.5
Butylene	1.7	9.0	Methylbutyl ketone	1.2	8.0
Carbon disulphide	1.2	50.0	Methyl chloride	8.0	19.0
Carbon monoxide	12.5	74.0	Methyl cyclohexane	1.2	-----
Croton aldehyde	2.1	15.5	Methyl ethyl ether	2.0	10.1
Cyclohexane	1.3	8.3	Methyl ethyl ketone	1.8	10.0
Cyclopropane	2.4	10.3	Methyl formate	5.0	23.0
Decane	0.7	-----	Methyl propyl ketone	1.5	8.5
Dichlorethylene	9.7	12.8	Natural gas	4.8	13.5
Diethyl selenide	2.5	-----	Nonane	0.8	-----
Dioxan	2.0	22.0	Octane	0.9	-----
Ethane	3.2	12.5	o-Xylene	1.0	6.0
Ethyl acetate	2 0	11.5	Paraldehyde	1.3	-----
Ethyl alcohol	3.3	19.0	Pentane	1.4	7.8
Ethyl bromide	6.7	11.2	Propane	2.4	9.5
Ethyl chloride	4 0	14.8	Propyl acetate	1.8	8.0
Ethylene dichloride	6 2	15.9	Propyl alcohol	2.5	-----
Ethylene	3.0	29.0	Propylene	2.0	11.1
Ethyl ether	1.8	36.5	Propylene dichloride	3.4	14.5
Ethyl formate	2.7	16.5	Propylene oxide	2.1	21.5
Ethyl nitrite	3.0	-----	Pyridine (70 C)	1.8	12.4
Ethylene oxide	3.0	80.0	Toluene	1.3	6.7
Furfural (125 C)	2 1	-----	Vinyl ether	1.7	27.0
Gasoline	1.4	6.5	Vinyl chloride	4.0	22.0
Heptane	1.0	-----	Water gas	6 to 9	55 to 70

[a]Limits of Inflammability of Gases and Vapors, by H. F. Coward and G. W. Jones (*U. S. Bureau of Mines, Bulletin* No 279, 1939).

[b]Turbulent mixture.

as far east as Boston was affected noticeably and particles as large as 40 to 50 microns were actually carried half way across the continent before they settled out. In similar manner volcanic ash has been carried even further. It is not surprising, therefore, that fly-ash from furnace gases, cement dust and the like, can be carried for considerable distances and

occasionally the engineer is confronted with the problem of removing such material before the air in question is suitable for use in building ventilation.

The physical properties of the particulate impurities of air are summarized conveniently in the chart of Fig. 1.

In the case of gases, the objectionable features are the injurious physiological effects and the danger from inflammability. (See Table 1.)

Dust Concentrations

It is customary to report dust concentrations as grains per 1000 cu ft or milligrams per cubic meter (except for dusts that may cause pneumoconiosis, which are reported as so many particles per cubic foot of air). Gas concentrations are commonly recorded as milligrams per cubic meter or as parts per million or as per cent by volume. Typical ranges in dust concentrations as now found in practical applications are given in Table 2.

Table 2. Dust Concentration Ranges in Practical Applications[a]

Application	Grains Per 1000 Cu Ft	Mgs Per Cu M
Rural and suburban districts	0.2 to 0.4	0.4 to 0.8
Metropolitan districts	0.4 to 0.8	0.9 to 1.8
Industrial districts	0.8 to 1.5	1.8 to 3.5
Dusty factories or mines	4.0 to 80.0	10 to 200
Explosive concentrations (as of flour or soft coal)	4000 to 8000	10,000 to 20,000

[a]1 grains per 1000 cu ft = 2.3 mgs per cubic meter; 1 oz per cubic foot = 1 g per liter.

The engineer frequently desires information regarding the effects of various concentrations of gases or dusts upon man, as the success of a particular installation may depend upon the maintenance of air which is adequately clean. At the present time there are several organizations working on this problem all of them publishing literature of various kinds.[1] References to books covering the hygienic significance, determination and control of dust are listed at the end of this chapter.

AIR POLLUTION AND HEALTH

The prevention of various diseases which result from exposure to atmospheric impurities is an engineering problem. It is important for the engineer to insure, by proper ventilation, suitable environments for working or for general living. If the equipment used is to be successful, it must operate automatically as in the modern air conditioned theater or railroad train.

In Table 3 are given data on permissible concentrations of various substances, gases and dusts, which occur in industry. The prudent engineer will design equipment using these bench marks as the upper limits of pollution. In general it is good practice to avoid recirculation of air which originally contains toxic substances. Obviously there may be

[1]National Institute for Health, U. S. Public Health Service; Division of Labor Standards, U. S. Department of Labor; University of Toronto Medical School, Canada; Saranac Laboratories, Saranac Lake, N. Y.; Industrial Hygiene Foundation, Inc., Pittsburgh, Pa.; Harvard School of Public Health, Boston, Mass.; Haskell Laboratory, Wilmington, Del.; and the Departments of Health and of Labor in the United States and in various provinces of Canada.

exceptions to this rule, but it is one which is generally being followed in current practice.

Affections of the respiratory tract are associated with exposure to thick dust, and may follow inhalation of practically any kind of insoluble and non-colloidal dust. Atmospheric dust in itself cannot be blamed for causing tuberculosis, but it may aggravate the disease once it has started.[2]

TABLE 3. TOXICITY OF GASES, VAPORS, DUSTS AND FUMES[a]

SUBSTANCE	SINGLE EXPOSURES			MAX. ALLOWABLE AVERAGE CONCENTRATION FOR REPEATED EXPOSURES
	Rapidly Fatal	Dangerous for from ½ to 1 Hour	Max. Safe Concentration for ½ to 1 Hour	
	(ppm)	(ppm)	(ppm)	(ppm)
Ammonia	5,000–10,000	2500	300	100
Aniline	......	......	105–160	5
Arsine	250	15	6	1
Benzene	190	......	31–47	100
Carbon bisulfide	4800	3200–3850	960–1600	20
Carbon dioxide	80,000–100,000	......	......	......
Carbon monoxide	4000	1500–2000	400	100
Carbon tetrachloride	10,000	......	1000	100
Chlorine	900	14–21	3.5	1
Hydrogen cyanide	270	110–135	45–54	20
Hydrogen chloride	1250–1750	1000–1350	40–90	10
Hydrogen fluoride	660	50–250	10	3
Hydrogen sulfide	1000–2000	360–500	200–300	20
Methyl bromide	20,000–40,000	2000–4000	1000	......
Methyl chloride	150,000–300,000	20,000–40,000	7000	......
Nitrobenzene	......	......	200	5
Oxides of nitrogen	320–530	117–154	......	10
Phosgene	90	12.5	......	1
Phosphine	2000	400–600	100–200	2
Sulfur dioxide	400–500	150–190	50–100	10
Tetrachlorethane	7300	......	......	10
Toluene and xylene	......	......	......	200
Trichlorethylene	7800	......	3700	200

DUSTS:	
Containing high percentages of free silica	5 million[b]
Containing 25 to 35 percentages of free silica	10–20 million[b]
Containing small percentages of free silica	30 million[b]
Cadmium	0.1[c]
Lead	0.15[c]
Mercury	0.1[c]
Manganese	30[c]
Zinc	15[c]

[a]Adapted from Sayers and DallaValle, *Mechanical Engineering*, April, 1935; *American Standards Association; Journal of Industrial Hygiene and Toxicology*, June, 1940; and other authoritative sources.
[b]Particles (less than 10 microns) per cubic foot of air.
[c]Milligrams per cubic meter of air.

The sulphurous fumes and tarry matter in smoke are more dangerous than the carbon. In foggy weather the accumulation of these substances in the lower strata may be such as to cause irritation of the eyes, nose, and respiratory passages. The Meuse Valley fog disaster will probably become

[2]Physiological Response of the Peritoneal Tissue to Dusts Introduced as Foreign Bodies, by Miller and Sayers (*U. S. Public Health Reports*, 49:80, 1934).

a classic example in the history of gaseous air pollution. Released in a rare combination of atmospheric calm and dense fog, it is believed that sulphur dioxide and other toxic gases from the industrial region of the valley caused 63 sudden deaths, and injuries to several hundred persons.

Carbon monoxide from automobiles and from chimney gases constitutes another important source of aerial pollution in busy cities. During heavy traffic hours and under atmospheric conditions favorable to concentration, the air of congested streets may contain enough *CO* to affect those exposed over a period of several hours, particularly if their activities call for deep and rapid breathing. In open air under ordinary conditions the concentration of *CO* in city air is insufficient to affect the average city dweller or pedestrian.

Occlusion of Solar Radiation

The loss of light, particularly the occlusion of solar ultra-violet light due to smoke and soot, is beginning to be recognized as a health problem in many industrial cities. Measurements of solar radiation in Baltimore[3] by actinic methods show that the ultra-violet light in the country was 50 per cent greater than in the city. In New York City[4] a loss as great as 50 per cent in visible light was found by the photo-electric cell method.

The aesthetic and economic objections to air pollution are so definite, and the effect of air-borne pollen can be shown so readily as the cause of hay-fever and other allergic diseases, that means and expenses of prevention or elimination of this pollution are justified.

SMOKE AND AIR POLLUTION ABATEMENT

Successful abatement of atmospheric pollution requires the combined efforts of the combustion engineer, the public health officer, and the public itself. The complete electrification of industry and railroads, and the separation of industrial and residential communities would aid materially in the effective solution of the problem.

In the large cities where the nuisance from smoke, dust and cinders is the most serious, limited areas obtain some relief by the use of district heating. The boilers in these plants are of large size designed and operated to burn the fuel without smoke, and some of them are equipped with dust catching devices. The gases of combustion are usually discharged at a much higher level than is possible in the case of buildings that operate their own boiler plants.

In general, time, temperature and turbulence are the essential requirements for smokeless combustion. Anything that can be done to increase any one of these factors will reduce the quantity of smoke discharged. Especial care must be taken in hand-firing bituminous coals. (See Chapter 7.)

Checker or alternate firing, in which the fuel is fired alternately on separate parts of the grate, maintains a higher furnace temperature and thereby decreases the amount of smoke.

[3]Effects of Atmospheric Pollution Upon Incidence of Solar Ultra-Violet Light, by J. H. Shrader, M. H Coblentz and F. A. Korff (*American Journal of Public Health*, p. 7, Vol. 19, 1929).

[4]Studies in Illumination, by J. E. Ives (*U. S. Public Health Service Bulletin* No. 197, 1930).

Coking and firing, in which the fuel is first fired close to the firing door and the coke pushed back into the furnace just before firing again, produces the same effect. The volatiles as they are distilled thus have to pass over the hot fuel bed where they will be burned if they are mixed with sufficient air and are not cooled too quickly by the heat-absorbing surfaces of the boiler.

Steam or compressed air jets, admitted over the fire, create turbulence in the furnace and bring the volatiles of the fuel more quickly into contact with the air required for combustion. These jets are especially helpful for the first few minutes after each firing. *Frequent firings* of small charges shorten the smoking period and reduce the density. *Thinner fuel beds* on the grate increase the effective combustion space in the furnace, supply more air for combustion, and are sometimes effective in reducing the smoke emitted, but care should be taken that holes are not formed in the fire. A *lower volatile coal* or a higher gravity oil always produces less smoke than a high volatile coal or low gravity oil used in the same furnace and fired in the same manner.

The installation of more modern or better designed fuel burning equipment, or a change in the construction of the furnace, will often reduce smoke. The installation of a Dutch oven which will increase the furnace volume and raise the furnace temperature often produces satisfactory results.

In the case of new installations, the problem of smoke abatement can be solved by the selection of the proper fuel-burning equipment and furnace design for the particular fuel to be burned and by the proper operation of that equipment. Constant vigilance is necessary to make certain that the equipment is properly operated. In old installations the solution of the problem presents many difficulties, and a considerable investment in special apparatus is often necessary.

Legislative measures at the present time are largely concerned with the smoke discharged from the chimneys of boiler plants. Practically all of the ordinances limit the number of minutes in any one hour that smoke of a specified density, as measured by comparison with a Ringelmann Chart (Chapter 35), may be discharged.

These ordinances do not cover the smoke discharged at low levels by automobiles, and, although they have been instrumental in reducing the smoke emitted by boiler plants, they have, in many instances, increased the output of chimney dust and cinders due to the use of more excess air and to greater turbulence in the furnaces.

Legislative measures in general have not as yet covered the noxious gases, such as sulphur dioxide, nor sulphuric acid fog, which are discharged with the gases of combustion. Where high sulphur coals are burned, these sulphur gases present a serious problem.

DUST AND CINDERS

The impurities in the air other than smoke come from so many sources that they are difficult to control. Only those which are produced in large quantities at a comparatively few points, such as the dust, cinders and fly-ash discharged to the atmosphere along with the gases of combustion from burning solid fuel, can be readily controlled.

Dusts and cinders in flue gas may be caught by various devices on the market, such as fabric filters, dust traps, settling chambers, centrifugal separators, electrical precipitators, and gas scrubbers, described in Chapter 29.

The cinder particles are usually larger in size than the dust particles; they are gray or black in color, and are abrasive. Being of a larger size, the range within which they may annoy is limited.

The dust particles are usually extremely fine; they are light gray or yellow in color, and are not as abrasive as cinder particles. Being extremely fine, they are readily distributed over a large area by air currents.

The nuisance created by the solid particles in the air is dependent on the size and physical characteristics of the individual particles. The difficulty of catching the dust and cinder particles is principally a function of the size and specific gravity of the particles.

Lower rates of combustion per square foot of grate area will reduce the quantity of solid matter discharged from the chimney with the gases of combustion. The burning of coke, coking coal, and sized coal from which the extremely fine coal has been removed will not as a general rule produce as much dust and cinders as will result from the burning of non-coking coals and slack coals when they are burned on a grate.

Modern boiler installations are usually designed for high capacity per square foot of ground area because such designs give the lowest cost of construction per unit of capacity. Designs of this type discharge a large quantity of dust and cinders with the gases of combustion, and if pollution of the atmosphere is to be prevented, some type of catcher must be installed.

NATURE'S DUST CATCHER

Nature has provided means for catching solid particles in the air and depositing them upon the earth. A dust particle forms the nucleus for each rain drop and the rain picks up dust as it falls from the clouds to the earth. However, it was found in recent studies[5] that rain was not a good air cleaner of the material below about 0.7 micron.

[5] Atmospheric Pollution of American Cities for the years 1931-1933, by J. E. Ives et al (*U. S. Public Health Bulletin* No. 224, March, 1936).

REFERENCES

Bulletins, Industrial Hygiene Foundation, Inc., Pittsburgh, Pa.

Determination and Control of Industrial Dust, by J. J. Bloomfield and J. M. Dalla Valle (*U. S. Public Health Bulletin*, No. 217, 1935).

Journal of Industrial Hygiene and Toxicology, Harvard School of Public Health, Boston, Mass.

Reports of the National Silicosis Conference, Washington, D. C. (To be published by the U. S. Department of Labor.)

Symposiums on Silicosis, 1937 and 1939, Saranac Laboratory, Saranac, N. Y.

Symposium on Tuberculosis, 1941, Saranac Laboratory, Saranac, N. Y.

Industrial Dust, by Philip Drinker and Theodore Hatch, McGraw-Hill Co., N. Y.

Noxious Gases, by Y. Henderson and H. Haggard, Chemical Catalog Co., N. Y.

Occupation and Health, International Labour Office.

Preventive Medicine and Hygiene, by Milton J. Rosenau, D. Appleton-Century Co., N. Y.

Chapter 29

AIR CLEANING DEVICES

Damage Caused by Dust, Classification of Air Cleaning Devices, Viscous-Impingement Filters, Dry Air Filters, Air Washers, Electrical Precipitators, Cleaning of Gases from Exhaust Systems

IN this chapter the term *cleaning* is assumed to mean the removal of particulate matter from the air. The removal of foreign gases and vapors requires entirely different methods and is discussed in Chapter 40.

The cleaning of air involves the removal of many kinds of materials having a wide range of particle sizes and concentrations. The degree of air purification required varies widely, consequently, many types of devices having radically different design characteristics are available.

The various materials that pollute the air are discussed in Chapter 28, Fig. 1, which shows characteristics of particles ranging in size from 8000 to 0.001 microns. The importance of particles in the range from 0.1 to 0.001 microns is open to argument. Particles below 0.1 micron can be seen in some microscopes as specks of reflected light, and a few microscopes using ultra-violet light have a resolving power of 0.1 micron, but the smallest particle which is really resolved in microscopes using ordinary light is about 0.25 micron. The performance of particles below 0.1 micron is, therefore, controversial because no means have been developed for reliably counting or measuring the sizes of the particles.

Even if the discussion is limited to the range from 0.1 micron to 50 microns, from the smallest particle observable in the microscope to the smallest particle distinguishable to the naked eye, this range is so far outside the usual experience that it is difficult to visualize. If particles could be examined through a super microscope having a magnification of 250,000 diameters, a tobacco smoke particle of 0.1 micron would appear to be 1 in. in diameter, or approximately the size of a golf ball; a soft coal smoke particle 0.3 micron in diameter would appear like a baseball; a ragweed pollen grain 20 microns in diameter would appear 16.5 ft in diameter, while the 50 micron particle, just visible to the naked eye and able to pass through a 270 mesh screen, would appear to be 50 ft in diameter. Picturing this range in particle size from a golf ball to a sphere 50 ft in diameter may aid in appreciating the problem of cleaning air and the difficulty of devising any single test to adequately measure the performance of air cleaning devices under all conditions of service.

DAMAGE CAUSED BY DUST

Dust may cause damage in many ways, but usually it must first lodge on a surface. Larger particles settle rapidly out onto surfaces. The rate of fall of particles is given in Chapter 28, Fig. 1. Those visible to the unaided eye (50 microns or over) fall so rapidly that few remain in the air. However, any large ones which are carried into a room are almost certain to settle and are so noticeable when they have settled as to be very objectionable.

Any air movement, particularly over fabrics or unpolished surfaces, tends to deposit dust on the surface. The smallest particles observable in the microscope are deposited in this way.

A phenomenon of great importance in air conditioning and one not yet generally appreciated is thermal precipitation of dust. This is the tendency for dust to be deposited on any surface which is cooler than the adjacent air. It is largely responsible for outside walls becoming dirtier than partitions. In the case of plaster on wood lath, the dark streaks following the spaces between laths are very noticeable and frequently beams and other structural members can be traced by the difference in blackness of the wall or ceiling. Thermal precipitation deposits particles of all sizes apparently with very little differentiation as to size.

By keeping the surface temperature higher and more uniform, modern thermal insulation decreases the deposit of dust and makes it more uniform and less noticeable.

REQUIREMENTS

The removal of larger particles is always important because they quickly settle out of the air and become noticeable on surfaces as visible dustiness. In many applications the removal of these larger particles will constitute satisfactory performance, and a relatively simple device can be used.

In cities where large quantities of soft coal are burned, the air becomes contaminated with fine black smoke particles. These are deposited on walls, draperies, and other interior surfaces by air movement and by thermal precipitation. Their removal is much more difficult than that of the large particles.

Hay fever is usually due to pollen in the air, and many people, once this trouble has started, are sensitive to minute amounts of pollen, so that almost perfect cleaning may be required. Asthma may be caused by many things, including pollens and fine dusts. Many afflicted with either disease, who have not obtained relief from usual remedies, have found it in a room where the air is efficiently cleaned from small as well as large particles.

In air conditioning systems the maintenance of constant air flow is essential. In summer cooling, a decrease in air flow may cause a reduction in temperature of discharged air. This, combined with reduced velocity, may completely upset the air distribution objective resulting in drafts. The air cleaning equipment must function so that reduction in air flow due to the normal accumulation of dust will not cause faulty operation of the system.

In addition to requirements which are specified the following features are desirable:

1. Low resistance to air flow.
2. Ease of cleaning and maintenance.
3. Efficiency over a wide range of velocities.
4. Binding liquid to catch particles must not contaminate the air.

TESTING

The wide variety of materials and particle sizes which may be present in the air makes the testing of air cleaning devices difficult. Probably no standardized test can cover all of the conditions which may be encountered in service.

Tests have been devised which compare the ability of air cleaners to remove a certain artificial dust under specified conditions. This seems to be the most practical way of comparing various devices, but it may lead to misleading results if the dust is not representative of the dust to be removed in service.

The most common test is that specified by the A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work[1]. This Code specifies an artificial dust consisting of a mixture of dusts, which has passed through a 200 mesh screen. The efficiency is measured in terms of the ratio of the weight of dust removed to the weight of dust injected into the air. This is a well-worked-out method for testing the ability of a device to remove the coarser particles, but it is not a satisfactory measure of the ability of a device to remove extremely fine particles. In general, any weight method tends to measure the efficiency of removal of the larger particles present in the test dust. The largest particle specified in this Code is one just passing a 200 mesh screen or one 70 microns in diameter. Assuming a dust containing one such particle to 1,000,000 particles having a diameter of 0.1 micron, the 70 micron particle will have 3.43×10^8 times the weight of one of the 0.1 micron particles, all particles having the same density. If the cleaning device removes the one large particle but removes none of the 0.1 micron particles, the weight efficiency will be the ratio of the weight removed to the weight of dust injected or 99.71 per cent by weight even though only one particle out of 1,000,000 has been removed. This is obviously an exaggerated case, but illustrates the weakness of a weight test in measuring efficiency of removal of fine particles.

In testing air filters at the *National Bureau of Standards*, a much finer dust is used and the efficiency is measured by determining the relative blackness of pieces of filter paper through which air is passed[2]. The test dust used is a sample of dust collected by a precipitator in a local power plant, which of course is not as fine as atmospheric dust. However, this method of measuring efficiency can be used with atmospheric dust to test an air cleaning device under actual operating conditions. The efficiency is determined by drawing samples of filtered and unfiltered air through pieces of filter paper, the volumes of air being adjusted until

[1]A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 225).

[2]A Test Method for Air Filters, by Richard S. Dill (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 379)

equal blackness is obtained. If, for example, one unit of volume of unfiltered air gives the same blackness as four units of volume of filtered air, the efficiency is said to be 75 per cent. This method gives a much better measure of the tendency to blacken walls than the weight efficiency.

CLASSIFICATION OF AIR CLEANING DEVICES

Considering the wide variety of materials and particle sizes to be removed and the variety of requirements, it is natural that many kinds of devices are used which cannot be shown satisfactorily in a single simple classification. The following outline gives classifications on three different bases:

1. Methods of cleaning.
   - a. Automatic.
   - b. Non-automatic.
     - (1) Throw-away, replaceable elements.
     - (2) Manually cleaned in place.
     - (3) Removable for cleaning.
     - (4) Adsorption of gases and odors.
2. Principle of air cleaning.
   - a. Viscous-impingement filters.
   - b. Dry filters.
   - c. Washers.
   - d. Centrifugal devices.
   - e. Electrical precipitators.
   - f. Carbon adsorbers.
3. Classification according to application.
   - a. General air conditioning.
     - (1) Central cleaning system.
     - (2) Unit ventilator.
     - (3) Window installation.
     - (4) Warm air furnace.
   - b. Removal of smoke and fumes from stack gases.
   - c. Collection of dusts from exhaust systems.
   - d. Removal of odors from intake, exhaust and recirculation.

VISCOUS IMPINGEMENT TYPE FILTERS

The principle of air cleaning used in viscous filters is that of *adhesive impingement*. Dust and dirt in the air, especially soot and carbons, are trapped and retained by successive impingements on coated surfaces. The arrangements of filtering mediums and the kind of materials used are almost unlimited. Since this type of device depends on impingement, it is more effective in catching large particles than small ones. To secure maximum cleaning efficiency, it is necessary to divide the air into innumerable fine streams, to obtain intimate contact between the air and the viscous-coated mediums.

The following are desirable characteristics of a binding liquid:

1. Its surface tension should be such as to produce a homogeneous film-like coating on the filter medium.

2. The viscosity should vary only slightly with normal changes of temperature.
3. It should prevent the development of mold spores and bacteria on the filter mediums.
4. The liquid should have high capillarity, or ability to wet and retain the dust at all operating temperatures.
5. Evaporation should be low.
6. It should be fire resistant.
7. It should be odorless.

Viscous Type Unit Filters

In the unit type viscous filter, the filtering mediums are arranged in units of convenient size to facilitate installation, maintenance, and cleaning. Each unit consists of an interchangeable cell or replaceable filter pad and a substantial frame which may be bolted to the frames of other like units to form a partition between the source of dusty air and the fan inlet.

To secure the maximum dust-holding capacity, the filter medium is usually arranged with large pores or air passages on the entering air side, the filter density increasing gradually toward the leaving air side. This arrangement provides relatively large spaces for the collection of dirt in

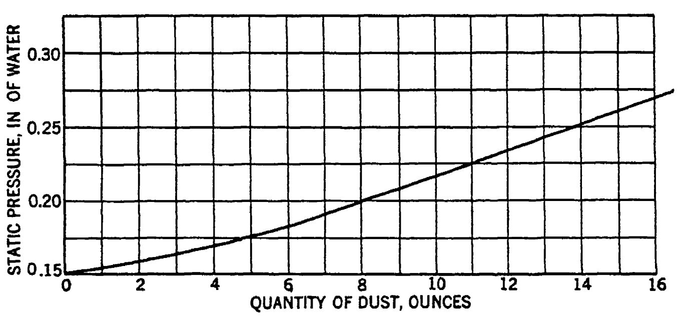


FIG. 1. CHART SHOWING CHANGE IN RESISTANCE DUE TO DUST ACCUMULATION

the front of the filter where the bulk of the dust is taken out without undue increase in resistance, while in the back of the filter the openings are smaller to secure high efficiency in the removal of the finer dust particles.

The resistance of a well-designed unit filter of the adhesive impingement type usually depends upon the velocity at which the air is handled and upon whether the unit is clean or dirty. The cleaning efficiency is usually highest after it has accumulated a certain portion of its maximum load of dirt because some dust collected in the cell acts as an efficient medium for the further seizing of solids from the air. By periodically cleaning a predetermined number of cells, the resistance and capacity of a built-up filter may be held at any desired figure. The frequency of cleaning any installation depends upon the dust concentration of air being cleaned, and on the amount of dirt which can be accumulated in the medium without causing excessive resistance. (Figs. 1, 2 and 3.)

The type of dust being collected has a marked effect on the increase in resistance and the dust holding capacity of a filter. Where appreciable quantities of lint, or other fibrous air impurities are encountered, the

entering surface of a filter may clog very rapidly. For such a condition, a dry type of filter is usually preferable.

A chart showing the increase in resistance of a unit filter of the viscous impingement type, when tested with the standard test dust described in the Code, is given in Fig. 1. The resistance to air flow of three typical clean viscous impingement type filters having different densities of mediums is shown in Fig. 2. Type A is a dense pack used in bacterium control; Type B is a medium pack used for general ventilation work,

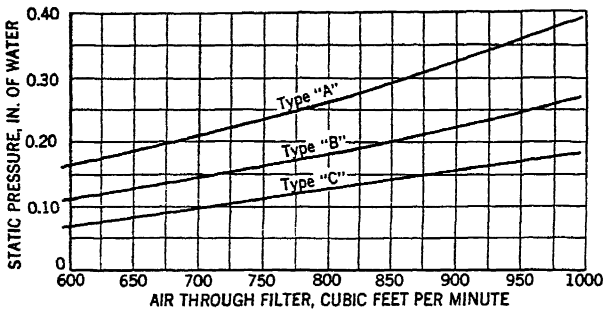


FIG. 2. RESISTANCE TO AIR FLOW OF TYPICAL UNIT AIR FILTERS

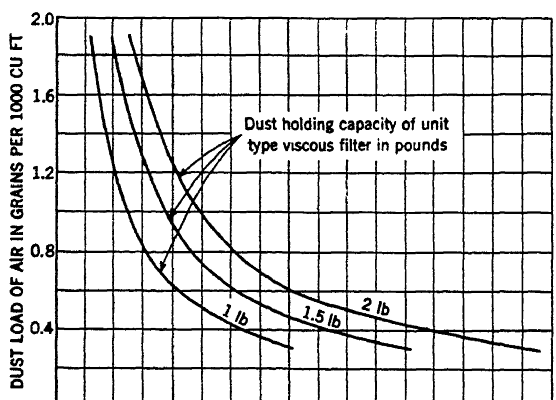


FIG. 3. MAINTENANCE CHART FOR UNIT TYPE VISCOUS FILTERS

and Type C is a low resistance unit for use where low resistance is the important factor and maximum cleaning efficiencies are not essential. The operating characteristics which might be expected under various dust concentrations with air filters having different dust-holding capacities are illustrated in Fig. 3.

Filters consisting of inexpensive frames of cardboard or similar material filled with viscous-coated glass wool, steel wool or the like are available. Because of their construction these units may be discarded when dirty and replaced with new units at relatively little expense. They are used in general ventilation work and with warm-air furnaces and other instal-

lations where low first cost and low resistance to air flow are essential. The operating characteristics of these units conform in general with those of the rigid frame type.

Viscous Automatic Filters

In this type of filter, the removal of the accumulated dust is done automatically instead of by hand. The automatic cleaning and recoating of these filters is based on the principle that the viscous fluid itself will perform the cleaning function, thereby eliminating a separate washing agent. The dust collected by the filter is deposited finally in the bottom of the viscous fluid reservoir from which it may be removed by different methods, depending on the design of the filter.

The filtering medium of the automatic filter is assembled in the form of an endless curtain suspended vertically, with its lower portion submerged in a viscous fluid reservoir. Cleaning and recoating is accomplished by slowly moving the curtain through the bath. The curtain is driven by an electric motor, and its motion may be continuous or intermittent.

The customary resistance to air flow is ⅜-in. water gage at an air velocity of 500 fpm, measured at the filter entrance. Automatic viscous filters are made up in units which are delivered either fully assembled or in parts to be assembled at the point of installation.

DRY AIR FILTERS

Dry air filters, in which dust is impinged upon or trapped in screens made of felt, cloth, cellulose, or other fabrics, are available in various types. These filters require no adhesive liquid, but depend on the straining or screening action of the filtering medium. Because of the close texture of the materials used in most of the dry filters, the surface velocity, or velocity of the entering air, ranges between 10 and 50 fpm, depending on the nature and texture of the fabric. This necessitates a relatively large screen surface, which is usually arranged in the form of pockets to bring the frontal area within customary space requirements.

As with viscous unit filters, an average constant resistance and air volume may be obtained by periodic reconditioning or renewal of the filter screens. Since some materials suitable for dry filtering mediums are affected considerably by moisture which tends to cause a rapid increase in resistance, they should be treated or processed to minimize the effect of changes in humidity.

Filters using felt and similar materials usually depend upon vacuum cleaning for reconditioning. A special nozzle, operated from a portable or stationary vacuum cleaner, is shaped to reach all parts of the filter pockets. Permanent filter mediums should be capable of withstanding repeated vacuum cleanings without loss in dust removal efficiency. While most dry filters are cleaned by replacing an inexpensive filter sheet, the useful life of these sheets often may be lengthened by vibrating or vacuum cleaning.

AIR WASHERS

Air washers, originally designed as the name implies to wash air, are now used for humidification (see Chapter 27). Their ability to cleanse

air depends upon the nature of the dust; fine particles, especially those having no affinity for water, are not efficiently removed.

ELECTRICAL PRECIPITATORS

Electrostatic precipitation has long been used for the precipitation of smokes and fumes from smoke stacks, but until recently such equipment was unsuited to air conditioning and ventilation applications. The operating principles of this device are shown schematically in Fig. 4. A discharge takes place from a small wire to grounded electrodes and as the air passes through this discharge the dust particles receive an electrical charge. The air then passes between parallel plates with a high potential difference between plates. The resulting electric field attracts the dust particles to one set of plates.

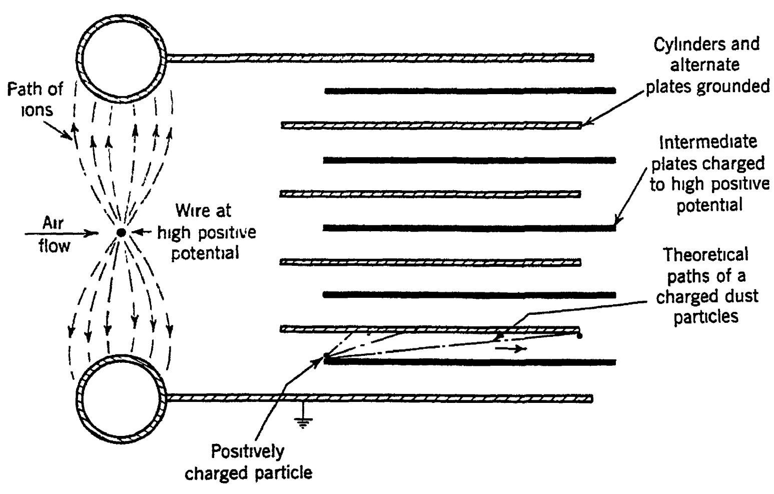


FIG. 4. DIAGRAMMATIC CROSS-SECTION OF ELECTROSTATIC PRECIPITATOR

The electric force is effective in depositing the particle on the plate, but does not hold it there. Some dusts stick to the plates, but in general there should be a coating of adhesive. The dust is removed by washing. As this device is sensitive to air velocities, care should be taken to avoid any local currents that exceed rated velocity. An advantage of electrostatic precipitation is the ability to remove fine particles at high efficiency and with a low pressure drop, and thus with little change in the volume of air delivered. This feature must be balanced against the greater first cost.

CLEANING OF GASES FROM EXHAUST SYSTEMS

The gases from exhaust systems vary widely, but in general contain heavy concentrations of dust as compared to ventilation practice. A dust loading of one grain per cubic foot of air is not unusual, which is about 1000 times the dirt usually found in ventilating air. For this

reason even though the cleaning efficiency is high it is still usually desirable to discharge the exhaust out of doors. The purpose of the cleaning is then to prevent a nuisance in the neighborhood or to collect valuable material.

Gravitational Settling Chambers

Settling chambers are simple but effective only in removing large particles. The rate of fall of various particles is given in Fig. 1 of Chapter 28. Even though the vertical height is kept small to reduce the time of fall, limitations of space and volume of air to be handled usually allow time for only the larger particles to settle, even if the flow is perfectly streamlined. Actually, eddies retard the settling of particles so that the full velocity of fall is not realized.

Cyclones or Centrifugal Separators

The force causing settling can be increased many times that of gravitation by giving the air a whirling motion and introducing centrifugal force. The settling rate then becomes dependent upon the peripheral air velocity and the radius of curvature as well as upon the other factors.

In cyclone separators, air is introduced tangentially into a vertical cylinder and passes out from the center of the top. The gas velocity and curvature of the cylinder cause whirling which throws the particles to the surface. The particles slide down this surface and are removed through a hopper in the cone bottom, or are thrown through slits in the periphery.

Assumptions regarding streamline flow and turbulence make general calculations of centrifugal settling rate quite involved and rough. Centrifugal separators have wide application in connection with industrial operations such as grinding, screening and combustion, but have little or no effect upon the finer particles.

When a high collection efficiency is desired, or the material is unusually fine, multicyclones may be used. These are merely small cyclones arranged in parallel which utilize the principle of high centrifugal velocity to attain separation.

Cloth Filters

Filters are used when the material collected by an exhaust system is valuable or cannot be separated efficiently from the air with an ordinary cyclone. They are also employed when it is desirable to recirculate the air drawn from a room by the exhaust system, which otherwise might entail considerable loss in heat. Bag filters which are properly housed may be operated under suction. Bag houses used in manufacture of zinc oxide and other chemical products are operated on the positive side of the fan.

Wool, cotton and asbestos cloths are commonly used as filtering mediums. When woolen cloths are employed, the filtering capacities vary from ½ to 10 cfm per square foot of filtering surface, depending on the character of the material collected. The rates for cotton and asbestos cloths are lower. The type of filter cloth and the rates of filtration depend, of course, on the material to be collected and the fan capacity.

The collected dust particles themselves aid in agglomerating and retaining others. Periodic shaking with the fans off or reversed, at intervals of a few hours, drops the excess dust into a lower header or hopper for removal.

Readily removed filters built in small sections in which filter mediums can be replaced are of distinct advantage where deterioration is rapid. Various styles of construction are available which combine quick interchangeability and large filtering area per square foot cross-sectional area. Use of several independent units in parallel is important for the reconditioning of each unit separately. Both continuous and intermittent shaking and sweeping devices remove excess dust and maintain a low resistance.

Such filters should be frequently inspected for leaks and the bags or screens should be in readily replaceable units. General practice in this field is to use a low flow per square foot of cloth and a high pressure drop, giving a relatively high efficiency as compared to general ventilation practice. However, because of the high dust loading, the discharged air may still carry excessive dust for ventilating purposes.

SMOKE STACKS

Gases discharged from smoke stacks may contain relatively large particles of cinders and carbon as well as the fine smoke which remains in relatively permanent suspension. Large particles settle quickly and are likely to constitute a serious nuisance in the immediate neighborhood. Smoke tends to spread over a large area, but in cities the large number of stacks may pollute the atmosphere over the entire city.

A variety of cinder catchers is available for catching the large cinders. Devices of the cyclone type remove much finer particles, but for anything approaching complete cleaning, electrostatic precipitators are usually required. For this purpose they usually consist of wires (at potential of 30,000 to 100,000 volts negative) suspended between vertical plates or hanging in the center of vertical cylinders. The dust collects on the plates or cylinders which are periodically vibrated or rapped to cause the dust to fall down into hoppers.

ODOR ADSORPTION

In recent years adsorption has been applied to ventilation for the removal of odors and also of gases or vapors which might be harmful or undesirable. A suitable adsorbing medium known for application to ventilation systems is cocoanut shell activated carbon. It is a hard granular very porous substance which has the property of selectively adsorbing gases. It adsorbs, in most cases, 60 per cent or more of its own weight and retains more than half of this amount. At ordinary temperatures none of the gases which are normal ingredients of the atmosphere are adsorbed. With very few exceptions, nearly all other gases are adsorbed. Two exceptions are carbon monoxide and ammonia. For water vapor, which is readily adsorbed, the carbon has no retentivity, and therefore the water is displaced by other gases when they are adsorbed. It may be used to remove such odors or gases as food odors, sweat and

body odors, putrefaction and sewage odors, animal odors, perfumes, alchohols, hydrocarbon, solvents, hydrogen sulphide and sulphur dioxide.

The apparatus consists in principle of relatively thin granular beds through which the air stream passes. Upon contact with the grains of carbon, the odors and other gases are adsorbed and held within the structure of the activated carbon. When saturated, the carbon may be revivified without loss. The period between revivification will vary with the concentration of adsorbable gases in the air stream, and have been found in ordinary odor conditions in ventilation practice to vary from four years down to four months.

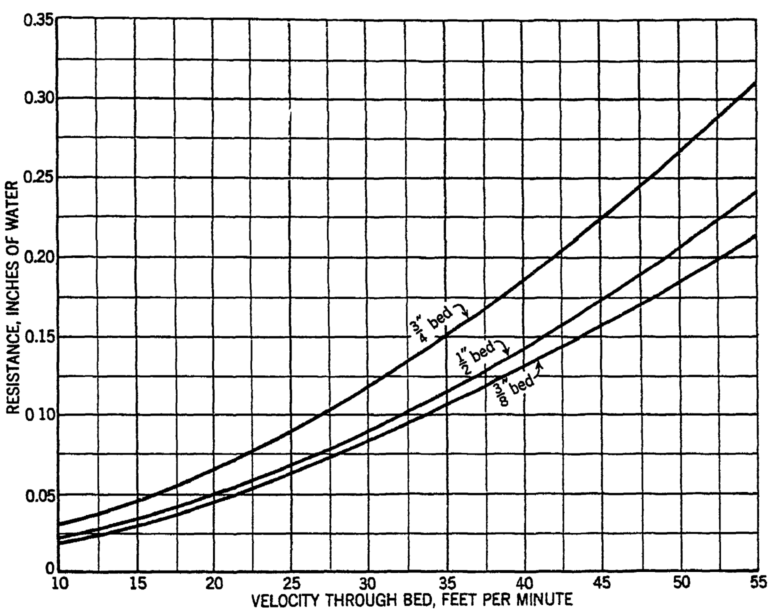


Fig. 5. Resistance Curves for 6-14 Mesh Activated Carbon

The resistance curves shown in Fig. 5 are for 6-14 mesh activated carbon used in conventional equipment. Resistances given are overall for customary arranged equipment. The average practical velocity for such equipment is 40 fpm, although common velocities may range from 30 to 50 fpm.

REFERENCES

Design and Application of Oil-Coated Air Filters, by H. C. Murphy (A.S.H.V.E. Transactions, Vol. 33, 1927, p. 73).

Operation and Maintenance of Air Filters, by W. G. Frank (*Heating, Piping and Air Conditioning*, May, 1931, p. 378).

Size and Characteristics of Air-Borne Impurities, by W. G. Frank (*Heating, Piping and Air Conditioning*, January, 1932, p. 35).

Fundamental Principles in the Design of Dry Air Filters, by Otto Wechsberg (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, April, 1933, p. 217).

Operation, Maintenance of Cloth-Screened Dust Collectors, by W. F. Terry (*Heating, Piping and Air Conditioning*, May, p. 259, June, p. 304, 1933).

Testing and Rating of Air Cleaning Devices Used in General Ventilation Work, by Samuel R. Lewis (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 277).

A.S.H.V.E. RESEARCH REPORT No. 1094—Air Filter Performance as Affected by Kind of Dust, Rate of Dust Feed, and Air Velocity Through Filter, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 415).

A.S.H.V.E. RESEARCH REPORT No. 1122—Air Filter Performance as Affected by Low Rate of Dust Feed, Various Types of Carbon, and Dust Particle Size and Density, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 339).

A.S.H.V.E. RESEARCH REPORT No. 1145—The Effect of Lint on Air Filter Performance, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940).

A.S.H.V.E. RESEARCH PAPER—Comparison of the Weight, Particle Count and Discoloration Methods of Testing Air Filters, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, April, 1941, p. 246).

A.S.H.V.E. RESEARCH PAPER—Economical Air Velocities for Mechanical Air Filtration, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, August, 1941, p. 524).

The Dust-Free Space Surrounding Hot Bodies, by H. H. Watson (*Transactions of the Faraday Society*, Vol. 32, 1936, Pt. 2, p. 1073).

A New Electrostatic Precipitator, by G. W. Penney (*Electrical Engineering*, January, 1937, p. 159).

Chapter 30

FANS

Classification, Performance, Fan Efficiency, Characteristic Curves, System Characteristics, Selection of Fans, Fan Designations, Control, Motive Power

IN heating and ventilating practice, fans are used to produce air flow except where positive displacement is required, in which case compressors or rotary blowers are used. All fans or blowers are classified according to the direction of air flow through the fan with relation to the axis of rotation and are either of the (1) *axial flow* or *propeller* type, in which the flow is parallel to the axis or (2) *radial flow* or *centrifugal* type, in which the flow is parallel to the radius of rotation.

Axial flow fans are made with various numbers of blades, the latter varying widely in form. The blades may be of uniform thickness and made of cast or sheet metal, and either flat or cambered or of screw form; or they may vary in thickness, in the latter case usually being designed to conform to so-called *airfoil* sections of known characteristics, similar to those which have been developed for airplane propellers. Likewise, blade angle, or the angular relation of the blades to the plane of rotation, varies over a wide range. For operation against comparatively high pressures, it is customary to resort to enlarged hubs in proportion to fan diameter (large hub ratio) and correspondingly short blade length. The term *disc fan* has sometimes been loosely applied to such large hub fans, though it has long been generally used in connection with any propeller fan of comparatively short axial length whose blades are relatively flat; in other words, for fan wheels which occupy a space which is more or less disc-shaped.

Radial flow or *centrifugal fans* include steel plate fans, pressure blowers, cone fans, and the so-called multiblade fans. All the foregoing types have variations which may be obtained by modification of the proportions or change in the curvature and angularity of the blades. The angularity of the blades determines the operating characteristics of a fan; a forward curved blade is found in a fan having slow speed operating characteristics, while a backward curved blade is found in a fan having high speed operating characteristics.

A wide variation exists in the demands which have to be met by fan installations. A fan may be required to move large quantities of air against little or no resistance or it may be required to move small quantities against high resistances. Between these two extremes innumerable specific requirements must be met. In general, fans of all types in each

general class can be made to perform the same duty, although mechanical difficulties, noise or lack of efficiency may limit the use to one or another type. Until a few years ago, the most common field of service for fans of the propeller type was in moving air against moderate pressures, where no long ducts were involved, or no heavy frictional resistance had to be overcome. However, recent developments in the design of axial flow fans based on the application of aero-dynamic principles, and furthermore, the use of multi-stage fans, have greatly increased the range of pressures against which the modern propeller fan can be applied. Single stage axial flow fans of moderate diameter are now available to operate against static pressures of 3 and 4 in. of water, while maintaining moderate noise levels. These pressures are readily doubled by the simple device of double staging. In multi-stage units, intermediate guide vanes are employed to properly redirect the air discharged by the first stage into the second stage wheel. In the most common form of two-stage axial flow fan, the motor is provided with shaft extensions on each end, one carrying the first stage fan, and the other the second stage. The motor is supported radially from a cylindrical casing and the intermediate guide vanes are mounted in the annular space around the motor.

FAN PERFORMANCE

Fans of all types follow certain laws of performance which are useful in determining the effect of changes in the conditions of operation. These laws apply to installations comprising any type of fan, any given piping system and constant air density, and are as follows:

1. The air capacity varies directly as the fan speed.
2. The pressure (static, velocity, and total) varies as the square of the fan speed.
3. The power demand varies as the cube of the fan speed.

Example 1. A certain fan delivers 12,000 cfm at a static pressure of 1 in. of water when operating at a speed of 400 rpm and requires an input of 4 hp. If in the same installation 15,000 cfm are desired, what will be the speed, static pressure, and power?

$$\text{Speed} = 400 \times \frac{15{,}000}{12{,}000} = 500 \text{ rpm}$$

$$\text{Static pressure} = 1 \times \left(\frac{500}{400}\right)^2 = 1.56 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{500}{400}\right)^3 = 7.81 \text{ hp}$$

When the density of the air varies the following laws apply:

4. At constant speed and capacity the pressure and power vary directly as the density.

Example 2. A certain fan delivers 12,000 cfm at 70 F and normal barometric pressure (density 0.075 lb per cubic foot) at a static pressure of 1 in. of water when operating at 400 rpm, and requires 4 hp. If the air temperature is increased to 200 F (density 0.0602 lb) and the speed of the fan remains the same, what will be the static pressure and power? .

$$\text{Static pressure} = 1 \times \frac{0.0602}{0.075} = 0.80 \text{ in.}$$

$$\text{Power} = 4 \times \frac{0.0602}{0.075} = 3.20 \text{ hp}$$

5. At constant pressure the speed, capacity and power vary inversely as the square root of the density.

Example 3. If the speed of the fan of Example 2 is increased so as to produce a static pressure of 1 in. of water at the 200 F temperature, what will be the speed, capacity, and power?

$$\text{Speed} = 400 \times \sqrt{\frac{0.075}{0.0602}} = 446 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \sqrt{\frac{0.075}{0.0602}} = 13{,}392 \text{ cfm (measured at 200 F)}$$

$$\text{Power} = 4 \times \sqrt{\frac{0.075}{0.0602}} = 4.46 \text{ hp}$$

6. For a constant weight of air:

(*a*) The speed, capacity, and pressure vary inversely as the density.

(*b*) The horsepower varies inversely as the square of the density.

Example 4. If the speed of the fan of the previous examples is increased so as to deliver the same weight of air at 200 F as at 70 F, what will be the speed, capacity, static pressure, and power?

$$\text{Speed} = 400 \times \frac{0.075}{0.0602} = 498 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \frac{0.075}{0.0602} = 14{,}945 \text{ cfm (measured at 200 F)}$$

$$\text{Static pressure} = 1 \times \frac{0.075}{0.0602} = 1.25 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{0.075}{0.0602}\right)^2 = 6.20 \text{ hp}$$

FAN EFFICIENCY

The efficiency of a fan may be defined as the ratio of the horsepower output to the horsepower input.

The horsepower output is expressed by the formula:

$$\text{Air Horsepower}^1 = \frac{\text{cfm} \times \text{total pressure in inches of water}}{6356} \tag{1}$$

When the static pressure is used in the computation in place of total pressure it is assumed that this represents the useful pressure and that the velocity pressure is lost in the piping system and in the air which leaves the system. Since in most installations a higher velocity exists at the fan outlet than at the point of delivery into the atmosphere, some of the velocity pressure at the fan outlet may be utilized by conversion to static pressure within the system, but, owing to the uncertainty of friction losses which occur at the places where changes in velocity take place, the amount of velocity pressure which is actually utilized is seldom known, and the static pressure alone may best represent the useful pressure. In the standards for published capacity tables as adopted by the *National Association of Fan Manufacturers*, the term *static pressure* refers to the

[1]See Standard Test Code for Centrifugal and Axial Fans, Third Edition of 1938.

true resistance to air flow. Such tables charge both the inlet and outlet velocity of the fan, to the fan performance, and may be used directly where the static pressure of the system as calculated represents only the actual resistance to flow of the air.

The efficiency based upon static pressure is known as the static efficiency and may be expressed as follows:

$$\text{Static efficiency}^{1} = \frac{\text{cfm} \times \text{static pressure in inches of water}}{6356 \times \text{Horsepower input}} \qquad (2)$$

Different fans may develop the same capacity against the same static pressure and with the same power input, and therefore operate at the same static efficiency, while maintaining different outlet velocities. Where a high outlet velocity is desirable or can be utilized effectively, the static efficiency fails to be a satisfactory measurement of the performance. In many applications of propeller fans, air is circulated without encountering resistance and no static pressure is developed. The static efficiency is zero and its calculation is meaningless. Because of such situations where the static efficiency fails to indicate the true performance, many engineers prefer to base the calculation of efficiency upon the total pressure. This efficiency is variously known as the total, or mechanical efficiency, and may be expressed as follows:

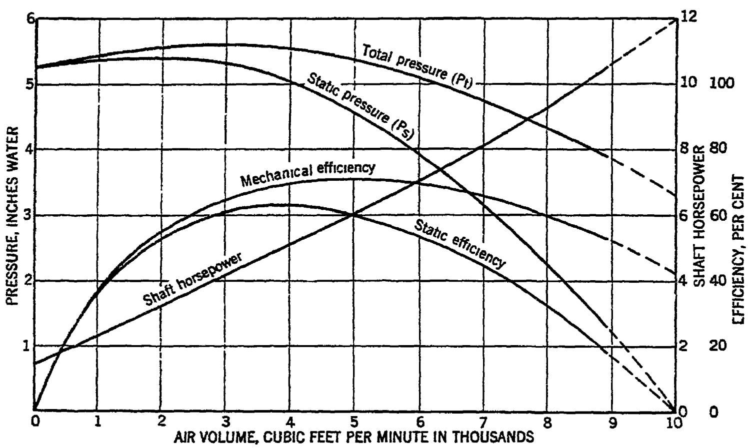


FIG. 1. TYPICAL FAN PERFORMANCE CURVE

$$\text{Mechanical or Total efficiency}^{1} = \frac{\text{cfm} \times \text{total pressure in inches of water}}{6356 \times \text{Horsepower input}} \qquad (3)$$

CHARACTERISTIC CURVES

In the operation of a fan at a fixed speed the static and total efficiencies vary with any change in the resistance which is imposed. With different designs the peak of efficiency occurs when the fans deliver different percentages of their wide-open capacity. Variations in efficiency accompany

variations in pressures and power consumption which are characteristic of the individual designs and which are influenced particularly by the shape and angularity of the blades. Such variations in pressure, power, and efficiency are shown by characteristic curves.

Characteristic curves of fans are determined by tests performed in accordance with the Standard Test Code for Centrifugal and Axial Fans[2] prepared jointly by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers.* The results of tests are plotted in different ways: the abscissae may be the ratio of delivery, assuming full open discharge as 100 per cent, and the ordinates may be static pressure, total pressure, horsepower and efficiency. A typical fan performance curve is shown in Fig. 1.

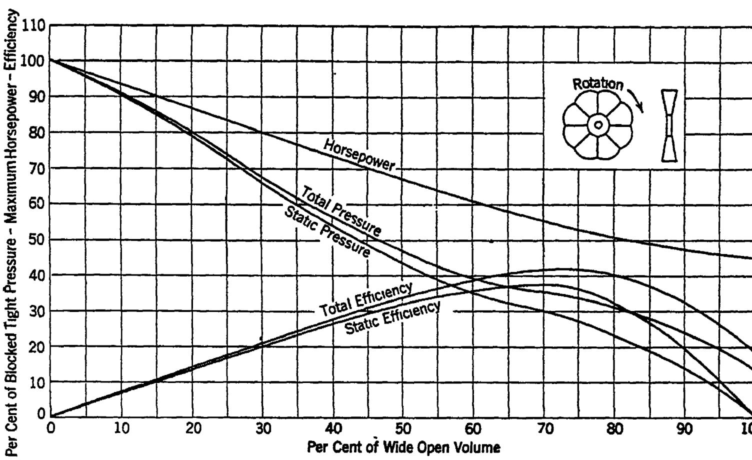


FIG. 2. OPERATING CHARACTERISTICS OF AN AXIAL FLOW FAN WITH BLADES OF UNIFORM THICKNESS

In the selection of all but very small fans, power consumption is usually a major consideration. It must be borne in mind that the horsepower at peak efficiency alone may be misleading, as actual operation is apt to occur at some point on the pressure-volume curve varying considerably from that specified, due to inaccuracies of the estimated system resistance or to fluctuating resistance caused by damper or louver adjustments. To cope with such variations a fan should be selected having a high efficiency over a wide range, that is, a *flat* or broad efficiency curve is more desirable than a sharp or narrow curve which, though reaching a high peak, falls off rapidly to either side of a narrow range. When the point of operation varies only within narrow limits and both volume and pressure requirements are accurately known in advance, the designer can select a fan operating at maximum efficiency, irrespective of performance over the entire range.

Generally fans are selected either at the peak of the static efficiency or to the right of the peak depending on the requirements of the particular

[2]A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 407. Amended in A.S.H.V.E. TRANSACTIONS, Vol. 37 1931, p. 363. Third Edition of 1938.

selection. Fans selected to the right of the peak will be smaller but will require more power, run at higher speeds and have a higher sound rating. Where first cost is important and added horsepower and noise are not important, smaller fans may be used. Where efficient and quiet operation are most important, fans are selected at or near the peak of the static efficiency curve. Fans are not ordinarily selected to the left of the peak of the static efficiency curve as this results in larger, more costly fans, requiring more power and in some cases producing objectionable noise.

The curves shown in Figs. 2, 3, 4 and 5 show operating characteristics for various types including the backwardly inclined blade design for comparison purposes. These curves are not applicable for rigid comparison or actual selection and are shown to indicate the variations in operating characteristics.

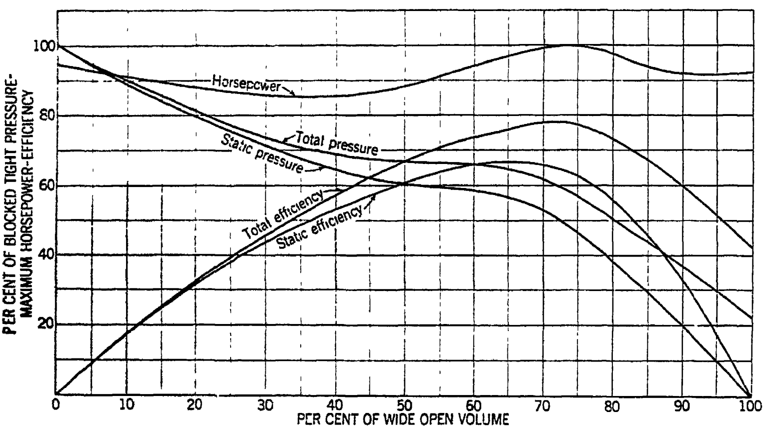


FIG. 3. OPERATING CHARACTERISTICS OF AXIAL FLOW AIRFOIL TYPE FAN

Axial flow fans having blades of *uniform thickness* at any given radius are characterized by rapid rise in power consumed as the resistance increases, as illustrated in Fig. 2. When operating against high resistance, this type of blade permits some of the air to pass back between the blades near the hub, where blade speed is much lower than near the tip or periphery. Obviously, a fan of such characteristics should only be used against low resistance.

The curves in Fig. 3 show the characteristics of typical *axial flow fans* with *airfoil* design. This type of fan shows characteristics of non-overloading horsepower and high efficiency at relatively high static pressure, as contrasted with a fan blade of uniform thickness. These results are obtained by more uniform pressure throughout the blade annulus, so that back flow does not occur until high pressures are reached. This reduction in turbulence also has a tendency to reduce noise. Fans of this type are now available, operating against static pressures up to 3 and 4 in. water, single stage and 6 to 8 in. water, double stage. The capacity and efficiency of axial fans can be improved, particularly when operating against considerable pressure, by the use of either inlet or outlet guide

vanes or both. Generally, the effect of such vanes is to increase the level of the pressure-volume curve, and properly designed vanes on the discharge side of the fan have the advantage of eliminating the rotational component of the air stream, thus quickly restoring uniform axial flow. As high pressures usually require large hubs in proportion to the fan diameter, performance is improved by the use of round-nosed or conical forms mounted coaxially with the direct-connected fan (sometimes partly or wholly enclosing the motor) so as to make the changes in velocity to and from the fan blade annulus as space conditions permit. When axial flow fans are installed in ducts, provisions may also be made to install the driving motor outside the duct, by employing slots in the duct to permit a belt drive from the motor to the fan sheave.

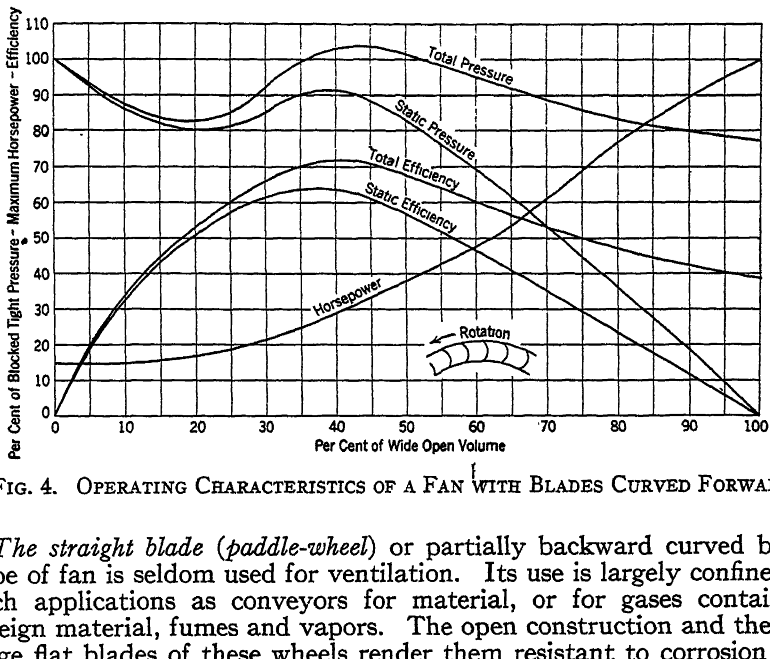


FIG. 4. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED FORWARD

The straight blade (*paddle-wheel*) or partially backward curved blade type of fan is seldom used for ventilation. Its use is largely confined to such applications as conveyors for material, or for gases containing foreign material, fumes and vapors. The open construction and the few large flat blades of these wheels render them resistant to corrosion and tend to prevent material from collecting on the blades. This type of fan has a good efficiency, but the power steadily increases as the static pressure falls off, which requires that the motor be selected with a moderate reserve in power to take care of possible error in calculation of duct resistance.

The *forward curve* multiblade fan and the backward curve types are used extensively in heating and ventilating work. The forward curve type has a low peripheral speed and a large capacity, and is quiet in operation. (See Fig. 4.) The point of maximum efficiency for this fan occurs near the point of maximum static pressure. The static pressure drops consistently from the point of maximum efficiency to full open operation. The power curve rises continually from low to peak capacity and if reasonable care is exercised in calculating resistance, a moderate reserve in power in the motor selection will prevent overloading.

The backward curve types would include the full backward curve blade and the double curve blade having a forward curve heel and a backward curve tip. These types have steep pressure curves and non-overloading power characteristics and relatively high speed. (See Fig. 5.) These fans operate at a peripheral speed approximately 175 per cent of the forward curve multiblade types for like results. Pressure curves begin to drop at very low capacity and continue to fall consistently to full outlet opening. The steep pressure curves tend to produce nearly constant capacity under changing pressures. Where wide fluctuations in demand occur, this type of fan is desirable to prevent overloading of motors. The maximum power requirement occurs at about the maximum efficiency. Consequently a motor selected to carry the load at this point will be of sufficient capacity to drive the fan over its full range of capa-

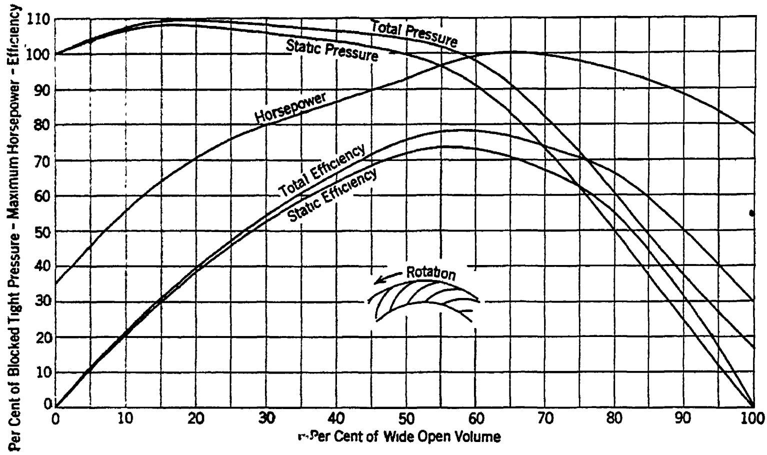


FIG. 5. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED BACKWARD

cities at a given speed. The high speed of this type makes it adaptable for direct connected electric motor drives. The dimensional bulk of this type of fan is usually greater than that of the forward curve multiblade type. The newer designs of these backward curve types have proven to be extremely quiet.

Between the extremes of the forward and backward curve blade type centrifugal fans, a number of modified designs exists, differing in angularity and in the shape of the blades. Characteristic curves of these types show varying degrees of resemblance to the curves of Figs. 4 and 5.

SYSTEM CHARACTERISTICS

Any ventilating system consisting of duct work, heaters, air washers, filters, etc., has a system characteristic which is individual to that system and is independent of any fan which may be applied to the system. This characteristic may be expressed in curve form in exactly the same manner that fan characteristics may be shown. Typical system characteristic

curves are shown as A, B and C in Fig. 6. These curves are drawn to follow the simple parabolic law in which the static pressure or resistance to flow of air varies as the square of the volume flowing through the system. Heating and ventilating systems follow this law very closely and no serious error is introduced by its use.

When a constant speed fan curve for a given size fan is super-imposed upon a system characteristic curve, the relation between the two is at once apparent. The only point common to the two curves is the point at the intersection of the system characteristic curve and the fan characteristic curve, and it is at this point that the combination will operate. In Fig. 6, curves A, B and C cross the fan characteristic curve at points X, Y and Z. This means that when the fan whose curve is shown is applied

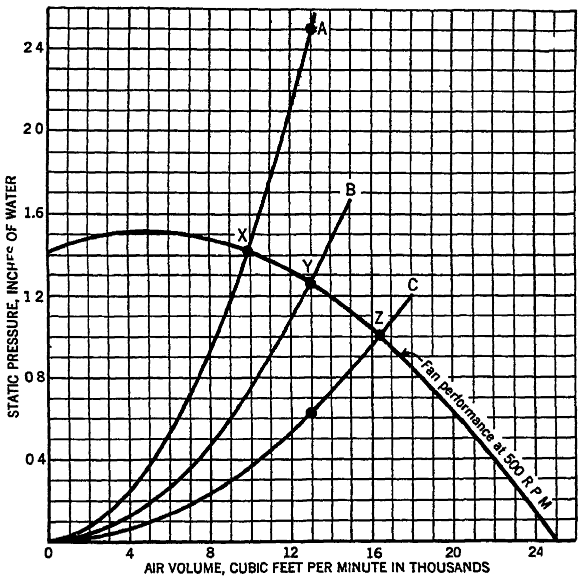


FIG. 6. PARABOLIC SYSTEM CHARACTERISTIC CURVES

to system A, 10,000 cfm will flow through the system. If it is applied to system B, 13,000 cfm will flow, and applied to system C, 16,400 cfm will flow through that system.

The curves in Fig. 6 also illustrate the effect of errors which may be determined by calculating the resistance of a ventilating system. For instance, a given system requires 13,000 cfm and the resistance to flow of the system has been computed as 1.25 in. static pressure. Such a system may be represented by curve B in Fig. 6. Assume that 100 per cent error has been made and the resistance calculated should have been 2.5 in. instead of 1.25 in. Then the system would be as shown in curve A. This new system curve crosses the fan curve at 10,000 cfm. Such an error would result in the flow of air being decreased from a design volume of 13,000 cfm to 10,000 cfm. In case the resistance to flow had been over estimated and instead of 1.25 in. being required, actually the resistance

should have been 0.625 in., this would correspond with a system curve as shown at *C* and on this curve the fan would deliver 16,400 cfm to the system instead of the design volume of 13,000 cfm.

In this example extreme errors have been selected to emphasize the effect the square function of the system characteristic has in maintaining the fan performance within comparatively narrow limits. In the first example a system estimated at half what it should have been, resulted in a drop of 23 per cent in volume; and in the second example, a system estimated at twice what it should have been resulted in an increase of 26 per cent in volume.

In some instances fans may be applied to variable flow systems. In such cases the limiting systems may be plotted and the effect on fan performance examined. For instance, a system might vary between system *A*, shown in Fig. 6 as one limit; and system *B* as the other limit. The fan performance will then fall between points *X* and *Y* on the fan curve at a point determined by the system characteristics at that particular time. If curves *A* and *B* are the limiting systems, the fan performance will never be outside the points *X* or *Y*.

SELECTION OF FANS

The following information is required to select the proper type of fan:

1. Cubic feet of air per minute to be moved.
2. Static pressure required to move the air through the system.
3. Type of motive power available.
4. Whether fans are to operate singly or in parallel on any one duct.
5. What degree of noise is permissible.
6. Nature of the load, such as variable air quantities or pressures.

In order to facilitate the choice of apparatus, the various fan manufacturers supply fan tables or curves which usually show the following factors for each size of fan operating against a wide range of static pressures: (1) volume of air in cubic feet per minute (68 F, 50 per cent relative humidity, 0.075 lb per cubic foot, (2) outlet velocity, (3) revolutions per minute, (4) brake horsepower, (5) tip or peripheral speed, and (6) static pressure. The most efficient operating point of the fan is usually shown by either bold-face or italicized figures in the capacity tables.

Other important factors to be considered in selecting fans are: (1) efficiency, (2) space occupied, (3) sound emission, (4) first cost, and (5) speed (both peripheral and revolutions per minute). These factors are not necessarily shown in the order of importance. In some installations space occupied may be of first importance. In others lowest power consumption is desirable. In many cases quietness of operation of the entire system is essential. Practically all fans operate at their lowest sound level when selected at or near the peak of the static efficiency so that in selecting a fan for highest static efficiency the quietest operating range of the fan will also be obtained. Tables 1 and 2 show desirable outlet velocities and tip speeds, or peripheral velocities, for various static pressures. Fans selected accordingly will operate at or near the peak of the static efficiency with resulting low power consumption and noise

TABLE 1. GOOD OPERATING VELOCITIES AND TIP SPEEDS FOR FORWARD CURVED MULTIBLADE VENTILATING FANS

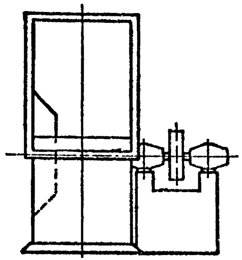
STATIC PRESSURE INCHES OF WATER	OUTLET VELOCITY FEET PER MINUTE	TIP SPEED FEET PER MINUTE
1/4	1000–1100	1520–1700
3/8	1000–1100	1760–1900
1/2	1000–1200	1970–2150
5/8	1200–1400	2225–2450
3/4	1300–1500	2480–2700
7/8	1400–1700	2660–2910
1	1500–1800	2820–3120
1 1/4	1600–1900	3162–3450
1 1/2	1800–2100	3480–3810
1 3/4	1900–2200	3760–4205
2	2000–2400	4000–4500
2 1/4	2200–2600	4250–4740
2 1/2	2300–2600	4475–4970
3	2500–2800	4900–5365

levels. Smaller fans with higher outlet velocities may be used if the installation requirements are such as to warrant the additional power and increased sound level. When space for duct expansion from a fan outlet is not available there may be advantages in selecting a larger fan for reducing duct noises, although lower outlet velocities generally results in lower fan efficiencies which cannot always be justified on the basis of increased cost and space requirements.

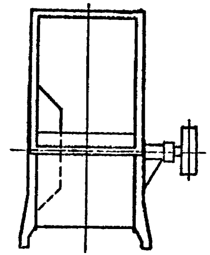
Having selected a fan for its quietest operating point consistent with the requirements of the installation, it must be recognized that ventilating fans, even so selected, emit noise and precautions must be taken in the installation of the fans to prevent this noise from being transmitted to occupied portions of the building. Fans operating against high static pressures produce more noise than fans operating against low static pressures. Consequently, from a noise standpoint, the system should be designed to operate against the lowest static pressure possible. In many modern air conditioning systems it is necessary to introduce devices into the air stream for conditioning the air in various ways, the result of which

TABLE 2. GOOD OPERATING VELOCITIES AND TIP SPEEDS FOR MULTIBLADE VENTILATING FANS WITH BACKWARD TIPPED AND DOUBLE CURVED BLADES

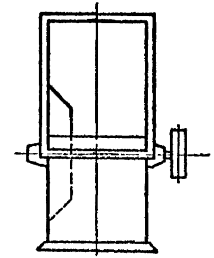
STATIC PRESSURE INCHES OF WATER	OUTLET VELOCITY FEET PER MINUTE	TIP SPEED FEET PER MINUTE
1/4	800–1100	2600–3100
3/8	800–1150	3000–3500
1/2	900–1300	3400–4000
5/8	1000–1500	3800–4500
3/4	1100–1650	4200–5000
7/8	1200–1750	4500–5300
1	1200–1900	4800–5750
1 1/4	1300–2100	5300–6350
1 1/2	1400–2300	5750–6950
1 3/4	1500–2500	6200–7550
2	1600–2700	6650–8050
2 1/4	1700–2800	7050–8550
2 1/2	1800–2950	7450–9000
3	2000–3200	8200–9850



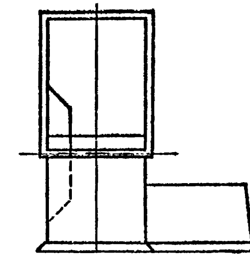
Arr. 1.
For belt drive.
Wheel overhung. Bearings on pedestal



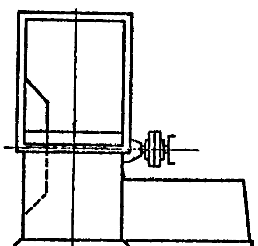
Arr. 2.
For belt drive.
Pulley and wheel overhung. Bearings in bracket on fan housing. Made only in smaller sizes for reversible discharge.



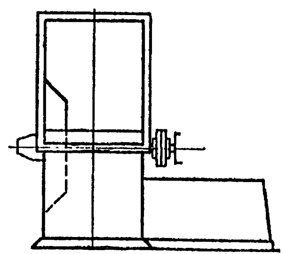
Arr. 3.
For belt drive.
Pulley overhung. Bearings supported on fan housing.



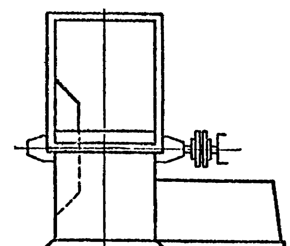
Arr. 4.
For direct drive.
Wheel overhung. No bearings on fan. Wheel mounted on motor or engine shaft. Pedestal for motor or engine.



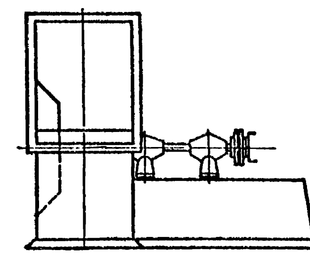
Arr. 5.
For direct drive.
Wheel overhung. Includes housing, wheel, shaft, one intermediate bearing, flanged coupling and pedestal only for motor or engine.



Arr. 6.
For direct drive.
Three-bearing arrangement with fan bearing at inlet side. Includes housing, wheel, shaft, one bearing (in inlet), rigid coupling, and pedestal only for motor or engine.



Arr. 7.
For direct drive.
Similar to Arr. 6, but with two bearings on fan, and flexible instead of rigid coupling.



Arr. 8.
For direct drive.
Similar to Arr. 5, but with two bearings on pedestal with motor, and flexible instead of rigid coupling.

FIG. 7. ARRANGEMENT OF FAN DRIVES

is to set up a rather high static pressure against which the fan must operate. In such cases the sound level at the fan may be too high to be neglected and special sound treatment of the installation must be considered. When a fan is operating against higher pressures it should be located in a room either removed from the occupied areas, or in a room which has been acoustically treated to prevent sound being carried through the walls to adjoining spaces. The fan should be mounted on a resilient base along with its driving motor to absorb any noise or vibration which might be transmitted to the floor and thence to the building structure. All ducts should be connected to fans with unpainted canvas, or other flexible material, to prevent any vibrations being transmitted to the duct work. Ducts leading into the fan room or from the fan, should be acoustically treated on the interior and in special cases, should be provided with sound traps or filters. Many ventilating systems encounter noises which are connected with the fan in no way. Noises due to high duct velocities, abrupt turns, grilles, etc., may be present. Treatment of such problems is covered in Chapter 33.

FAN DESIGNATIONS

Facing the driving side of the fan, blower, or blast wheel, if the proper direction of rotation is clockwise, the fan, blower, or blast wheel will be designated as *clockwise*. If the proper direction of rotation is counter-clockwise, the designation will be *counter-clockwise*. (The driving side of a single inlet fan is considered to be the side opposite the inlet regardless of the actual location of the drive.)[3]

This method of designation will apply to all centrifugal fans, single or double width, and single or double inlet. Do not use the word *hand* but specify *clockwise* or *counter-clockwise*.

The discharge of a fan will be determined by the direction of the line of air discharge and its relation to the fan shaft, as follows:

Bottom horizontal: If the line of air discharge is horizontal and below the shaft.

Top horizontal: If the line of air discharge is horizontal and above the shaft.

Up blast: If the line of air discharge is vertically up.

Down blast: If the line of air discharge is vertically down.

All intermediate discharges will be indicated as angular discharge as follows:

Either top or bottom angular up discharge or top or bottom angular down discharge, the smallest angle made by the line of air discharge with the horizontal being specified.

In order to prevent misunderstandings, which cause delays and losses, the arrangements of fan drives adopted by the *National Association of Fan Manufacturers* and indicated in Fig. 7 are suggested.

If double width, double inlet fans are selected, care must be taken that both inlets have the same free area. If one inlet of a fan is obstructed more than the other, the fan will not operate properly, as one half of the wheel will deliver more air than the other half. The *backward curved* and *double curved* types with backward tip operate satisfactorily in double or in parallel operation.

FAN CONTROL

In some heating and ventilating systems it is desirable to vary the volume of air handled by the fan, which may be accomplished by a number of methods. Where the change is made infrequently, the pulley or sheave

[3]Recommendations adopted by the *National Association of Fan Manufacturers*.

on the driving motor, or fan, may be changed to vary the speed of the fan thus altering the air volume. Dampers may be placed in the duct system to vary the volume. Variable speed pulleys or transmissions, such as fan belt change boxes or hydraulic couplings, may be used to vary the fan speed. Variable speed motors and variable fan inlet vanes may also be used to adjust the fan volume. All of these methods will give control. From a power consumption standpoint, a reduction of the fan speed is most efficient. Inlet vanes save some power and dampers save the least. From the standpoint of first cost, dampers usually are the lowest in cost. In some installations adjustments of volume are desirable at various times during the day or continuously. In others an increased supply of air in summer over that needed in winter is demanded. The demands of each case will dictate which type of control is most desirable. Where noise is a factor, lowering the fan speed if possible is preferred as a control means, because of the resulting reduction in sound level.

MOTIVE POWER

Heating and ventilating fans are usually driven by electric motors, although they may be driven by gasoline or oil engines, steam engines or turbines. Fans may be direct-connected to the operating unit, but it is the usual practice to use belt driven fans for large units.

In selecting the size motor to be used, it is general practice to provide a rather liberal allowance over the actual fan power required when fan has a rising horsepower characteristic. Actual static pressures may vary from those estimated and if less than estimated, the fan may deliver more air than required and take more power. Justification for liberal power provision exists also in the possibility of varying demand, due to change in ventilation requirement, intensity of occupation and weather conditions. The degree of allowance may vary with fan types due to their inherent characteristics. The backward curved blade type fan requires maximum power at or near the peak of the efficiency, hence this fan would require less allowance in driving power than other types not having this characteristic. Reference to Fig. 5 indicates that there is no justification for allowing large spare motor capacity, and it is generally more economical to operate motors well loaded.

REFERENCES

Mine Ventilation, by J. J. Walsh (A.S.H.V.E. Transactions, Vol. 23, 1917, p. 659).

Fan Blower Design, by H. F. Hagen (A.S.H.V.E. Transactions, Vol. 28, 1922, p. 175).

The Specific Characteristics of Fans, by M. C. Stuart and J. B. Lusk (A.S.H.V.E Journal Section, *Heating, Piping and Air Conditioning*, September, 1936, p. 507).

Non-Dimensional Fan Characteristics, by H. Carlton Moore (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, September, 1937, p. 580).

Constructive Mechanism and the Centrifugal Fan, by George D. Beals.

Fans, by Theodore Baumeister, Jr.

Fan Engineering, Buffalo Forge Company.

Heating, Ventilating and Air Conditioning, by Harding and Willard, Revised Edition, 1932.

The Centrifugal Fan, by Frank L. Busey.

The Fan, by Charles H. Innes.

The Theory and Performance of Axial-Flow Fans, by L. S. Marks and J. R. Weske.

Chapter 31

AIR DISTRIBUTION

Definitions, Grille Locations, Standards for Satisfactory Conditions, Factors Affecting Distribution for Cooling and Heating, Air Outlet Noises, Selection of Supply Outlets, Balancing System

CORRECT air distribution contributes as much or more to the success of a forced air heating, ventilating, cooling or air conditioning system as does any other single factor. Supplying the proper amount of air is one problem; properly distributing it from the point where it leaves the fan is another. The distribution problem may be further divided into: (*a*) distribution *to* the various spaces served by the system, (*b*) distribution *in* these spaces. This discussion is primarily limited to division (*b*), reference being made to the duct system only insofar as it affects the performance of the air distribution outlets.

Definitions

1. *Supply Opening or Outlet:* Any opening through which air is delivered into a space which is being heated, or cooled, or humidified, or dehumidified, or ventilated.

2. *Exhaust Opening:* Any opening through which air is removed from a space which is being heated, or cooled, or humidified, or dehumidified, or ventilated.

3. *Outside Air Opening:* Any opening used as an entry for air from outdoors.

4. *Grille:* A covering for any opening and through which air passes.

5. *Damper:* A device used to vary the volume of air passing through a confined cross-section by varying the cross-sectional area.

6. *Multiple Louver Damper:* A damper having a number of adjustable blades.

7. *Single Louver Damper:* A damper having one adjustable blade.

8. *Face:* A grille with provision for attaching a damper.

9. *Register:* A face with a damper attached.

10. *Flange:* The portion (either integral or separate) of a grille, face, or register extending into the duct opening for the purpose of mounting.

11. *Frame:* The portion (either integral or separate) of a grille, face, or register extending around the duct opening for the purpose of mounting.

12. *Margin:* The margin of a grille, face, or register is one-half of the difference between the duct dimension and overall dimension measured either horizontally or vertically.

13. *Fret:* The member separating the openings of a grille, face, or register.

14. *Free Area:* The total minimum area of the openings in the grille, face, or register through which air can pass.

15. *Core Area:* The total plane area of the portion of a grille, face, or register bounded by a line tangent to the outer edges of the outer openings through which air can pass.

16. *Mean Area:* The total of the core and free areas divided by two.

17. *Duct Area:* The area of a cross-section of the duct based on the inside dimensions at the point where the grille, face or register is mounted.

18. *Percentage Free Area:* The ratio of the free area to the core area expressed in percentage.

19. *Aspect Ratio:* The ratio of length of the core of a grille, face or register to the width.

20. *Throw:* The distance air will carry measured along the axis of an air stream from the supply opening to the position in the stream at which air motion reduces to 50 fpm.

21. *Envelope:* The outer boundary of an air stream.

GRILLE LOCATIONS

The location of supply and exhaust openings is extremely important if a satisfactory installation is to be secured. Very frequently, however, the room or building is planned and constructed with practically no consideration of this problem. The engineer of today is more likely than not to have as his problem a building that was constructed long before any consideration whatever was given to air conditioning it. Consequently, the room shapes, the location of columns and beams, and other details of architecture frequently make it difficult to properly locate the supply openings. In general, for a cooling installation, the grilles should be located high enough from the floor to prevent the discharge of air directly upon the occupants of the room, and far enough down from the ceiling to minimize the possibility of streaking, and to permit induction of air from all sides of the stream. If the stream actually strikes the ceiling, but at a

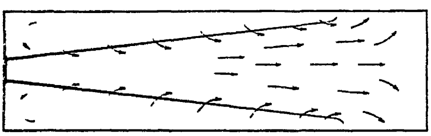


FIG. 1. PLAN VIEW LONG THROW SUPPLY OPENING

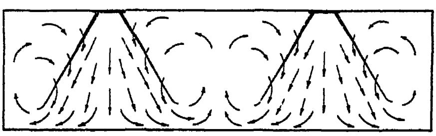


FIG. 2. PLAN VIEW SHORT THROW SUPPLY OPENINGS

small angle, the throw will be increased somewhat if the ceiling is smooth. If the angle at which the stream hits the ceiling is 20 deg or more, or if the flow along the ceiling is obstructed by panel mouldings or beams, air velocity may be rapidly lost and a decreased throw result. The air stream also should be so directed that it will not strike nearby columns or beams in such a way as to cause misdirection of the air stream or drafts. Where the room is of irregular shape, as an ell, or where it has an alcove in one side, consideration should be given to obtaining satisfactory circulation in these corners. Frequently this cannot be done except by the use of multiple supply openings. In using multiple supply openings, care must be taken that the several air streams do not interfere with each other, until their velocities have been reduced to values which will not cause high turbulence and a drafty condition. Beams and offsets in the ceiling will cause little difficulty when substantially parallel to the direction of flow, unless they are of considerable depth, but, when positioned across the air stream, may cause drafts and failure to secure satisfactory circulation in that portion of the room farthest from the supply opening. In the case of a heating installation, down-drafts produced by such obstructions may not be serious, because the air will rapidly lose its downward motion, but the possibility of failure to obtain satisfactory circulation still exists.

Alternative methods of distributing air to a room of oblong shape are illustrated in Figs. 1 and 2. The former usually is less expensive, but

requires a higher velocity to attain the greater throw. If the ceiling height is limited and the room is long, the air stream may enter the occupied zone before the completion of throw, due to natural spread and drop of the stream. In addition, the higher velocity and limited section area of the room perpendicular to the stream may, by induction, result in a general room air movement exceeding the limit permissible for comfort. Although more expensive, Fig. 2, is favored for rooms of low ceiling height since room induction is reduced, and spilling into the occupied zone is less likely to occur. With this latter arrangement, however, precautions should be taken to limit the throw in order to avoid drafts down the wall opposite the grilles.

In solving the problem of properly conditioning a room of irregular shape, where multiple wall supply grilles are objectionable, a ceiling supply opening of the type illustrated in Figs. 3 and 4 may very often be the best solution.

In choosing the most desirable location for the return air grille, consideration should be given to its effect on circulation of the air through

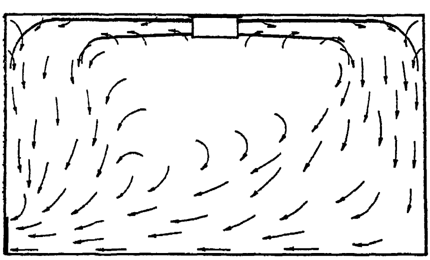


Fig. 3. Elevation View, Ceiling Supply Outlet with Return Opening in Wall

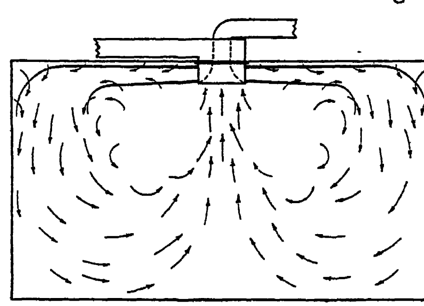


Fig. 4. Elevation View, Ceiling Supply and Return Openings

the room. The return air grille should preferably be placed on the same wall as the supply and near the floor level. This results in a U-shaped air path (Fig. 5) which covers the room thoroughly. The arrangement shown in Fig. 6 is less desirable, because it tends to create a stagnant section below the supply grille, but it may be entirely satisfactory where the supply grill produces an induction effect adequate to mix thoroughly the air introduced. What would otherwise be an unsatisfactory dead spot in a room may in some instances be taken care of by location of the return air grille near that area (Fig. 7).

STANDARDS FOR SATISFACTORY CONDITIONS

The most satisfactory air condition cannot be definitely stated for any particular individual without conducting a series of tests with that individual as subject; some persons are less sensitive than others to variations in temperature, humidity, air velocity and noise. The best that can be done is to attempt to set limiting conditions leaning toward the values of these variables which produce a condition of comfort for the greatest number of individuals. On a cooling installation, the allowable deviation from average room temperature, that is, the temperature of

puffs of air which may strike a person momentarily, is a function of the room temperature as well as the velocity of the air. For instance, in a room controlled at 72 F, a puff of air at 70 F might be uncomfortable to an individual, even at relatively low velocities, whereas if the average room temperature were 80 F, air at 78 F, even at moderate velocities, might be very satisfactory. However, air at 78 F in an average room temperature of 83 F would be cold. In general, other conditions being equal, for the range of temperatures normally encountered in living quarters on cooling installations, the permissible deviation from average room temperature varies from approximately 1 F at the low end of the range to about 3 F at the high end of the range. In this matter, it is important to consider the particular problem in the light of the type of

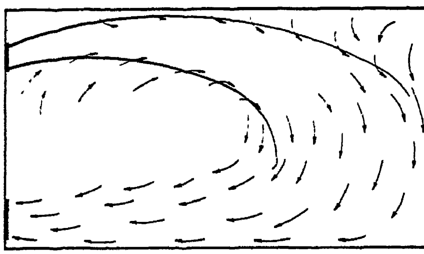


FIG. 5. PREFERRED LOCATION OF RETURN OPENING

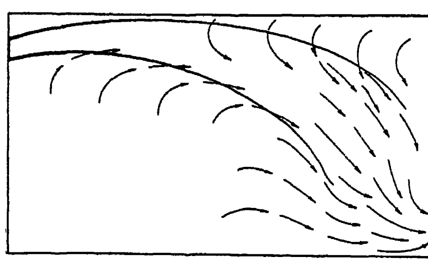


FIG. 6. POSSIBLE SHORT-CIRCUITING WITH RETURN OPENING OPPOSITE SUPPLY

occupancy. For instance, greater deviations from room temperature and higher velocities may be permitted in a garage or a hotel hallway than would be permissible in an office or living room. The velocity which may be considered the permissible maximum differs with the temperature deviation for a given installation, but an absolute maximum under any conditions might be considered that which would produce a mechanical disturbance, such as the movement of a person's hair or disturbance of papers on a desk. Humidity is an important consideration in the determination of one's feeling of comfort; however, if the room generally is assumed to be at a satisfactory value of relative humidity, the designer is justified in neglecting this factor when considering permissible fluctuations in temperature and velocity in the occupancy zone. This is true because the maximum allowable temperature fluctuation results in an unnoticeable humidity change.

The standards that might be set up for maximum allowable room

temperature deviation and air velocity would not be the same for both heating and cooling installations. In the former case, any appreciable temperature deviation is likely to be above rather than below the average room temperature, whereas the reverse is most likely to be true on a cooling installation. Further, because air movement has a cooling effect in itself, the feeling of warmth due to temperatures above room temperature is counteracted to a certain extent so that an individual may be subjected to higher velocities of warm air without the feeling of discomfort occasioned by the same velocities of cool air. In every case, it should be the purpose of the designing engineer to keep the conditions within the zone of occupancy as nearly uniform as possible, securing minimum temperature deviations and low velocities.

It is impractical to measure momentary temperature differences with any degree of accuracy in the field, but in checking a given installation it will generally be found satisfactory to measure velocity only, since on cooling installations high velocities normally occur with low temperatures, and on heating installations high velocities occur with high temperatures.

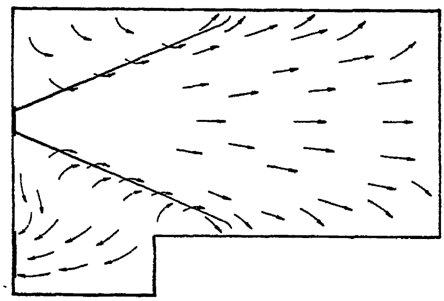


FIG. 7. PLAN VIEW CORRECTLY LOCATED RETURN OPENING ELIMINATING STAGNANT SPACE

That is, in the former case, the chilled supply air loses its velocity and undergoes an increase in temperature as it settles into the occupancy zone, whereas in the latter case the heated supply air loses its velocity and undergoes a decrease in temperature during this process. Therefore, if the average velocities within the occupancy zone are not excessive, one is fairly safe in assuming that the temperature difference is also within permissible limits.

The subject of sound control is covered in Chapter 33 and it is recommended for detailed review before consideration of the problem of air supply opening noise. An understanding of the relation between sound intensity and loudness level in decibels, as well as the effect of the presence of sound absorbent materials in the room, is particularly necessary. A more detailed discussion of the nature of this problem appears later; whereas the following comments refer to what constitutes a satisfactory noise condition.

Obviously, the nature of the conditioned space is important when considering the allowable supply opening noise. In factories, press rooms, and similar spaces where the noise level is 65 db or higher, no complaints of grille noise are likely to be made. On the other hand, some homes,

offices, hospitals, and, most of all, radio broadcasting and movie sound studios, present a real problem which must be intelligently attacked if a satisfactory installation is to be made. In this chapter the noise of the air supply openings (and returns) only is considered, it being assumed that the noise or sound level of the room without the supply opening noise includes that which may be contributed by fans, motors, duct work, and other items of conditioning equipment. The control of noise from these sources is another problem (see Chapter 33). Where sound control is important, the actual room sound level without conditioning equipment should be known. If feasible, the contribution of the conditioning equipment, less supply openings, should be estimated to secure the working sound level. If this correction is not made, the use of the first value errs in the direction of safety.

It is evident that the point within the room which should concern the designer in this problem is that at which the supply opening noise is greatest. A tentative standard *listening point* relative to the supply opening is suggested later in this discussion, and it is assumed that the supply opening noise data are taken with reference to this point. If it is desired that the supply opening noise result in an inaudible addition to the existing noise level, it is safe to assume the total supply opening noise to be 5 db below room level. This results in an increase in total noise of slightly over 1 db, which is unnoticeable. If an increase of 3 db is permissible, the supply opening noise level may be equal to the room noise level alone. All supply openings in the room must be considered, as will appear later; the returns may be ignored only if they are so sized that the velocity of air through them is much less than through the supply opening.

DISTRIBUTION FACTORS IN ROOM COOLING

In attempting to design a satisfactory air distributing system, it is first necessary to properly locate the grilles in accordance with the recommendations already stated. Assuming that the best locations have been selected, it then becomes necessary to choose the proper grille for that location. The considerations involved are the amount of air to be handled, the velocity permissible from the standpoint of noise, and the distance the air should carry. The distance it will carry, assuming no obstructions, is affected by face velocity, core area, aspect ratio and included angle of effluent stream as determined by vanes. For low aspect ratios, the major variables of velocity, area and effluent angle are related[1] approximately as given in Equation 1 when the air stream is unaffected by obstructions of any kind.

$$X_a = \frac{kQ}{\sqrt{a_o b_o}} \tag{1}$$

where

X_a = throw, feet.

Q = air volume flow rate, cubic feet per minute.

a_o and b_o = grille width and height, inches.

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[1]The Rationale of Air Distribution and Grille Performance, by C. O. Mackey (*Refrigerating Engineering*, Vol. 35, No. 6, June, 1938, p. 417).

k = dimensionless constant with the following approximate empirical values:
Vanes set straight ahead .... = 0.77
Vanes causing a spread on each horizontal side of 15 deg = 0.66
30 deg = 0.45
45 deg = 0.34

The temperature difference between incoming air and room does not appreciably effect distance of throw, but does modify the point in the room at which the stream enters the occupied zone, due to rise or drop of the entire stream[2].

In consideration of what constitutes the possible throw of a supply opening under a given set of conditions, it is important to remember that the throw may be unsatisfactory for any one of several reasons:

1. It may be so long that it will strike the far side of the room and come down the wall with velocities higher than are permissible,
2. It may be so short that it will fail to carry the full length of the room, and short-circuit to the return air supply opening, or
3. It may spill into the center of the room.

In the first case, the system fails for lack of uniform distribution and the presence of cold areas. In the second case, the standards as to velocity and temperature difference in the zone of occupancy may be satisfactorily met, but air distribution and circulation throughout the entire room is not accomplished, with the result that the end of the room away from the outlet would not be satisfactorily conditioned. In the third case, the shortcomings of both case one and case two are present. It is evident, therefore, that for a given supply opening discharging air at a given velocity, there is a maximum and a minimum length of room which can be satisfactorily handled. In the latter, the velocity of the air down the far wall is just within the maximum permissible, while in the former, satisfactory circulation is barely accomplished.

In general, the higher the supply opening is above the floor, the greater may be the difference between room air and incoming air temperatures.

Assuming that proper supply openings for a given installation have been selected, unsatisfactory performance may still result due to the construction of the duct work immediately back of the supply openings. Performance data on the grilles and registers of various manufacturers are based upon results obtained with the air approaching the grille perpendicularly and at uniform velocity over the entire duct cross-section. Where this condition does not exist in practice, performance predictions based on published data cannot be expected to be realized. Every precaution should be taken to secure as nearly ideal conditions in the approaching air stream as are possible.

In addition to disturbances due to the construction of the duct work itself are those which may be created by dampers immediately behind the grille. Where either multiple louver or single blade dampers are used, considerable deflection of the air stream may result, if it is throttled appreciably by these means. This is particularly true when the fins of the register core are perpendicular to the damper blades. If the core has sufficient depth and the fins are parallel to the blades, there is a marked tendency to straighten the air stream, although some deflection may still result.

[2]A.S.H.V.E. RESEARCH REPORT No. 1076—Air Distribution from Side Wall Outlets, by D. W. Nelson and D. J. Stewart (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 77).

Any attempt to secure a low face velocity and high duct velocity by the construction of any expanding chamber immediately behind the grille is likely to be unsuccessful. In order to expand from a small duct to a larger one, and have the air stream fill the duct at the end of the diverging section without turbulence, angle A in Fig. 8 should be about 7 deg. From this it is apparent that an attempt to secure equivalent results with a short connection would be futile. What actually happens when this is attempted is illustrated by the arrows in Fig. 8. When localized high velocities through the supply opening exist from this cause or any other, the noise produced will naturally exceed that which the supply opening area and average face velocity would lead one to expect. This fact should be remembered in considering the use of register dampers, particularly in those cases where there must be considerable throttling with the damper to balance a poorly designed system. Where reduction of noise is important, it is recommended that balancing dampers be placed in the duct *ahead* of the acoustic duct lining.

Similar unequal face velocities, aggravated by a deflection of the air stream, are obtained with the arrangement shown in Fig. 9. The latter

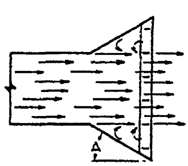


FIG. 8. EFFECTS OF EXPANDING DUCT

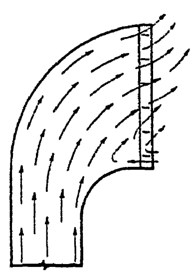


FIG. 9. UNEQUAL FACE VELOCITIES

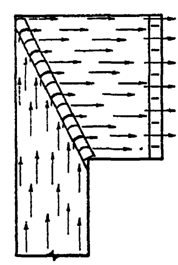


FIG. 10. EFFECT OF TURNING MEMBER

may be corrected by inserting a turning member in the elbow back of the outlet face as shown in Fig. 10. The importance of straightening the air stream and effecting uniform distribution over the entire face of the supply opening cannot be over-emphasized.

DISTRIBUTION FACTORS IN ROOM HEATING

The problem in the case of a heating installation is substantially the same as in cooling, with a few exceptions. Because the temperature of the incoming air is above that of the room, there is no tendency for it to drop and consequently the throw is not particularly affected by temperature difference in a low ceiling room. In general, the air should be deflected downward where the grille is above the occupancy zone, and this is particularly desirable where the ceiling is high. For the same reason, that is, to keep the heat in the occupancy zone and to avoid excessive temperature at the ceiling, it is desirable to have the grille comparatively low on the wall, and just slightly above the occupancy zone. If the grille is lower than this, it may create an unsatisfactory condition of very warm air at quite high velocities where it can possibly strike the occupants of the room. Where the velocities are very low, the grilles may even be satisfactorily located below the 6-ft level, although the immediate vicinity

of the supply openings will probably be useless for occupancy because of high temperature. Essentially, the problem is to keep the incoming air up for cooling, and down for heating, until it is thoroughly mixed with the room air. Grilles and registers which are adjustable for deflection upward and downward, either by moving the fins or inverting the grille, are in general use.

AIR SUPPLY OPENING NOISES

When air is introduced into a room through a grille or register at a constant velocity, sound energy is being introduced into the enclosure at a constant rate[3]. Due to partial reflection at the boundaries of the enclosure, the intensity of sound at any point in the space builds up to some maximum value. In a large room at a point remote from the source of sound (the supply opening) the intensity can be shown to be substantially proportional to the rate at which sound energy is generated and inversely proportional to the number of sound absorption units (sabins) in the room. It would thus appear that doubling the sound absorption of the room would halve the intensity and result in a noise level decrease of 3 db. However, it is not satisfactory to consider the grille noise on this basis (wherein the sound power received directly from the source is small compared with that received by reflection) since in practice the occupants of the room may be quite close to the grille. The nearer the listener is to the sound source, the greater the proportion of the sound intensity which is due to direct transmission.

In the absence of generally accepted standards at this time it is suggested that the loudness level 5 ft from the lower edge of the supply opening, measured downward at 45 deg in a plane perpendicular to the supply opening at its center, represents about the maximum within the zone of occupancy. The cases where persons are nearer to the supply opening than this are rare and are ignored in the consideration of this problem. Although the effect of sound absorbent material on the intensity at the 5-ft station is not nearly so great as at more remote points in the room, it should not be ignored without consideration of the error involved. An average living room may contain 100 sabins (absorption units). If this be decreased to 50 sabins, the *diffuse* or reflected sound level would be increased 3 db. However, at the 5-ft station the increase would be less than 2 db. If the absorption of the room be increased to 200 sabins, one might expect a reduction in diffuse noise of 3 db; but at the 5-ft station the reduction would be less than 1½ db. Furthermore, even though the absorption be increased without limit (as in free space) the reduction would still be less than 2 db because of proximity to the source.

In comparing sound ratings of various grilles, the following must be known if the information is to be intelligently applied:

1. The threshold intensity on which the decibel ratings are based.
2. The distance from the grille at which data were taken.
3. If stated as loudness level versus velocity for a given grille, the *core area* (not nominal area) must be known.
4. The sound absorbing characteristics of the test room.
5. Whether or not corrected for test room loudness level; if not, the room level (without grille noise) must be known.
6. Methods used for recording data. (Characteristics of sound meter).

[3]The Noise Characteristics of Air Supply Outlets, by D. J. Stewart and G. F. Drake (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 81).

Data mentioned in this chapter are based on these assumptions:

1. Threshold intensity = 10^{-16} watts per square centimeter[4].
2. Microphone location 5 ft from lower edge of supply opening on a line downward at 45 deg and in a plane bisecting the supply opening perpendicularly.
3. Where data are given as loudness level versus velocity, the rating is per square foot of core area.
4. The room is assumed to have 100 sabins absorption.
5. Plotted data are loudness levels of *supply openings only,* correction having been made for test room level.
6. Data taken with a direct reading sound-level meter with frequency weighing network intended to approximate the response of the human ear.

If the published ratings are in terms of decibels per square foot, correction must be made for area to secure the total sound level of supply

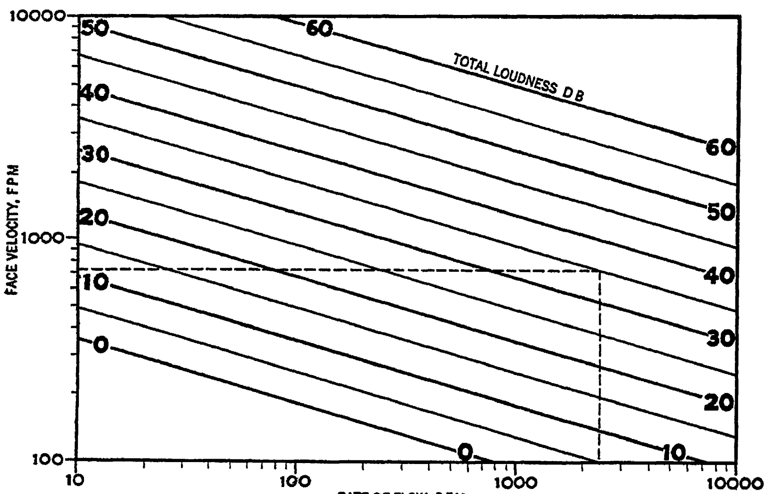


FIG. 11. AIR FLOW AND LOUDNESS CHART

openings of more or less than one square foot area from Equation 2.

$$\text{Decibel Addition} = 10 \log_{10} A \tag{2}$$

where

A = core area, square feet.

In practice the allowable total sound and the required air flow are usually known, and it is desired to determine the maximum allowable velocity. Since total loudness and air flow are both functions of velocity and area, the solution of the problem by use of the previous analysis implies a trial and error method. It has been found possible to present these data with sufficient practical accuracy as a family of uniform curves as illustrated in Fig. 11. With this chart it is possible to find directly the velocity in feet per minute which will give a predetermined total loudness at a predetermined rate of flow expressed in cubic feet per minute. The

[4]American Tentative Standards for Noise Measurement, *American Standards Association.*

values used are arbitrarily chosen for the purpose of discussion and do not necessarily represent data referring to any particular design of air supply opening. It is assumed that Fig. 11 is based on a room having 100 sabins of sound absorption. In such a room the sound level due to other sources may be 40 db. As previously stated a supply opening having a noise level of 35 db would be substantially inaudible in such a room.

If 2400 cfm are required with a total noise due to supply opening of 35 db, a velocity (Fig. 11) of about 725 fpm may be used. From this velocity and the rate of flow, the core area can be computed. This determination was on the basis of a room absorption of 100 sabins. If

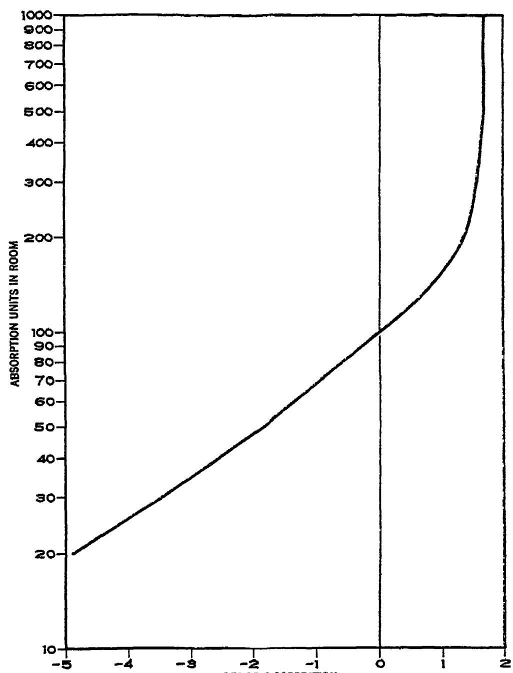


FIG. 12. ROOM ABSORPTION CORRECTION CHART

the absorption is greater, the 725 fpm velocity is safe, since the loudness level will go down. However, correction can be made if desired by the use of the chart of Fig. 12. Thus, if the absorption is 200 sabins, a correction of +1.3 db may be made and the permissible velocity becomes that corresponding to a total loudness level of 36.3 db or approximately 800 fpm. If the room is highly reflecting and has an absorption of less than 100, correction is much more important. For instance, for 35 sabins a correction of −3 db must be made and the maximum velocity corresponding to 32 db total loudness chosen is approximately 600 fpm.

Where more than one supply opening must be considered, the problem is more complicated. If a similar supply opening is added in a far corner of a highly absorbent room, the change in noise level at the 5-ft station at the first supply opening is small; however, if the room is small, or highly

reverberant or both, the intensity at the 5-ft station may be almost doubled and the noise level increased nearly 3 db thereby. The simplest method of handling this problem, and one which errs in the direction of safety, is to treat the room as though all the air were being supplied by one supply opening. Thus, if two outlets, each supplying 1000 cfm are used, the value 2000 cfm should be used with Fig. 11. Although this method may place an unwarranted limit on velocity when used in a large room, it is seldom that such a room has a noise level low enough to justify a more complicated though more exact procedure.

In general, return grilles are selected for velocities about half the supply velocity, and when this is done, they may be neglected in sound computations. However, if supply and return grilles are the same size, resulting in the same face velocity, they must be treated as two supply openings. That is, if 1000 cfm are supplied and exhausted through grilles of the same area, 2000 cfm must be used in the solution with Fig. 11.

SELECTION OF SUPPLY OPENINGS

After the heating and cooling load calculations have been made (Chapters 6 and 7), and a suitable supply air temperature selected, the volume of air required for each space can be determined. The next step is to determine the velocity at which the air may be introduced into the space quietly and without creating objectionable drafts.

Although grille face velocities up to 1500 and even 2500 fpm are sometimes feasible, such velocities are impractical in most cases, primarily because either the throw or the induced air motion is excessive in the particular room and secondarily on account of noise. With present emphasis on draftless distribution, the trend is toward velocities between 400 and 1000 fpm. Selection of the proper velocity requires that the designer have reliable data applicable to the particular make of grille proposed.

A method for selecting supply openings is outlined in the form of a sample cooling problem, using numerical values which have no reference to any particular make of supply opening.

1. The load calculations have been made; a suitable temperature differential has been selected (it is to be understood that the data referred to from this point on are based on this temperature differential), and the volume of air required determined. Assume that Fig. 13 represents a small general office having a noise level of 40 db and that 2400 cfm must be supplied for proper conditioning.

2. Select a tentative location for the supply opening or openings, having in mind the type of grille most likely to effect proper distribution. In this particular case, two supply openings having a wide spread appears to be a logical choice.

3. Data from which to determine velocity which corresponds to 2400 cfm and a noise rating at least 5 db below the noise level of the office may be presented in a number of forms, one of which is shown in Fig. 11. (Fig. 11 represents assumed values only. In practice similar data should be obtained from the manufacturer whose supply openings are being considered. Several similar charts or tables may be necessary to cover any one manufacturer's complete line.) From Fig. 11 it will be noted that for 2400 cfm the type of grille selected may be used at velocities up to 725 fpm without exceeding 35 db; that is, 5 db below the noise level of office.

4. Having determined the velocity, the core area becomes fixed at 3.31 sq ft or 397 sq in. per supply opening. In this problem, the two grilles in question are so close together that consideration of their combined area in determining the permissible velocity from the standpoint of noise introduces little error.

5. The type grille selected has thus far been found satisfactory from a noise standpoint, provided the face velocity does not exceed 725 fpm. The next consideration is throw, which may be assumed to be 16 ft, and by reference to a manufacturer's catalogue the proper correlative test data may be checked with the throw assumed. It is of course evident that one or more types of grilles may satisfy the requirements, and that in any one type there will be a choice of supply opening proportions. It will also be evident that the tentative selection of a supply opening having a wide spread may be unsatisfactory from the standpoint of throw, in which event a second choice should be made and the procedure repeated.

In the case of a heating problem, the method of solution is the same, but the manufacturer's data should, of course, be based on tests with air above room temperature. However, if data based on chilled air are used for a heating problem the grille selection will err on the side of safety.

TYPES OF SUPPLY OPENINGS

Grille, register or supply opening design for attaining uniform distribution and minimum air resistance consists of various fixed and adjustable

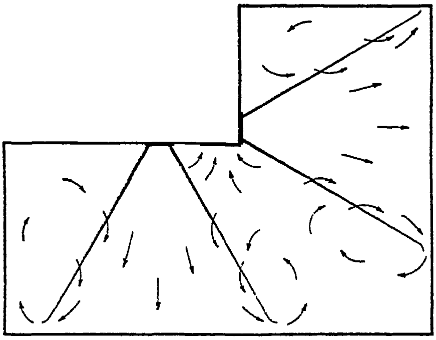


FIG. 13. PLAN VIEW TYPICAL GENERAL OFFICE

arrangements. Some types are designed with directing air blades, fins, bars, louvers, or thin metal strips shaped into a series of grooves or tubes, all of which may be set into a suitable round, square or rectangular frame. In order to attain desired long or short air throws, the emergence of air from the supply opening may be directed to straight, deflecting, converging or jet air streams depending upon the supply opening design. Designs which direct the air stream to produce an ejector effect within the enclosed space tend to mix the room air with the conditioned air.

There are several types of centrally located ceiling or wall type supply openings. One of these consists of several round, hollow, cone-shaped flaring members with one or more of the smaller members acting as ejectors and injectors. An idea for producing even distribution of air consists of a perforated ceiling made of a suitable architectural surface and installed a small distance below the normal ceiling level of the room. In the space provided by this suspended ceiling a plenum chamber is formed into which the conditioned air is introduced. From the plenum space the air is permitted to diffuse through the large number of small ceiling openings into the room.

BALANCING SYSTEM

In designing an air conditioning system, it should be the aim of the engineer to so proportion the duct system that proper distribution of air to every supply opening will be obtained. Since this is almost impossible to accomplish in practice, it becomes necessary to have means of balancing the system to secure the desired amount of air in each space. There are a number of ways in which this may be accomplished, some of which are:

1. Dampers on the supply and return grilles.
2. Dampers in the supply and return ducts.
3. Reducing the effective area of some supply openings by blank-offs.
4. Combinations of dampers in both supply and return air.

Dampers on the supply grilles themselves are objectionable because of their effect on the air stream. Dampers on the return grilles are frequently helpful in building up a static pressure in the room to prevent infiltration of outside air, and at the same time reduce the volume of incoming air. However, it is frequently impossible to sufficiently reduce the incoming air by this method alone. A damper in the supply duct some distance back of the supply opening forms a very satisfactory means of regulating the flow without disturbing distribution across the supply opening face. A damper in the return air duct has the advantage over one immediately behind the grille in that it does not tend to create high localized velocities through the grille as the latter might do if nearly closed. Blank-offs consisting of pieces of sheet metal covering a portion of the supply opening face can frequently be used satisfactorily, although determination of just what is required is a matter of experiment, and the balancing of the system is not nearly so conveniently accomplished as with dampers. Dampers in both supply and return air form the most flexible means of controlling the supply to the room and the static pressure within the room. When feasible, these dampers, particularly those in the supply ducts, should be a substantial distance from the supply opening, and ahead of the acoustic duct lining if used. Due consideration should also be given to the use of the several volume control and uniform distribution devices now available. See *Catalog Data Section*.

REFERENCES

A.S.H.V.E. RESEARCH REPORT No. 1140—The Use of Air Velocity Meters, by Prof. G. L. Tuve, D. K. Wright, Jr. and L. J. Seigel (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 645).

A.S.H.V.E. RESEARCH REPORT No. 1156—The Performance of Stack Heads, by D. W. Nelson, D. H. Krans and A. F. Tuthill (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940).

A.S.H.V.E. RESEARCH REPORT No. 1163—Air Flow Measurements at Intake and Discharge Openings and Grilles, by G. L. Tuve and D. K. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940).

A.S.H.V.E. RESEARCH REPORT No. 1166—Development of Instruments for the Study of Air Distribution in Rooms, by A. P. Kratz, A. E. Hershey and R. B. Engdahl (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940).

A.S.H.V.E. RESEARCH PAPER—Entrainment and Jet-Pump Action of Air Streams, by G. L. Tuve, G. B. Priester and D. K. Wright, Jr. (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, November, 1941, p. 708).

Chapter 32

AIR DUCT DESIGN

Pressure Losses, Friction Losses, Friction Loss Chart, Elbow Friction Losses, Proportioning the Losses, Duct Sizes, Procedure for Duct Design, Velocities, Main Trunk Ducts, Proportioning the Size for Friction, Velocity Method, Equal Friction Method, Duct Construction Details

THE resistance of an air handling system can be computed from the methods and data given in this chapter. The actual resistance for any given installation, however, may vary considerably from the calculated resistance because of variation in the smoothness of materials, the type of joints used and the ability of the mechanics to fabricate in accordance with the design. It is best to select fans and motors of sufficient size to allow a factor of safety. Volume dampers should be installed in each branch outlet to balance the system. It is improbable that the required quantities of air will be delivered at each outlet without adjustment of the dampers, which usually results in a total pressure exceeding that of the design, unless a liberal factor of safety is allowed.

The flow of air due to large pressure differences is most accurately stated by thermodynamic formulae for air discharge under conditions of adiabatic flow, but such formulae are complicated, and the error occasioned by the assumption that the gas density remains constant throughout the flow may be considered negligible when only such pressure differences are involved as occur in ordinary heating and ventilating practice.

In the development of the formulae, diagrams, and tables for the flow of air, use is made of the following basic equation for the flow of fluids:

If H_V be the velocity head in feet of a fluid, and the velocity, V, be expressed in feet per minute, the fundamental equation is

$$V = 60\sqrt{2g\,H_V}$$

The factor g is the acceleration due to gravity, or 32.16 ft per second per second.

It is usual to express the head in inches of water for ventilating work and, since the heads are inversely proportional to the densities of the fluids,

$$\frac{H_V}{\frac{h_V}{12}} = \frac{62.4}{d}$$

or

$$H_V = 5.2\,\frac{h_V}{d}$$

therefore,

$$V = 1096.5 \sqrt{\frac{h_v}{d}} \tag{1}$$

where

V = velocity, feet per minute.
h_v = velocity head or pressure, inches of water.
d = weight of air, pounds per cubic foot.

For dry air (70 F and 29.921 in. *Hg* barometer) d = 0.075 lb per cubic foot[1]. Substituting this value in Equation 1:

$$V = 1096.5 \sqrt{\frac{h_v}{0.075}} = 4005 \sqrt{h_v} \tag{2}$$

The relation of air velocity and velocity head expressed in Equation 2 is shown diagrammatically in Fig. 1 for air at 70 F and 29.92 in. *Hg* barometer.

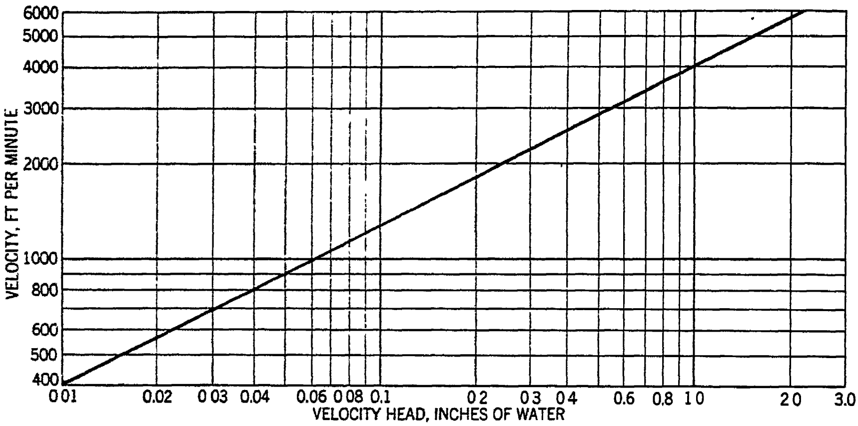


Fig. 1. Relation Between Velocity and Velocity Head for Dry Air

The drop in pressure in air distributing systems is due to the *dynamic* losses and the *friction* losses. The friction losses for turbulent flow (which occur in all practical air flow problems) are due to the friction of air against the sides of the duct and to internal friction between air molecules. The dynamic losses are those due to the change in the direction or in the velocity of air flow.

Dynamic losses occur principally at the entrance to the piping, in the elbows, and wherever a change in velocity occurs. The entrance loss is the difference between the actual pressure required to produce flow and the pressure corresponding to the flow produced; it may vary from 0.1 to 0.5 times the velocity head. The pressure loss in elbows must also be allowed for in the design.

FRICTION LOSSES

A study of the frictional resistance to the flow of air in ducts was begun by the A.S.H.V.E. Research Laboratory in 1938. This study

[1]See Chapter 47 for definition of standard air.

resulted in modifications of the Fanning friction loss formula, for 100 ft of round galvanized iron duct and for air at standard conditions[2]:

For round duct with no joints,

$$H_S = \frac{1.157}{D^{1.212}} \left(\frac{V}{4000}\right)^{1.83} \tag{3}$$

For round duct with 40 joints per 100 ft,

$$H_S = \frac{1.48}{D^{1.287}} \left(\frac{V}{4000}\right)^{1.83} \tag{4}$$

where

H_S = friction loss, inches of water at standard conditions.
V = velocity of air, feet per minute.
D = diameter of duct, feet.

The chart shown in Fig. 2 was constructed from Equation 4 and therefore applies only for round galvanized iron duct of good construction with 40 joints per 100 ft, and for air at standard conditions. *No factor of safety has been applied.* In view of the many variations that may occur in duct construction and application, it is recommended that a factor of safety be used, which in the judgment of the engineer, will make due allowance for these variations. In the Laboratory tests the variation found in pressure loss between the best joints and the worst joints was approximately 10 per cent. This would suggest a minimum factor of safety of 10 per cent.

Since the friction chart applies only for standard conditions, it is necessary to apply correction factors for other than standard conditions. These corrections are as follows:

$$H_A = H_S \times S \left(\frac{\gamma_A}{\gamma_S}\right)^{0.17} \tag{5}$$

where

H_A = friction loss, inches of water at actual conditions.
S = ratio of density of air at actual conditions to density of air at standard conditions.
γ_A = kinematic viscosity at actual conditions.
γ_S = kinematic viscosity at standard conditions.

$$\text{Kinematic Viscosity} = \frac{\mu}{\rho}$$

where

μ = absolute viscosity, pounds per foot second (see Fig. 3).
ρ = density, pounds per cubic foot (see Chapter 1).

The absolute viscosity of dry air at various temperatures is given in Fig. 3. It is assumed that the viscosity is not appreciably affected by the moisture content.

For temperatures ordinarily used in heating, ventilating and air conditioning work, the correction for viscosity may be neglected without

[2]Frictional Resistance to the Flow of Air in Straight Ducts, by F. C. Houghten, J. B. Schmieler, J. A. Zalovcik and N. Ivanovic (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 35). Analysis of Factors Affecting Duct Friction, by J. B. Schmieler, F. C. Houghten and H. T. Olson (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940).

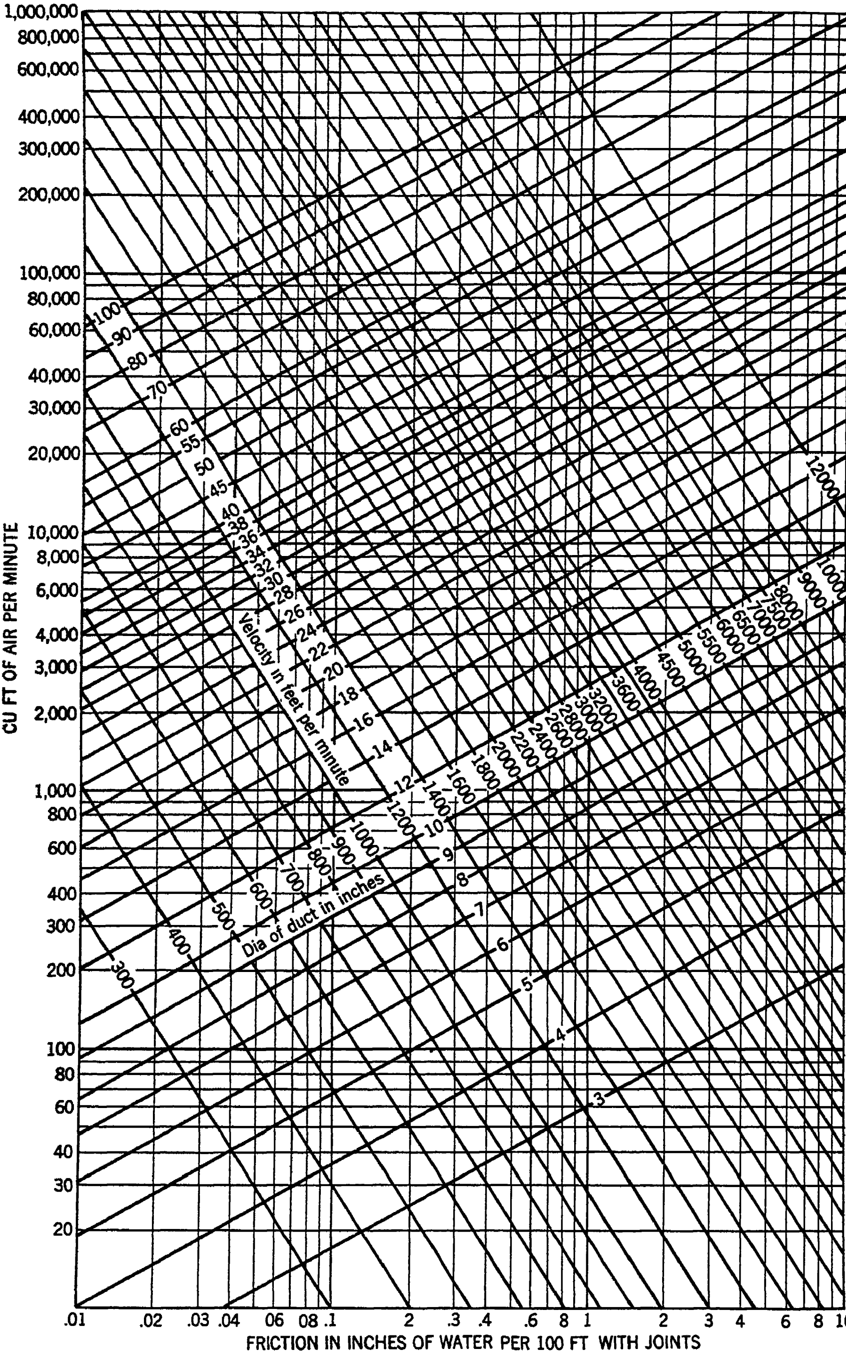


FIG. 2. FRICTION OF AIR IN PIPES

serious error. The correction equation would then be simplified to the following:

$$H_A = H_s \times S \tag{6}$$

Example 1. Assume that it is desired to circulate 10,000 cfm of air through 75 ft of 24 in. diameter pipe. Find 10,000 cfm on the right scale of Fig. 2 and move horizontally left to the diagonal line marked 24 in. The other intersecting diagonal shows that the velocity in the pipe is 3200 fpm. Directly below the intersection it is found that the friction per 100 ft is 0.41 in.; then for 75 ft the friction will be 0.75 × 0.41 = 0.31 in. In a like manner any two variables may be determined by the intersection of the lines representing the other two variables.

Circular Equivalents of Rectangular Ducts

Where rectangular ducts are used it is frequently desirable to know the equivalent diameter of round pipe to carry the same capacity and have

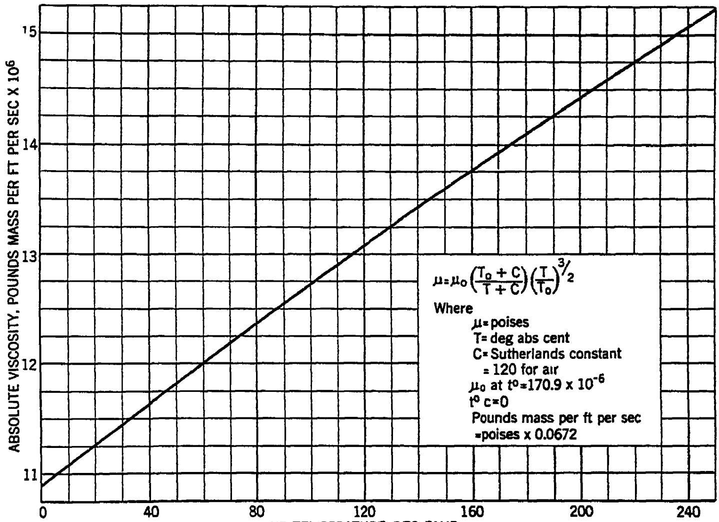


FIG. 3. TEMPERATURE—VISCOSITY CURVE FOR DRY AIR

the same friction per foot of length. Table 1 gives directly the circular equivalents of rectangular ducts for equal friction and capacity, which are based on values determined from Equation 7:

$$d = 1.265 \sqrt[5]{\frac{(a\,b)^3}{a + b}} \tag{7}$$

where

a = one side of rectangular pipe, feet or inches.
b = other side of rectangular pipe, feet or inches.
d = equivalent diameter of round pipe for equal friction per foot of length to carry the same capacity, feet or inches.

Rectangular equivalents of round ducts are also given in the curves of Fig. 4 which are plotted from data based on Formula 7. To use the chart, locate the curve giving the diameter of the round duct. The width and height of an equivalent rectangular duct may then be read as abscissa and ordinate of any point of the curve.

Multiplying or dividing the length of each side of a pipe by a constant is the same as multiplying or dividing the equivalent round size by the same constant. Thus, if the circular equivalent of an 80 x 24-in. duct is required, it will be twice that of a 40 x 12-in. duct, or 2 $\times$ 23.3 = 46.6 in.

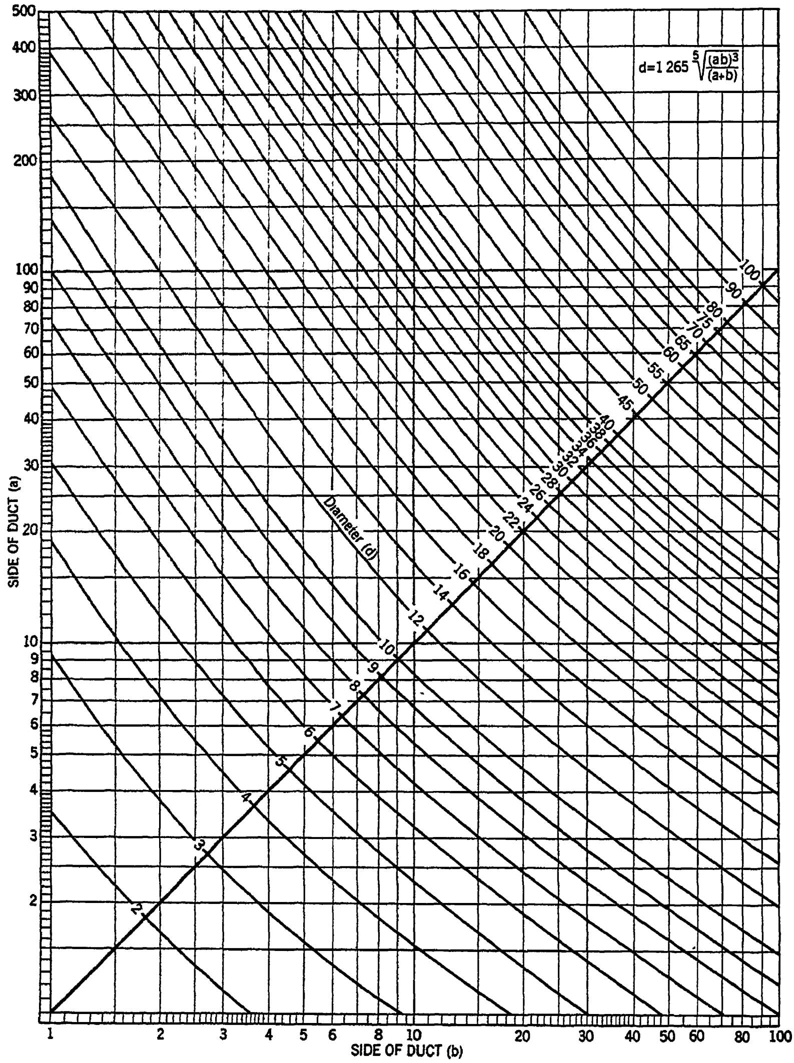


Fig. 4. Rectangular Equivalents of Round Ducts

Elbow Friction Losses

It is customary to express the dynamic and friction losses in elbows as equal to a number of diameters of round pipe or a number of widths of rectangular pipe. The curves in Fig. 5 are arranged to read the number

of diameters or widths for determining the lineal feet of pipe having a frictional resistance equivalent to the pressure drop in the elbows. Curves *B* and *C* are based on tests of round and square elbows[3] of ordinary good sheet metal construction having a surface factor of $C = 50$.

Values obtained from Curve *A* should be used when there is any doubt as to quality of duct construction. It is suggested that this curve be used for rectangular elbows and five piece elbows as it will thus allow an additional factor of safety without seriously affecting the design.

As indicated on the chart, long radius elbows will offer much less resistance to the flow of air than short radius elbows. Experience has shown that good results may be expected when the radius to the center

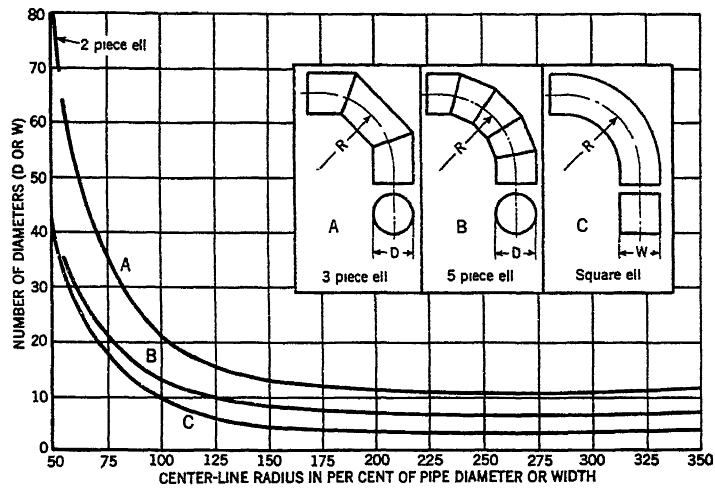


FIG. 5. LOSS OF PRESSURE IN ELBOWS

of the elbow is 1.5 times the pipe diameter or duct width parallel to the radius. Examination of the curve will indicate that little advantage is to be gained by selecting elbows having a centerline radius of more than two

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION

Side Rectangular Duct	8	8.5	9	9.5	10	10.5	11	11.5	12	12.5	13	13.5	14	14.5	15	15.5	16
3	5.2	5.4	5.5	5.7	5.8	5.9	6.0	6.2	6.3	6.4	6.5	6.6	6.7	6.8	6.9	7.0	7.1
3.5	5.7	5.9	6.0	6.2	6.3	6.5	6.6	6.7	6.9	7.0	7.1	7.3	7.4	7.5	7.6	7.7	7.8
4	6.1	6.3	6.5	6.7	6.8	7.0	7.1	7.2	7.4	7.5	7.7	7.8	7.9	8.1	8.2	8.3	8.4
4.5	6.5	6.7	6.9	7.1	7.2	7.4	7.6	7.7	7.9	8.0	8.2	8.4	8.5	8.6	8.7	8.9	9.0
5	6.9	7.1	7.3	7.5	7.7	7.8	8.0	8.2	8.3	8.5	8.7	8.8	8.9	9.1	9.2	9.4	9.5
5.5	7.3	7.5	7.7	7.8	8.1	8.3	8.5	8.6	8.8	9.0	9.2	9.4	9.5	9.6	9.8	9.9	10.1

[3]Loss of Pressure Due to Elbows in the Transmission of Air Through Pipes or Ducts, by F. L. Busey (A.S.H.V.E. TRANSACTIONS, Vol. 19, 1913, p. 366).

Table 1. Circular Equivalents of Rectangular Ducts for Equal Friction[a]—(Continued)

Side Rectangular Duct	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	24
8	6.1	6.9	7.6	8.2	8.8															
9	6.5	7.3	8.0	8.7	9.3	9.9														
10	6.8	7.7	8.4	9.2	9.8	10.4	11.0													
11	7.1	8.0	8.8	9.6	10.2	10.9	11.5	12.1												
12	7.4	8.3	9.2	10.0	10.7	11.4	12.0	12.6	13.2											
13	7.6	8.7	9.6	10.4	11.1	11.8	12.5	13.1	13.7	14.3										
14	7.9	8.9	9.9	10.8	11.5	12.3	12.9	13.6	14.3	14.9	15.4									
15	8.2	9.2	10.2	11.1	11.9	12.7	13.4	14.1	14.7	15.3	16.0	16.5								
16	8.4	9.5	10.5	11.4	12.3	13.1	13.8	14.5	15.2	15.8	16.5	17.1	17.6							
17	8.6	9.8	10.8	11.8	12.6	13.5	14.2	15.0	15.7	16.3	17.0	17.6	18.2	18.7						
18	8.9	10.0	11.1	12.1	13.0	13.8	14.6	15.4	16.1	16.8	17.4	18.1	18.7	19.2	19.8					
19	9.1	10.3	11.4	12.4	13.3	14.2	15.0	15.8	16.5	17.2	17.9	18.6	19.2	19.8	20.4	20.9				
20	9.3	10.5	11.6	12.7	13.6	14.5	15.4	16.2	17.0	17.6	18.4	19.0	19.7	20.3	20.9	21.5	22.0			
22	9.7	11.0	12.1	13.2	14.2	15.2	16.1	16.9	17.8	18.5	19.2	19.9	20.6	21.3	21.9	22.5	23.1	23.6	24.2	
24	10.0	11.4	12.6	13.8	14.8	15.8	16.8	17.6	18.5	19.3	20.0	20.8	21.5	22.2	22.8	23.5	24.0	24.7	25.2	26.4
26	10.4	11.8	13.1	14.3	15.4	16.4	17.3	18.3	19.2	20.0	20.8	21.6	22.3	23.0	23.8	24.4	25.1	25.7	26.3	27.5
28	10.8	12.2	13.5	14.8	15.9	17.0	18.0	19.0	19.8	20.7	21.5	22.4	23.1	23.9	24.6	25.3	26.0	26.6	27.3	28.5
30	11.0	12.6	13.9	15.2	16.4	17.5	18.5	19.5	20.5	21.4	22.2	23.1	23.9	24.7	25.4	26.2	26.8	27.5	28.2	29.5
32	11.3	12.9	14.3	15.6	16.9	18.0	19.1	20.1	21.1	22.0	22.9	23.8	24.6	25.4	26.2	27.0	27.7	28.4	29.1	30.5
34	11.6	13.2	14.7	16.1	17.3	18.5	19.6	20.7	21.6	22.6	23.5	24.4	25.3	26.2	26.9	27.7	28.5	29.2	30.0	31.3
36	11.9	13.6	15.1	16.4	17.7	19.0	20.1	21.2	22.2	23.2	24.2	25.1	26.0	26.8	27.7	28.5	29.3	30.0	30.8	32.2
38	12.2	13.9	15.4	16.8	18.2	19.4	20.6	21.7	22.8	23.8	24.8	25.8	26.7	27.5	28.4	29.2	30.0	30.8	31.5	33.1
40	12.5	14.3	15.7	17.2	18.6	19.8	21.1	22.2	23.3	24.4	25.4	26.4	27.3	28.2	29.1	29.9	30.8	31.6	32.4	33.9
42	12.7	14.5	16.1	17.6	19.0	20.3	21.6	22.7	23.8	24.9	25.9	26.9	27.9	28.8	29.8	30.7	31.4	32.2	33.0	34.5
44	13.0	14.8	16.4	18.0	19.4	20.7	22.0	23.1	24.3	25.4	26.5	27.5	28.5	29.5	30.3	31.2	32.1	32.9	33.7	35.3
46	13.3	15.1	16.7	18.4	19.8	21.1	22.4	23.6	24.8	25.9	27.0	28.1	29.1	30.1	31.0	31.9	32.8	33.8	34.6	36.2
48	13.5	15.4	17.0	18.7	20.1	21.5	22.8	24.1	25.2	26.4	27.5	28.6	29.6	30.5	31.6	32.5	33.4	34.3	35.2	37.0
50	13.7	15.7	17.3	19.0	20.4	21.9	23.2	24.5	25.7	26.9	28.0	29.2	30.3	31.3	32.2	33.1	34.1	35.0	35.9	37.6
52	13.9	15.9	17.6	19.2	20.8	22.2	23.6	24.9	26.2	27.4	28.5	29.6	30.7	31.8	32.9	33.8	34.7	35.6	36.5	38.3
54	14.1	16.1	17.9	19.6	21.1	22.6	24.0	25.3	26.6	27.8	29.0	30.1	31.2	32.3	33.4	34.4	35.3	36.3	37.2	38.9
56	14.3	16.3	18.2	19.9	21.5	22.9	24.4	25.7	27.0	28.3	29.5	30.6	31.7	32.8	33.9	34.9	35.9	36.9	37.8	39.6
58	14.6	16.6	18.4	20.2	21.8	23.3	24.7	26.1	27.4	28.7	30.0	31.1	32.2	33.3	34.4	35.4	36.4	37.4	38.4	40.3
60	14.7	16.8	18.7	20.4	22.1	23.6	25.1	26.5	27.8	29.1	30.5	31.6	32.7	33.8	34.9	36.1	37.1	38.1	39.1	40.9
62	15.0	17.0	19.0	20.7	22.4	24.0	25.5	26.9	28.2	29.5	30.9	32.1	33.2	34.3	35.4	36.6	37.7	38.7	39.6	41.6
64	15.1	17.3	19.2	21.0	22.7	24.3	25.9	27.3	28.6	29.9	31.3	32.6	33.7	34.8	35.9	37.1	38.2	39.2	40.2	42.2
66	15.3	17.5	19.5	21.2	23.0	24.6	26.2	27.7	29.0	30.3	31.7	33.0	34.2	35.3	36.4	37.6	38.7	39.8	40.8	42.8

[a]Additional sizes: $4 \times 5 = 4.9$; $4 \times 6 = 5.4$; $4 \times 7 = 5.8$; $5 \times 5 = 5.5$; $5 \times 6 = 6.3$; $5 \times 7 = 6.5$.

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION—(Concluded)

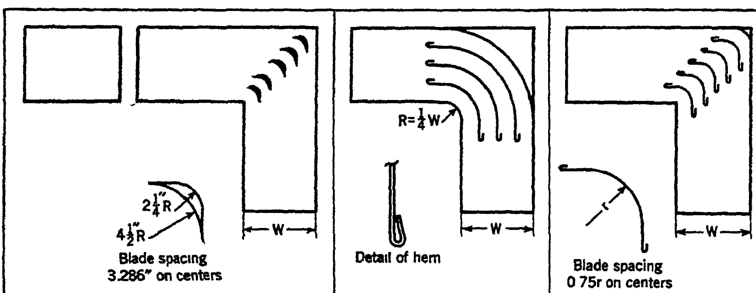
Side Rectangular Duct	26	28	30	32	34	36	38	40	42	44	46	48
26	28.6											
28	29.7	30.8										
30	30.7	31.9	33.0									
32	31.7	32.9	34.1	35.2								
34	32.7	33.9	35.1	36.3	37.4							
36	33.7	34.9	36.1	37.3	38.5	39.6						
38	34.6	35.9	37.1	38.4	39.5	40.7	41.8					
40	35.3	36.7	38.0	39.3	40.5	41.7	42.9	44.0				
42	36.0	37.6	39.0	40.3	41.5	42.7	44.0	45.1	46.2			
44	36.9	38.5	39.9	41.2	42.5	43.7	44.9	46.1	47.2	48.4		
46	37.8	39.3	40.8	42.2	43.5	44.8	46.0	47.2	48.4	49.5	50.6	
48	38.5	40.0	41.5	43.0	44.4	45.6	46.9	48.1	49.3	50.5	51.6	52.8
50	39.2	40.8	42.3	43.8	45.2	46.5	47.9	49.1	50.4	51.6	52.9	54.0
52	40.0	41.6	43.1	44.7	46.1	47.5	48.9	50.1	51.3	52.5	53.8	55.0
54	40.7	42.4	44.0	45.5	47.0	48.4	49.9	51.1	52.3	53.5	54.8	56.0
56	41.3	43.0	44.6	46.2	47.7	49.1	50.6	52.0	53.3	54.6	55.9	57.0
58	42.1	43.8	45.4	47.0	48.5	50.0	51.5	52.9	54.2	55.5	56.8	58.0
60	42.7	44.5	46.1	47.8	49.3	50.9	52.3	53.8	55.0	56.4	57.7	58.9
62	43.4	45.1	46.8	48.4	50.0	51.7	53.0	54.5	55.9	57.2	58.5	59.7
64	44.0	45.8	47.5	49.2	50.9	52.4	53.9	55.4	56.8	58.1	59.4	60.6
66	44.7	46.5	48.2	50.0	51.6	53.1	54.7	56.2	57.6	59.1	60.4	61.6
68	45.3	47.2	48.9	50.7	52.2	53.8	55.5	56.9	58.4	59.9	61.3	62.6
70	46.0	47.8	49.5	51.3	52.9	54.5	56.2	57.7	59.1	60.6	62.1	63.5
72	46.5	48.4	50.1	51.9	53.7	55.4	57.0	58.7	60.0	61.3	63.0	64.5

Side Rectangular Duct	50	54	60	66	72	78	84	88
50	55.0							
52	56.1							
54	57.2	59.4						
56	58.3	60.5						
58	59.3	61.6						
60	60.3	62.7	66.0					
62	61.3	63.7	67.1					
64	62.2	64.7	68.2					
66	63.2	65.7	69.3	72.6				
68	64.1	66.6	70.3	73.7				
70	65.0	67.6	71.3	74.8				
72	65.9	68.5	72.3	75.9	79.2			
74	66.8	69.4	73.3	76.9	80.3			
76	67.6	70.3	74.2	77.9	81.4			
78	68.4	71.2	75.2	78.9	82.5	85.8		
80	69.2	72.1	76.1	79.9	83.6	86.9		
82	70.1	73.0	77.1	80.9	84.6	88.0		
84	70.9	73.8	78.0	81.9	85.6	89.1	92.4	
86	71.7	74.6	78.9	82.9	86.6	90.2	93.5	
88	72.5	75.5	79.8	83.9	87.5	91.2	94.6	96.8
90	73.3	76.3	80.6	84.7	88.5	92.2	95.7	97.9
92	74.1	77.1	81.4	85.6	89.5	93.2	96.7	99.0
94	74.8	77.8	82.2	86.5	90.4	94.2	97.8	100.1
96	75.5	78.7	83.0	87.4	91.3	95.2	98.8	101.2

diameters[4]. Elbows having a radius of more than three diameters show a slightly increased resistance due to the increased length of pipe but, when used, they reduce the overall resistance of the system and therefore should not be avoided.

Where space conditions necessitate the use of short radius or square throat elbows in rectangular duct work, turning vanes should be used to reduce the pressure losses. Rough or raw edges on the vanes should be avoided to prevent objectionable noise. Three typical types of vanes are shown in Fig. 6 which gives the approximate number of duct widths recommended to be used in estimating the resistance of each type.

The pressure loss through elbows of less than 90 deg may be assumed to be directly proportional to the ratio of the angle through which the turn is made. The resistance will vary widely for the large degree turns depending upon the aspect ratio and the length of straight pipe between the elbows, but for practical purposes, it may be assumed that the ratio remains proportional to the angle through which the turn is made.



Number of duct widths for equivalent resistance	13xW	12.4xW	28xW

FIG. 6. DESIGN AND CORRESPONDING LOSS OF PRESSURE IN ELBOWS WITH TURNING VANES

Reverse 90 deg elbow turns should be avoided wherever possible but, where used, the friction of the elbows indicated in Fig. 5 should be doubled for the second elbow.

PROPORTIONING THE LOSSES

The entrance loss through the outside air intake louvers will vary with the design of the louvers and method of connection to the system. The louvers and connecting duct will have a friction resistance of from 0.25 to 1.00 times the velocity pressure. Therefore, the total entrance loss will vary from 1.25 h_v to 2.00 h_v. Common practice is to use 1.5 h_v for a 75 per cent free area louver with connecting duct having 15 deg tapered sides. Wherever air passes through a plenum space having a negligible velocity, allowance must be made for the loss in velocity head. This may be taken as the velocity head corresponding to the difference in velocities in the plenum and the duct. Where the ducts are very smooth with long transformation fittings, a regain in static pressure is sometimes allowed,

[4]Pressure Losses in Rectangular Elbows, by R. D. Madison and J. R. Parker (*Heating, Piping and Air Conditioning*, July, p. 365, August, p. 427, September, p. 483, 1936).

but generally ordinary construction does not warrant a consideration of this factor, and it is customary to neglect it. When it is allowed, the regain is estimated at one-half the difference between the velocity pressure at the fan outlet and at the last run of pipe.

Other losses of pressure occur through the heating units, at the air washer and at air filters. In ordinary practice in ventilation work it is usual to keep the sum of the duct losses one-third to one-half and the loss through the other units at less than one-half of the static pressure. The remainder is then available for producing velocity. In the design of an ideal duct system, all factors should be taken into consideration and the air velocities proportioned so that the resistance will be practically equal in all ducts regardless of length.

DUCT SIZES

Ducts and flues for gravity circulation must be sized so that the friction loss will not exceed 50 per cent of the available aspirating effect due to the temperature and height of the column of heated air. Duct systems for mechanical circulation may be sized so as to have much higher pressure losses than gravity systems. The total pressure of these systems is limited to the available pressure from the fan used.

General Rules

The general rules to be followed in the design of a duct system are enumerated herewith:

1. The air should be conveyed as directly as possible at reasonable velocities to obtain the results desired with greatest economy of power, material and space.
2. Sharp elbows and bends should be avoided unless turning vanes are used.
3. Transformation pieces should be made as long as possible. The angle between the sides and axis of the duct should never exceed 30 deg and where possible, 15 deg should be made the maximum.
4. Especial care should be taken to maintain a true cross-section and not to restrict the air flow either in transformation pieces or in elbows.
5. Rectangular ducts or flues should be made as nearly square as possible. Good practice limits the ratio between the long side and the short side to 3 to 1. In no case should this ratio exceed 10 to 1.
6. Wherever possible, ducts should be constructed of smooth material such as sheet metal. Where masonry ducts are used, proper allowance for the surface coefficient should be made.
7. The use of furred spaces, spaces between joists, etc., should be avoided unless lined with sheet metal.

Procedure for Duct Design

The general procedure for designing a duct system is outlined in the several items listed herewith:

1. Study the plan of the building and draw in roughly the most convenient system of ducts, taking cognizance of the building construction, avoiding all obstructions in steel work and equipment, and at the same time maintaining a simple design.
2. Arrange the positions of duct outlets to insure the proper distribution of air.
3. Divide the building into zones and proportion the volume of air necessary for each zone.
4. Determine the size of each outlet, based on the volume as obtained in the preceding paragraph, for the proper outlet velocity and throw.

5. Calculate the sizes of all main and branch ducts by either of the following two methods:

a. Velocity Method. Arbitrarily fix the velocity in the various sections, reducing the velocity from the point of leaving the fan to the point of discharge to the room. In this case the pressure loss of each section of the duct is calculated separately and the total loss found by adding together the losses of the various sections of the continuous run.

b. Friction Pressure Loss Method. Proportion the duct for equal friction pressure loss per foot of length.

6. Calculate the friction for the duct offering the greatest resistance to the flow of air, which resistance represents the static pressure which must be maintained in the fan outlet or in the plenum space to insure distribution of air in the duct system. The duct having the greatest resistance will usually be that having the longest run, although not necessarily so.

Air Velocities

The air velocities given in Table 2 have been found to give satisfactory results in engineering practice. Where the higher velocities are used, the ducts should be cross-braced to prevent breathing, buckling or vibration. High velocities at one point in the system offset the effect of proper design in all other parts of the system; hence the importance of air velocities, elbow design, location of dampers, fan connections, grille and register approach connections, and similar attention to details. For industrial buildings, noise is seldom given much consideration, and main duct velocities as high as 2800 or 3000 fpm are sometimes used but, when these velocities are used, due consideration should be given to duct design, resistance pressure, fan efficiencies and motor horsepower. For department stores and similar buildings, 2000 to 2200 fpm are sometimes used in main ducts where noise is not objectionable and space conditions warrant it. Wherever velocities higher than those shown in Table 2 are used, it is essential that the ducts should be of heavier gages, have additional bracing and be carefully constructed for a minimum resistance.

Where the high velocity diffusing outlets are used, the duct velocity should not be less than the throat velocity of the diffusers, as dynamic losses occur wherever velocities are stepped up or down. One recent trend in grille design is toward the use of much higher grille and branch duct velocities. Some installations have been made with velocities as high as 1600 fpm in branches and through the net area of grilles, but many of these have proven unsatisfactory because of noise and drafts.

Grille manufacturers publish selection tables which size the grilles for volume of air, temperature differential and distance of throw. In following these tables, maximums should be avoided and the manner in which the duct connects to the grille should be given careful consideration. Most of the selection tables are based on straight approach to the grille. Elbow connections to supply grilles should be provided with turning vanes to equalize the face velocity. See Chapter 31 for a discussion of grilles.

Fan outlet velocities are discussed in Chapter 30 and will not be dealt with here except to indicate that fan noises should be given proper consideration.

Main Trunk Ducts

Main trunk ducts with branches are commonly used to convey the air from the fan to the grille or register outlets in preference to individual

ducts from the fan to these outlets. The velocities in these ducts and branches vary according to the nature of the installation and the degree of quietness desired. The recommended velocities in Table 2, with good construction, should give satisfactory results. The maximum velocities indicated should not be used except in areas where noise is not a deciding factor.

Velocity Method

The velocity method of designing a duct system involves arbitrarily selecting velocities at various sections of the duct system with the highest velocities generally chosen at the fan and progressive lower velocities toward the duct openings to the room. To find the total static pressure against which the fan must operate, the static pressure loss of each section must be calculated separately and the total loss found by adding the individual losses of the various sections of the run having the highest resistance. Usually this is the longest run but in some cases a shorter run may have more elbows, transformations, booster heaters, etc., which will cause it to have a higher resistance pressure. This method requires judgment and experience in choosing the proper velocities to approach equal friction for all lengths of run but many engineers believe that the velocity method is handier to use than other methods and will give satisfactory results for most practical applications. The air velocities given earlier in this chapter are helpful in choosing proper velocities. Adjustable dampers or splitters are used to regulate air quantities delivered.

Equal Friction Method

The equal friction method of design is sometimes preferred because it does not require nearly so much judgment and experience in selecting the proper velocities in the various sections of a system. The usual procedure

TABLE 2. RECOMMENDED AND MAXIMUM DUCT VELOCITIES

Designation	Recommended Velocities, fpm			Maximum Velocities, fpm		
	Residences	Schools, Theaters, Public Buildings	Industrial Buildings	Residences	Schools, Theaters, Public Buildings	Industrial Buildings
Outside Air Intakes[a]	700	800	1000	800	900	1200
Filters[a]	250	300	350	300	350	350
Heating Coils[a]	450	500	600	500	600	700
Air Washers	500	500	500	500	500	500
Suction Connections	700	800	1000	900	1000	1400
Fan Outlets	1000–1600	1300–2000	1600–2400	1700	1500–2200	1700–2800
Main Ducts	700–900	1000–1300	1200–1800	800–1000	1100–1400	1300–2000
Branch Ducts	600	600–900	800–1000	700	800–1000	1000–1200
Branch Risers	500	600–700	800	650	800–900	1000

[a]These velocities are for total face area, not the net free area.

in this method of design is to select the main duct velocity to be consistent with good practice from a standpoint of noise for a particular type of building. This velocity should be less than the fan outlet velocity. All main ducts and branch ducts are sized for equal friction by the use of Fig. 2 and Table 1 or Fig. 4.

In cases where the fan or factory assembled air conditioning unit has a limited external resistance, it is necessary to divide the available resistance by the total equivalent length of the longest or most complicated run of duct to determine the resistance per 100 ft and then to size all ducts at this resistance value, which will automatically determine the duct velocities and give the desired total duct resistance. A further refinement

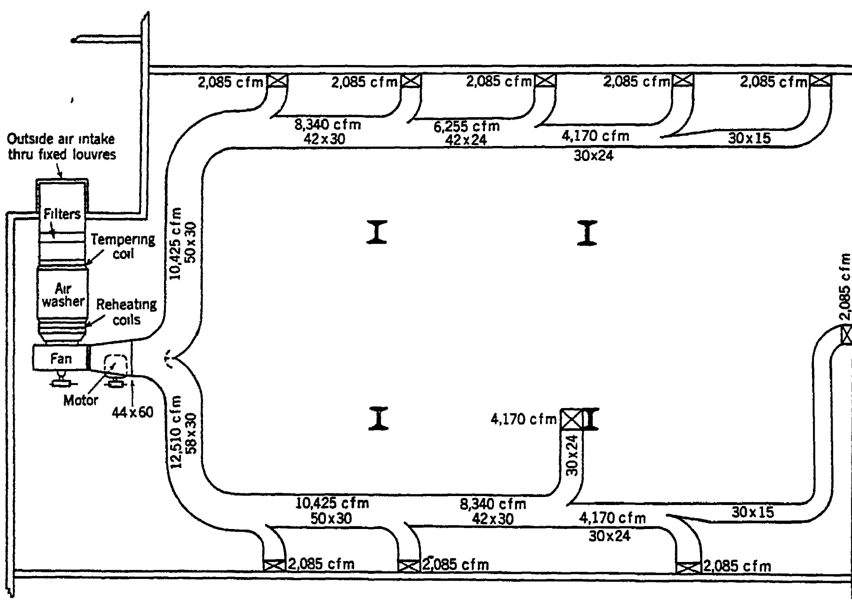


FIG. 7. TYPICAL LAYOUT OF AIR DISTRIBUTION SYSTEM

which is sometimes used in large systems is to size each branch duct so that it has a resistance equal to the resistance of the main system at the point of juncture. Even when this refinement is added, regulating dampers are recommended in each branch.

After the duct system is designed the frictional resistance is calculated and tabulated together with the resistance of all component parts. The fan is then selected for the required volume of air, static pressure and outlet velocity.

Example 2. Fig. 7 shows a typical layout of an air distribution system which is applicable for ventilation of hotel dining rooms and offices. The volume of air in cubic feet per minute for the room is determined on the basis of the number of air changes per hour required. In the example shown, the room ventilated is a hotel dining room 135 ft x 85 ft x 15 ft. A 7½-min air change (8 air changes per hour) is assumed for proper ventilation, giving 22,935 cfm as the air required.

The free area of the outdoor air inlet is based on a velocity of 1000 fpm or 22,395 ÷ 1000 = 22.94 sq ft. The main duct velocity selected from Table 2 is 1250 fpm which gives a main duct area of 22,935 ÷ 1250 = 18.354 sq ft (60 × 44 in.). From Table 1 a 60 × 44 in. duct is approximately equivalent to 56 in. diameter.

Referring to Fig. 2, a volume of 22,935 cfm through a 56 in. diameter duct gives a velocity of 1340 fpm and a resistance of 0.028 in. per 100 ft. The amount of air to be handled by each section of pipe is shown in Fig. 7, and by locating each of these values on the 0.028 in. friction line, the round pipe sizes are obtained and then, referring to Table 1, the equivalent rectangular sizes are selected as shown in Table 3.

The pressure at the outlets nearest the fan will be greater than at the pipes farther along the run so that the former will tend to deliver more than the calculated amount of air. To remedy this condition, volume regulating dampers should be located at the base of each riser, or in each branch duct, and adjusted for proper distribution. At points where branches leave the main it may be advisable, depending upon the nature of the installation, to install adjustable splitters similar to that shown in Fig. 7 where the main duct divides into the 58 x 30 in. and 50 x 30 in. branches.

TABLE 3. PIPE SIZES FOR EXAMPLE 2[a]

VOLUME OF AIR (CFM)	DIAMETER OF PIPE (INCHES)	EQUIVALENT SIZE OF RECTANGULAR DUCT (INCHES)
22,935	56	60 x 44
12,510	45	58 x 30
10,425	42	50 x 30
8,340	39	42 x 30
6,255	35	42 x 24
4,170	29.5	30 x 24
2,085	23	30 x 15

[a]Velocity through grilles (not shown) to be approximately 300 fpm.

Resistance Losses for the System

(1) Outdoor air intake, 1000 fpm velocity (1.5 heads × 0.0625) .... 0.094 in.

(2) Filters (from manufacturer's tables) .... 0.250 in.

(3) Tempering coil loss (from manufacturer's tables) .... 0.074 in.

(4) Air washer loss (from manufacturer's tables) .... 0.250 in.

(5) Reheating coil loss (from manufacturer's tables) .... 0.083 in.

(6) Duct resistance:

The longest run is .... 150 ft

Two, 58 x 30 in. elbows (150% ratio) $\frac{2 \times 13 \times 58}{12}$ .... = 126 ft

Two, 30 x 15 in. elbows (150% ratio) $\frac{2 \times 13 \times 30}{12}$ .... = 65 ft

Three, 15 x 30 in. elbows (75% ratio) $\frac{3 \times 35 \times 15}{12}$ .... = 131 ft

Total equivalent run .... 472 ft

472 ft at 0.028 in. per 100 ft .... 0.132 in.

(7) Allowance for damper adjustment, 25% of 0.132 .... 0.033 in.

(8) Supply grille resistance (from manufacturer's tables) .... 0.036 in.

Total static pressure loss of system .... 0.952 in.

The fan is selected from the manufacturer's ratings to deliver 22,935 cfm at a static pressure of 0.952 in. as outlined in Chapter 30.

TABLE 4. RECOMMENDED SHEET METAL GAGES FOR DUCTS[a]

U. S. STD. GAGE	ROUND DUCTS DIAMETER, INCHES	RECTANGULAR DUCTS Maximum Side, Inches	Type of Joint Connections	Bracing
26	Up to 18	Up to 12	*S* or Drive Slips	None
24	19 to 30	13 to 24	*S* or Drive Slips	None
24		25 to 30	*S* or Drive Slips	1 in. Angle 4 ft from Slips
22	31 to 45	31 to 48	Bar or Drive Slips	1¼ in. Angle 4 ft on Centers
20	46 to 60	49 to 60	1¼ in. Angle Bar Slips or 1¼ in. Angle Connections	1¼ in. Angle 2 ft 8 in. on Centers
18	61 and up	61 to 90	1½ in. Angle Connections	1½ in. Angle 2 ft 8 in. on Centers
18		91 and up	2 in. Angle Connections	2 in. Angle 2 ft 8 in. on Centers

[a]If flat sides are not cross-broken two gages heavier material should be used.

TABLE 5. WEIGHTS OF SHEET METAL USED FOR DUCT CONSTRUCTION

U. S. STD. GAGE	BLACK SHEETS Approximate Thickness, In. Steel	Iron	Weight Per Square Foot Ounces	Pounds	GALVANIZED SHEETS[b] Approximate Thickness, In. Steel	Iron	Weight Per Square Foot Ounces	Pounds
30	0.0123	0.0125	8	0.500	0.0163	0.0165	10.5	0.656
28	0.0153	0.0156	10	0.625	0.0193	0.0196	12.5	0.781
26	0.0184	0.0188	12	0.750	0.0224	0.0228	14.5	0.906
24	0.0245	0.0250	16	1.000	0.0285	0.0290	18.5	1.156
22	0.0306	0.0313	20	1.250	0.0346	0.0353	22.5	1.406
20	0.0368	0.0375	24	1.500	0.0408	0.0415	26.5	1.656
18	0.0490	0.0500	32	2.000	0.0530	0.0540	34.5	2.156
16	0.0613	0.0625	40	2.500	0.0653	0.0665	42.5	2.656
14	0.0766	0.0781	50	3.125	0.0806	0.0821	52.5	3.281
12	0.1072	0.1094	70	4.375	0.1112	0.1134	72.5	4.531
11	0.1225	0.1250	80	5.000	0.1265	0.1290	82.5	5.156
10	0.1379	0.1406	90	5.625	0.1419	0.1446	92.5	5.781

[b]Galvanized sheets are gaged before galvanizing and are therefore approximately 0.004 in. thicker.

TABLE 6. WEIGHTS AND THICKNESSES OF STANDARD COPPER SHEETS[c]
Rolled to Weight

WEIGHT PER SQUARE FOOT Ounces	Pounds	THICKNESS, INCHES Decimal Equivalent	Nearest Fraction	NEAREST GAGE NO. B. & S.	Stubs	U. S. STD.
10	0.625	0.0135	1/64	27	29	29
12	0.750	0.0162	1/64	26	27	28
14	0.875	0.0189	1/64	25	26	26
16	1.000	0.0216	1/32	23	24	25
18	1.125	0.0243	1/32	22	23	24
20	1.250	0.0270	1/32	21	22	23
24	1.500	0.0324	1/32	20	21	22
28	1.750	0.0378	1/32	19	20	20
32	2.000	0.0432	3/64	17	19	19
36	2.250	0.0486	3/64	16	18	18
40	2.500	0.0540	3/64	15	17	17
44	2.750	0.0594	1/16	15	17	17
48	3.000	0.0648	1/16	14	16	16
56	3.500	0.0756	5/64	13	15	14
64	4.000	0.0864	5/64	11	14	13

[c]Variations from these weights must be expected in practice.

Example 3. If the rooms and offices of the hotel building of Example 2 are to be served from a manufactured unit with a capacity of 22,935 cfm against an external resistance of 0.35 in., the known resistances are calculated as:

(1)	Outdoor air inlet	0.094 in.
(2)	Allowance for damper adjustment	0.033 in.
(3)	Supply grille resistance (from manufacturer's tables)	0.036 in.
	Total known resistance	0.163 in.

Subtracting this from the total available resistance: 0.35 in. − 0.163 in. = 0.187 in. available for duct resistance.

Known length of run	150 ft
The duct width is then estimated for the following elbow calculations:	
Four 150% ratio elbows, 4 x 13 x 3.5 ft	182 ft
Three 75% ratio elbows, 3 x 35 x 1.5 ft	158 ft
Total estimated length	490 ft

The duct friction per 100 ft is then 0.187 ÷ 4.90 = 0.0382 in. and the mains and branches are sized from the 0.038 in. friction line in Fig. 2.

If it is desired to size each branch for equal resistance, the total resistance back to the point of juncture is calculated and the branch is then sized in a manner similar to that outlined in Example 3.

SOUND CONTROL

Frequently the problem of sound prevention in a heating, ventilating or air conditioning system imposes more severe restrictions than the prevention of excessive pressure drop. Tendencies toward higher duct velocities have produced noise control problems which require consideration of enumerable factors in air duct design. Naturally some types of occupancy and application permit relatively higher sound levels to be maintained than others, but the design trend is progressively directed towards noise reduction wherever possible. Sound absorbent materials have been successfully applied to duct construction to reduce noise. The basis used for the selection of the proper amounts of absorbent materials will be found in Chapter 33.

DUCT CONSTRUCTION DETAILS

Straight sections of round duct are usually formed by rolling the sheets to the proper radius and grooving the longitudinal seam. Rectangular ducts are generally constructed by breaking the corners and grooving the longitudinal seam, although some fabricators still use the standing seam due to lack of equipment. Elbows and transformation sections are generally formed with *Pittsburgh* corner seams because this seam is easier to lock in place than the double seam, but complicated fittings such as double compounded elbows are usually constructed with double seam corners.

The construction of these various seams as well as the types of girth connections are shown in Fig. 8. The application of the various slips and connections are outlined in Table 4. The end slip may be used wherever *S* slips are recommended. Where drive slips are used the end slip may be applied on the narrow side of the duct and only the drive slips on the

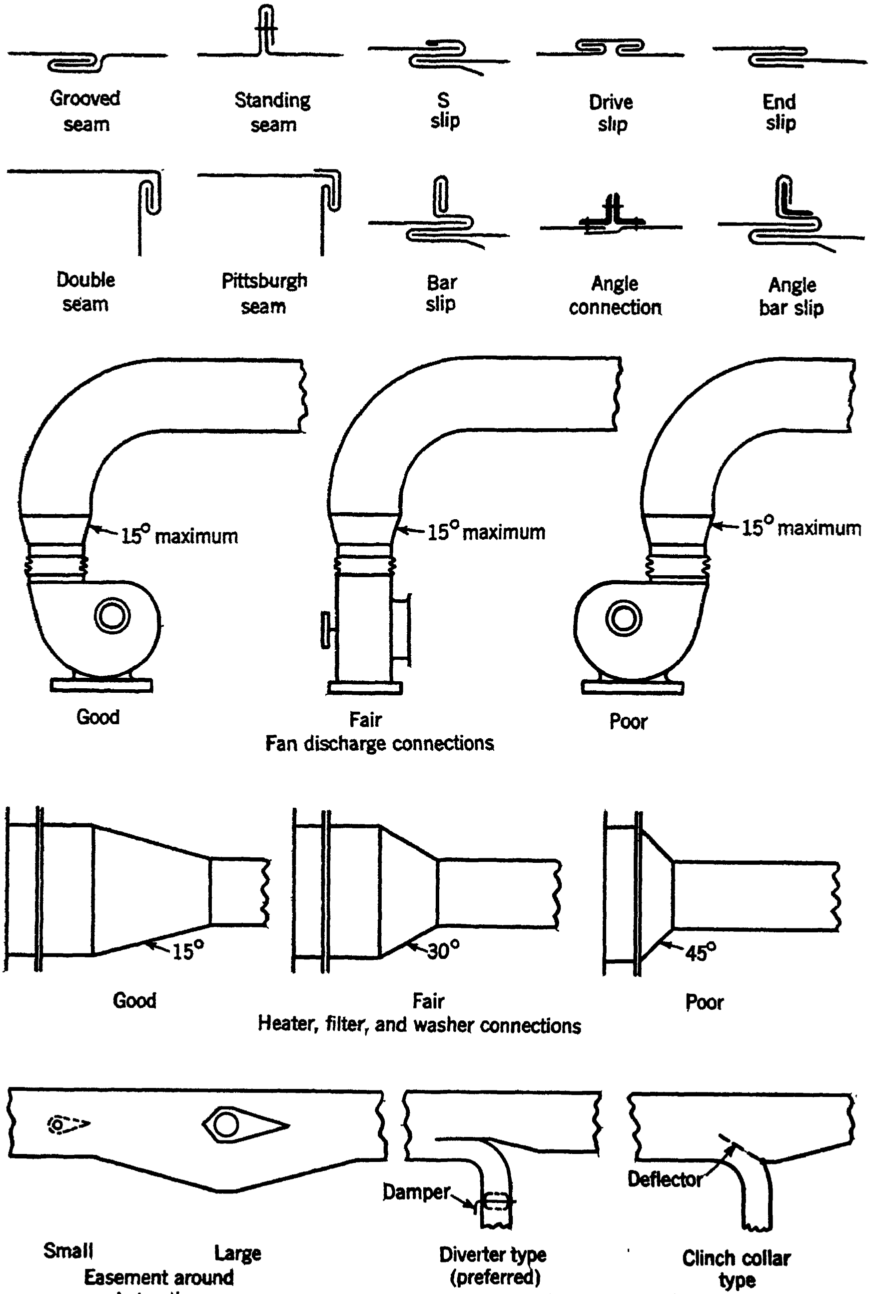


Fig. 8. Sheet Metal Duct and Arrangement Details

maximum side. Ducts 25 to 30 in. in size should be reinforced between the joints, but not necessarily at the joint. Ducts 31 in. and up should be reinforced at the joint and between the joints; if drive slips are used the angles are usually riveted to the duct about 2 in. from the slips. It is good practice to cross-break or kink all flat surfaces to prevent vibration or buckling due to the air flow and accompanying variations in internal

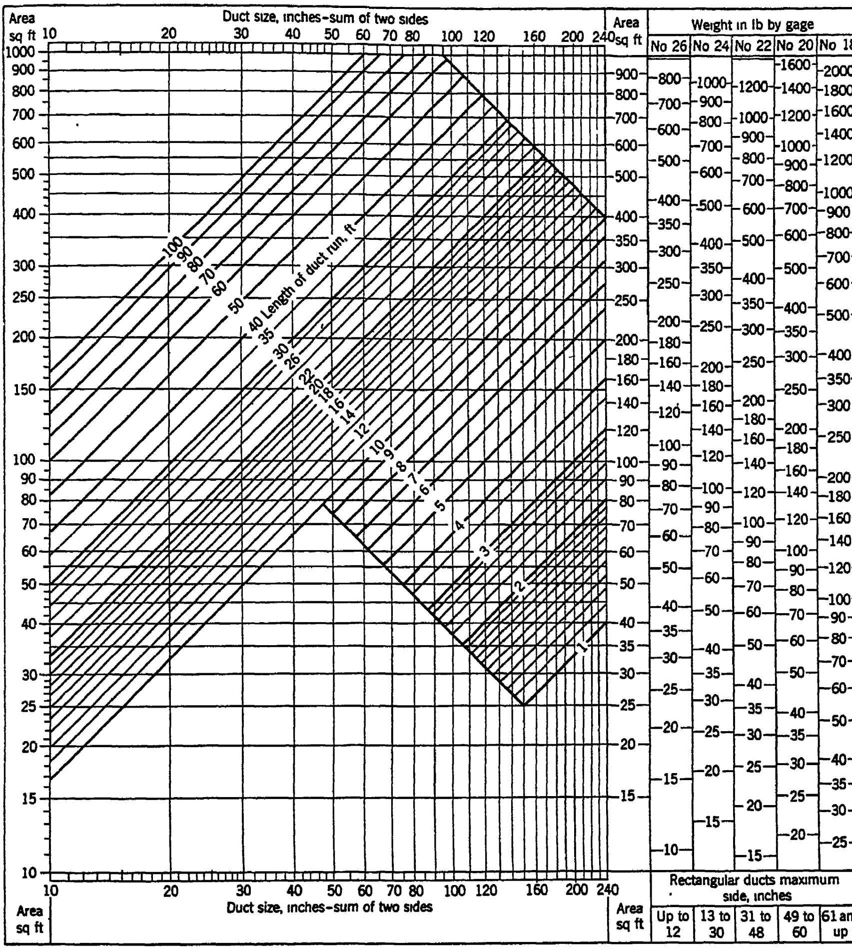


FIG. 9. AREA AND WEIGHT OF RECTANGULAR SHEET METAL DUCTS

pressure. Round ducts are sometimes swedged 1.5 in. from the ends so that the larger end will butt against the swedge and are held in place with sheet metal screws. Where swedges are not used it is general practice to paste the joint with asbestos paper to insure a tight joint.

The construction of elbows and changes of shape cannot be definitely outlined because of the varied conditions encountered in the field, but in

general long radius elbows and gradual changes in shape tend to maintain uniform velocities accompanied by decreased turbulence, lower resistance and a minimum of noise.

Heavy canvas connections are recommended on both the inlet and outlet to all fans. The fan discharge connections shown in Fig. 8 are marked *good*, *fair*, and *poor* in the order of the amount of turbulence produced. An inspection of the heater connections shown in Fig. 8 will readily show that uniform velocity through the heater cannot be expected in the diagram noted *poor*. When obstructions cannot be avoided, the duct area should never be decreased more than 10 per cent and then a *streamlined* collar should be used. Larger obstructions require an increase in the duct size in order to maintain as nearly uniform velocity as possible. Branch take-offs should always be arranged to cut or slice into the air stream in order to reduce as far as possible the losses in velocity head.

The recommended gages for sheet metal duct construction are given in Table 4. Weights of sheet metal per square foot of surface for different gages are given in Table 5. The weights of various gages and the areas for any length of run of rectangular sheet metal ducts may also be determined from Fig. 9. The bottom scale represents the sum of the two sides of the duct and the oblique lines give the length of run in feet. Proceeding horizontally to the right from the intersection of vertical and oblique lines on the chart, the area of the duct may be determined in the first vertical scale. The scales to the right give the weights of the duct run for different gages of metal. In calculating the weights of duct, it is considered good practice to allow 20 per cent additional for weights of joints and bracings. Various weights and thicknesses of standard copper sheets will be found in Table 6.

REFERENCES

Method of Determining Rectangular Equivalents and Weights of Ducts, by Peter Franck (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1940).

The Flow of Liquids, by W. H. McAdams (*Refrigerating Engineering*, February, 1925, p. 279).

A Study of the Data on the Flow of Fluids in Pipes, by Emory Kemler (*A.S.M.E. Transactions*, Hydraulics Section, August 31, 1933, p. 7).

Air Conditioning and Engineering, American Blower Corp.

Fan Engineering, Buffalo Forge Co.

Heat Power Engineering, by Barnard, Ellenwood, and Hirshfeld, Part III.

Mechanical Engineers' Handbook, by Lionel S. Marks, McGraw-Hill Book Co.

Chapter 33

SOUND CONTROL

Decibel Defined, Apparatus for Measuring Noise, Problem of Sound Control, Acceptable Noise Levels, Fan Noise, Attenuation in Duct System, Absorbers, Grille Noise, Cross Transmission of Noise Between Rooms, Controlling Vibration from Machine Mountings

IN ventilating and air conditioning a building or a room, the effect of the mechanical system employed must be considered on the acoustics of the space conditioned. It is important to consider also that the use of air conditioning often permits keeping the windows closed, thus giving relief from certain external noises, but at the same time increasing the necessity of providing adequate sound control.

It is not assumed that the ventilating and air conditioning engineer will attempt to improve the acoustics of the space that is being conditioned, but the designer should have at least enough fundamental knowledge of the acoustical effects of the system which is being designed to be sure that no damaging effects occur to the existing acoustical properties. It is assumed that in a given space the architect and acoustical engineer have produced a room or rooms which are satisfactory for speech, music, or other uses. The ventilating engineer's sole function is to ventilate and air condition these rooms properly so that they will be physically comfortable without adding any acoustical hazards.

UNIT OF NOISE MEASUREMENT

By a recently adopted international standard, two terms are used for noise measurement. The *decibel* (db) is the physical unit for expressing intensity or pressure levels. The *phon* is the unit of loudness level. The loudness level, in phons, of any sound is by definition equal to the intensity level in decibels of a thousand cycle tone which sounds equally loud. The decibel is defined by the relation $N = 10 \log_{10} \frac{I_i}{I_0}$, where N is the number of decibels by which the intensity flux I_i exceeds the intensity flux I_0. The intensity flux is the measure of the energy contained in a sound wave and is defined in terms of micro-watts per square centimeter of wave front in a freely traveling plane wave. It is usually more convenient to select an arbitary reference intensity for I_0 and express all other intensities in terms of decibels above that level. For this purpose a reference intensity of 10^{-16} watts per square centimeter has been

selected. This intensity is slightly less than the threshold of audibility for the average ear at a frequency of 1,000 cycles per second. This reference level also corresponds to a pressure of 0.0002 dynes per square centimeter.

A stated sound level in decibels, unless otherwise defined, will thus be related to a threshold of 10^{-16} watts. For example, a level of 60 db above this reference threshold is 10^{-10} watts. In a similar manner, when sound measurements are given in actual intensity or energy units, they can be converted to decibels by this relation.

Since the decibel is a ratio, it can only be employed when related to a reference threshold level as given. Noise levels, which vary with frequency as well as intensity, must not only be related to this reference threshold level, but also to a reference frequency, which is taken as 1000 cycles. These terms and procedures may be found in Tentative Standards[1] published by the *American Standards Association.*

APPARATUS FOR MEASURING NOISE

Since the relative loudness to the ear, rather than the actual physical intensity, is the quantity in which engineers are usually interested, it has been found necessary to allow for the varying sensitivity of the ear at different frequencies in designing noise measuring equipment. The most satisfactory method of measuring noise is by means of a sound level meter which usually consists of a microphone, a high gain audio-amplifier, and a rectifying milliammeter which will read directly in decibels. This meter is calibrated to give readings above the standard reference level and usually contains a weighing network to make it less sensitive at those frequencies where the ear is less sensitive. For complete specifications relative to the approved type of sound level meters refer to the information[2] published by the *American Standards Association.*

GENERAL PROBLEM OF SOUND CONTROL

As previously stated, the problem confronting the air conditioning engineer is that of designing a system which will operate without increasing the noise level in the conditioned space. To be sure that this is accomplished, it is necessary:

a. To determine the noise level existing without the equipment.

b. To ascertain the noise level which would exist if the equipment were installed without sound control.

c. To provide as a part of the installation sufficient sound control appliances to reduce the noise level substantially to that found in (*a*).

To accomplish this the engineer should have information of three kinds:

1. A knowledge of the noise levels currently considered acceptable in various rooms in order that he may have a basis on which to proceed.

2. A knowledge of the nature and intensity of the noise created by the various parts of the equipment.

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[1]American Tentative Standards for Noise Measurement, *American Standards Association.*

[2]American Tentative Standards for Sound Level Meters for Measurement of Noise and Other Sounds, *American Standards Association.*

3. A knowledge of how, when necessary, to vary and control the noise level between the equipment and the conditioned space.

In addition, the engineer should have information enabling him to deal with noises which may be transmitted by the duct system from one conditioned space to another, or from an outside space to the conditioned space.

While the general problem may be logically outlined and the items of knowledge necessary to its solution can be listed, the available information at present is lacking in certain respects. However, attention may be directed to that information which is currently available, and a solution of the noise problem, based on these data, may be outlined.

NOISE CREATED BY EQUIPMENT

Information concerning the noise levels created by ventilating and air conditioning equipment such as fans, motors, air washers, and similar items is not yet on a basis which permits tabular presentation although certain manufacturers are prepared to offer such data and do state the noise producing properties of their products.

Absence of this information makes it necessary to resort to indirect means in solving certain problems and also prevents a direct, logical solution.

KINDS OF NOISE

To solve a sound problem of this type it is desirable to consider separately the several means by which noise reaches the room. This avoids to some extent the necessity of knowing the noise level at the source, and instead, places the emphasis on ascertaining the level at the point where the sound enters the room.

The noise introduced into a room or building by ventilating or air conditioning equipment may be divided into two kinds depending on how it reaches the room as:

1. Noise transmitted through the ducts.
2. Noise transmitted through the building construction.

It is convenient to further sub-divide these two methods of delivery as:

1. Noise transmitted through the ducts.
   a. From equipment such as sprays, fans, etc.
   b. From outside, and transmitted through duct walls into air stream.
   c. From air current, including eddying noises.
   d. Cross talk and cross noises between rooms connected by the same duct system.
   e. Noise produced by the grilles.
2. Noise transmitted through the building construction.
   a. From machine mountings as vibration.
   b. From equipment through room wall surfaces.

The next step in the solution of this problem is to present data and discuss methods whereby solutions to the noise problem can be obtained when the allowable room noise level and the path through which the noise reaches the room are known.

NOISE TRANSMITTED THROUGH DUCTS

Operation of an air distribution system results in the generation of noise which may be transmitted through the ducts to the ventilated or conditioned room. The transmission of this noise may be controlled by the proper application of sound absorptive material within the ducts. The application of the absorptive material is a problem in balancing the room noise level requirements against the intensity of the noise generated. The four steps in the problem are:

1. Determination of acceptable room noise level resulting from the operation of the equipment.

2. Determination of noise level generated by the equipment.

The difference between steps 1 and 2 in decibels is the overall noise reduction required between the equipment and the room. In the following discussion reduction of noise will be referred to as *attenuation* of noise.

3. Determination of the natural attenuation of the duct system.

4. Selection of the proper sound treatment for the duct system.

The difference in decibels between the overall attenuation required and the natural attenuation (3) is the additional sound attenuation to be obtained by absorptive materials installed in the duct system.

DESIGN ROOM NOISE LEVEL

Measurements of noise levels have been observed by several investigators in various rooms and locations. They are listed in Table 1. The values given were determined with the air conditioning or ventilation equipment not in operation, and with all windows and doors closed simulating the conditions of an actual installation.

This is an important consideration, for in offices or stores adjacent to busy thoroughfares the difference between the typical noise level in the space with the windows and doors open and closed may be as high as 10 db. Minimum, representative, and maximum levels are given for each type of space. The values are intended to give the variation with respect to location and not to time, and may be roughly classified by the following:

Minimum loudness refers to: Spaces of expensive construction, typified by double windows, carpeted floors, heavy upholstered furniture, or accoustically treated walls and ceilings.

Representative loudness refers to: Spaces of average construction and furnishings which are exposed to external noises typical of the locality in which the space is usually found.

Maximum loudness refers to: (1) Any space of inexpensive construction, and bare furnishings where noise is not an important factor. (2) Spaces in close proximity to very intense street traffic or to intense factory noise. (3) Any space containing machinery which is a constant source of noise, typewriters, adding machines, printing presses, etc.

In general, if the noise level in the space resulting only from the operation of the air conditioning equipment is equivalent to or less than the typical level given in Table 1, the installation will prove satisfactory. If the typical level and the equipment level are heard together the resultant level will be 3 db higher than either of them.

In some cases it is desirable to keep the equipment noise level in the

ventilated or conditioned room at such a value that it actually will not increase the noise level in the room to any measureable degree. This can be accomplished if the equipment noise at the room can be kept 10 db below the noise level shown in the table.

NOISE GENERATED BY FANS

Noise generated by fan wheels may be divided into two classifications, rotational noise and vortex noise. In ventilation and air conditioning work, where the maximum ratio between the fan tip speed and the

TABLE 1. TYPICAL NOISE LEVELS

Rooms	Noise Level in Decibels to be Anticipated		
	Min.	Representative	Max.
Sound Film Studios	10	14	20
Radio Broadcasting Studios	10	14	20
Planetarium	15	20	25
Residence, Apartments, etc.	33	40	48
Theaters, Legitimate	25	30	35
Theaters, Motion Picture	30	35	40
Auditoriums, Concert Halls, etc.	25	30	40
Churches	25	30	35
Executive Offices, Acoustically Treated Private Offices	30	38	45
Private Offices, Acoustically Untreated	35	43	50
General Offices	50	60	70
Hospitals	25	40	55
Class Rooms	30	35	45
Libraries, Museums, Art Galleries	30	40	45
Public Buildings, Court Houses, Post Offices, etc.	45	55	60
Small Stores	40	50	60
Upper Floors Department Stores	40	50	55
Stores, General, Including Main Floor Dept. Stores	50	60	70
Hotel Dining Rooms	40	50	60
Restaurants and Cafeterias	50	60	70
Banking Rooms	50	55	60
Factories	65	77	90
Office Machine Rooms	60	70	80
Vehicles			
Railroad Coach	60[a]	70	80
Pullman Car	55[a]	65	75
Automobile	50	65	80
Vehicular Tunnel	75	85	95
Airplane	80	85	100

[a]For train standing in station a level of about 45 db is the maximum which can ordinarily be tolerated.

velocity of sound is not greater than 0.12, vortex noise is by far the most important. The rotational noise may be described as that due to the thrust and torque implied to the air. Vortex noise is that due to the shedding of vortices from the blade and is dependent on the angle of attack, velocity, air turbulence, and blade shape. Vortex noise is due to pressure variations on the blade as a result of variations of air circulation. Given the noise level at the outlet or inlet of one type of fan construction under specific conditions of size, tip speed, and total pressure, noise levels

at other values of tip speed, total pressure, and size may be approximated by the following relationships.

1. With total pressure and size constant.

$$db \text{ (change)} = 10 \operatorname{Log}_{10} \left[\frac{(\text{Tip Speed})_{\text{new}}}{(\text{Tip Speed})_{\text{old}}} \right]^6 \tag{1}$$

2. With total pressure and tip speed constant.

$$db \text{ (change)} = 10 \operatorname{Log}_{10} \left(\frac{\text{Size}_{\text{new}}}{\text{Size}_{\text{old}}} \right)^2 \tag{2}$$

3. With size and tip speed constant.

$$db \text{ (change)} = f\,(\text{Total Pressure}_{\text{old}} - \text{Total Pressure}_{\text{new}}) \tag{3}$$

The factor f is a function of the fan type. For a centrifugal fan with backward curved blades $f = 9.6$. For a single inlet single width type of ventilating fan with backward curved blades, the noise level at the fan discharge or intake operating at 4000 fmp tip speed and total pressure of 1 in. may be approximately 65 db. The noise level of a double inlet width fan may be 3 db higher than a single width fan at similar conditions

Table 2. Attenuation in Straight Sheet Metal Duct Runs

Duct	Size, In.	Attenuation per Ft, db
Small	6 x 6	0.10
Medium	24 x 24	0.05
Large	72 x 72	0.01

Table 3. Attenuation of Elbows[a]

Elbow	Size, In.	Attenuation per Elbow, db
Very small	2 wide	3
Small	3 to 15	2
Medium	15 to 36	1.5
Large	36 +	1

[a]The attenuation in vaned elbows should be considered the same as in elbows having the same dimensions as the radius of curvature of the vanes. If the vanes are lined for the purpose of damping any vibrations in them, one third may be added to the above attenuation values.

Table 4. Attenuation at Duct Branches or Outlets

Ratio $\frac{\text{Branch Duct + Outlet Area}}{\text{Supply Duct Area}}$ or $\frac{\text{Sum of Branch Areas}}{\text{Supply Duct Area}}$	Attenuation per Transformation, db
1.00	0.0
1.20	0.8
1.35	1.3
1.50	1.8
1.75	2.5
2.00	3.0

TABLE 5. APPROXIMATE ATTENUATION BETWEEN GRILLES AND ROOM

Outlet Velocity fpm	Air Change Min.	Live Room[b] $\alpha^{a} = 0.05$ db	Medium Room[c] $\alpha = 0.15$ db	Dead Room[d] $\alpha = 0.25$ db
	5	11	16	18
	10	14	19	21
500	15	16	21	23
	20	17	22	24
	5	13	18	20
	10	16	21	23
750	15	18	23	25
	20	19	24	26
	5	14	19	21
	10	17	22	24
1000	15	19	24	26
	20	20	25	28
	5	15	20	22
	10	18	23	25
1250	15	20	25	27
	20	21	26	28

[a]Alpha (α) is the average absorption coefficient for the room.

[b]Live room average absorption coefficient 0.05. Bare wood or concrete floor—hard plaster walls and ceiling—minimum of furniture.

[c]Medium room average absorption coefficient 0.15. Carpeted floor, upholstered furniture, hard plaster walls and ceiling or bare room with acoustically treated ceiling.

[d]Dead room average absorption coefficient 0.25. Heavy carpeted floor. Walls and ceiling acoustically treated. Upholstered furniture.

of tip speed, total pressure, and size. For the same tip speed and size the noise level of a fan with forward curved blades is higher than for one with backward curved blades, however, the capacity of the fan with the forward curved blades will be greater.

NATURAL ATTENUATION IN THE DUCT SYSTEM

The natural attenuation existing in a duct system consists of three component parts which are described herewith.

Straight Sheet Metal Ducts

The attenuation of sound in straight sheet metal ducts is a function of the length, shape, and size of the duct. Attenuation values are given in Table 2. In general, this attenuation is so negligible except for long runs that it may be neglected for all practical purposes.

Elbows and Transformations

Due to reflective interference, attenuation will take place at elbows and transformations. The magnitude of the attenuation will depend on the size and abruptness of the elbow or transformation as shown in Table 3.

When the area of a duct increases, an attenuation of noise level takes place in the duct. In duct design practice the total area of the branch ducts is greater than the supply duct. Similarly with outlets, the area of the outlet plus the area of the duct after the outlet is greater than the

duct area before the outlet. Therefore in an outlet run, attenuation occurs in the duct as it passes each outlet. Table 4 gives the *db* reduction for various ratios of total branch duct and outlet area to supply duct area.

Grilles to Room

The large abrupt change in area between the grilles and the surfaces within a room results in an appreciable noise attenuation. This attenuation is a function of the total grille area (supply and return) and the total sound absorption of the room in *sabines*. (The sound absorption of a room in *sabines* is the summation of the products of each surface of the room measured in square feet multiplied by its corresponding absorption coefficient). The attenuation is given in Equation 4 as:

$$db \begin{pmatrix} \text{Attenuation between} \\ \text{grilles and room} \end{pmatrix} = 10 \log_{10} \frac{\text{Total Room Absorption in Sabines}}{\text{Total Grille Area}} \qquad (4)$$

Values in Table 5 approximate the attenuation for various rates of air change, and general types of room surfaces.

DUCT SOUND ABSORBERS

The difference between the required sound attenuation and the natural attenuation is that which must be supplied by the proper sound treatment of the ducts.

Selection of the Absorptive Material

When a sound wave impinges on the surface of a porous material, a vibrating motion is set up within the small pores of the material by the alternating sound waves. As the ratio of the cross sectional area of the pores to their interior surface is small, the resistance to the movement of air in the pores is large. This viscous resistance within the pores of the material converts a portion of the sound energy into heat. The decimal fraction representing the absorbed portion of the incident sound wave is called the absorption coefficient. Considerable absorption may also result, particularly in the low frequency range, from the flexural vibrations of the duct. In the selection (and application) of the absorptive material the several points should be considered.

1. For the absorption of the low frequencies the material should be at least 1 to 2 in. thick. Thin materials, particularly when mounted on hard solid surfaces, will absorb the high frequencies and reflect the low.
2. In order to take advantage of low frequency noise absorption by panel vibration, it is advisable to fasten the absorptive sheets to stripping so that the panels themselves may vibrate. However, the exact resonance characteristics of the panels and thus their absorption is so unpredictable that panel resonance cannot be relied upon for a specific value of attenuation.

Sound absorption material for ducts should meet the several requirements listed herewith:

1. High absorption at low frequencies[3].
2. Adequate strength to avoid breakage.

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[3]For coefficients of commercial sound absorbent materials see Bulletin *Acoustical Materials Association*, 919 No. Michigan Ave., Chicago, Ill.

3. Fire resistant—should comply with national and local code requirements.
4. Low moisture absorption.
5. Freedom from attack by bacteria and algae.
6. Low surface coefficient of friction.
7. Particles should not fray off at the higher design velocities.
8. Odor free when either dry or wet.

The discussion which follows covers the application and design of the various types of absorbers. For each absorber an attenuation formula or table is given which will give results as accurate as predictable under the present status of our knowledge. With every application the use of sound absorptive material should be considered in the dual function of insulation and sound absorption. It has been shown theoretically[4] that the reduction, in decibels per linear foot, of sound transmitted through a duct lined with sound absorbing material is related in a rather complicated manner to the size and shape of the duct, to the frequency of the sound, and to the sound absorbing characteristics of the lining. Experimental

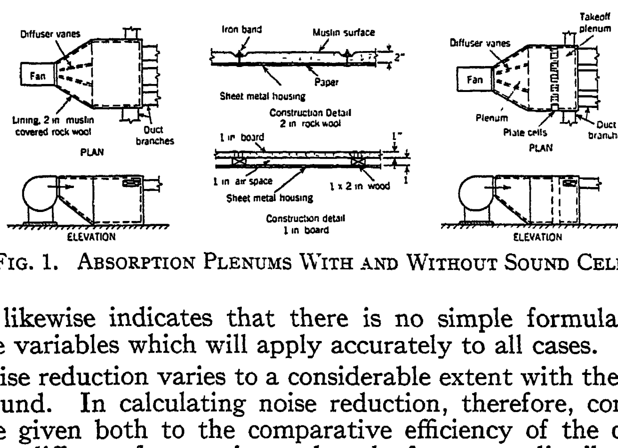

FIG. 1. ABSORPTION PLENUMS WITH AND WITHOUT SOUND CELLS

evidence likewise indicates that there is no simple formula involving the above variables which will apply accurately to all cases.

The noise reduction varies to a considerable extent with the frequency of the sound. In calculating noise reduction, therefore, consideration should be given both to the comparative efficiency of the duct lining material at different frequencies, and to the frequency distribution of the noise to be quieted. In the case of fan noise, it is recommended that calculations be based on the frequency 256 cycles, since most of the noise energy is in the region of this frequency. In quieting noise due to air turbulence and eddy currents, in which the high frequencies predominate, the frequency 1024 cycles should be used.

Plenum Absorption

In systems, where individual ducts are directed to a number of rooms and sound treatment is required in every duct, a sound absorption plenum on the fan discharge as shown in Fig. 1, will often prove the most economical arrangement. The absorption in the plenum may be approximated by Equation 5.

$$db\ (\text{attenuation}) = 10 \log_{10} \frac{\text{Plenum Absorption in Sabines}}{\text{Area Fan Discharge}} \tag{5}$$

[4]Sound Propagation in Ducts Lined with Absorbing Materials, by L. J. Sivian (*Journal Acoustical Society of America*, Vol. 9, p. 1937).

The area of the plenum should be at least ten times as great as the fan discharge area. The plenum should be lined with 2 in. of muslin covered rock wool blanket or 1 in. sound absorbing board preferably nailed to wood strips on the inside of the plenum. (With such a lining the plenum is particularly effective in reducing low frequency fan noise.) The absorption of the plenum in sabines is the sum of the products of each interior area of the plenum measured in square feet multiplied by its corresponding absorption coefficient.

Plate Cells

One of the most economical methods of applying sound absorbent material from the standpoint of both labor and material is the plate cell. The plate cell illustrated in Fig. 2 consists of ½ or 1 in. sound absorbent board, spaced on 2, 3 or 4 in. centers. The attenuation given in Table 6 depends on the spacing, depth, and the absorption of the material. At each end of the cell further attenuation results from the reflection of sound

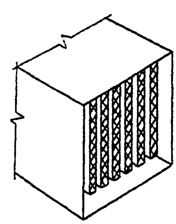

FIG. 2. PLATE CELL INSTALLED IN DUCT

from the face of the cell. An important objection to the plate cell is the increase in duct cross sectional area required. Often on the fan discharge, particularly with unitary equipment, where a number of branch ducts take off, the plate cell may be installed with little or no difficulty. The attenuation per foot of length for 1 in. board neglecting the end effect is given approximately by Equation 6.

$$R = \frac{7L\alpha^{1.4}}{S} \tag{6}$$

where

R = attenuation, decibels.
L = length of duct in linear, feet.
S = spacing between plates up to 3 in.
α = absorption coefficient of plates. For value of α see Table 7.

Duct Lining or Rectangular Cells

One series of experiments[5] made on a commonly used type of duct lining material (1 in. rock wool sheet) has shown that, subject to certain

[5]The Absorption of Noise in Ventilating Ducts. by Hale J. Sabine (*Journal Acoustical Society of America*, Vol. 12, p. 53, 1940).

TABLE 6. END REFLECTION OF PLATE ABSORBERS

PERCENTAGE FREE AREA OF ABSORBER	ATTENUATION, db
50	1
40	2
30	4
25	5
20	6

restrictions, the attenuation of single-frequency sounds may be expressed by the approximate Equation 7.

$$R = 12.6\, L \frac{P}{A}\, a^{1.4} \tag{7}$$

where

R = reduction, in decibels.
L = length of lined duct, feet.
P = perimeter of duct, inches.
A = cross sectional area of duct, square inches.
a = absorption coefficient of lining.

This formula is accurate within plus or minus 10 per cent for duct sizes ranging from 9 x 9 in. to 18 x 18 in., for cross-sectional dimension ratios of 1:1 to 2:1, for frequencies between 256 and 2048 cycles, and for absorption coefficients between 0.20 and 0.80. In Table 7, the absorption coefficients at different frequencies of a material of the above mentioned type are listed, together with the corresponding values of Equation 7.

Results of other experiments indicate, however, that Equation 7 may be in error when applied to other types of duct lining material and to duct sizes and shapes outside of the range specified. An empirically derived chart[6] representing the average experimental data on a number of different types of materials including the rock wool sheet mentioned as applicable to Equation 7 is shown in Fig. 3. Since individual materials vary somewhat, the curves in Fig. 3 are given only as representing the best available averages for duct sizes of square cross-sections from 6 x 6 in. to 48 x 48 in. As an illustration, the dotted lines in the chart show values calculated from Equation 7 which indicate that the slope for this particular material is somewhat different than from the average curves. The curves in Fig. 3, as well as Equation 7, show that the reduction in decibels is

TABLE 7. DECIBEL ATTENUATION FORMULAE FOR TYPICAL DUCT LINING MATERIAL

FREQUENCY	ABSORPTION COEFFICIENT	REDUCTION, db
256	0.37	3.0 L P/A
512	0.69	7.5 L P/A
1024	0.78	9.5 L P/A
2048	0.78	9.5 L P/A

[6]The Prediction of Noise Levels from Mechanical Equipment, by J. S. Parkinson (*Heating and Ventilating*, March, 1939, pp. 23-26).

directly proportional to the length of duct lined, and that the larger the duct the greater will be the length which must be lined in order to obtain a given noise reduction.

If, as is often the case, the length of duct from the main duct to a grille is shorter than the length of lining indicated by the calculations, this duct may be sub-divided into smaller ducts, as shown in Fig. 4. The increase in noise reduction thus obtained may be calculated from Equation 8, providing the splitters are installed parallel to the long side of the duct:

$$R_s = R_o \frac{a + bn}{a + b} \tag{8}$$

where

R_s = reduction with splitters, decibels.
R_o = reduction in same length of duct, without splitters, decibels.
a = dimension of short side of duct, inches or feet.
b = dimension of long side of duct, inches or feet.
n = number of channels formed by splitters.

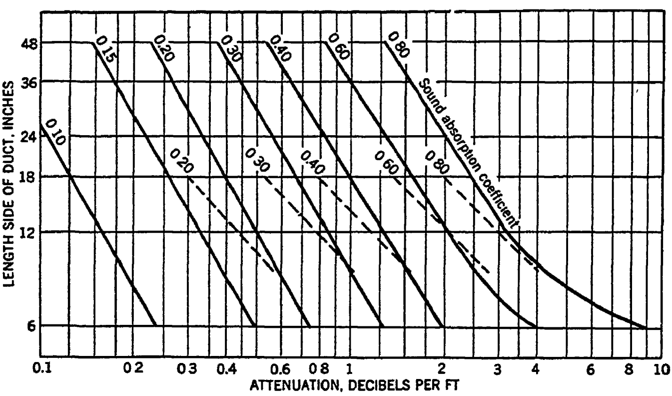

FIG. 3. SOUND ATTENUATION FOR VARIOUS ABSORBING DUCT LINERS

Example 1. An air conditioning installation is to be installed in a small theatre Determine the necessary sound treatment for the air distribution system to provide a satisfactory noise level in the theatre utilizing these conditions:

Fan tip speed 4000 fpm, total pressure 1.25 in.	77 db
Acceptable room noise level (Table 1)	40 db
Required attenuation	37 db

Solution: Natural attenuation of supply duct.

Sheet metal duct 50 ft long 48 in. x 36 in. (Table 2) 50 x 0.01	0.5 db
Elbows, two size 48 in. x 36 in. (Table 3) 2 x 1	2.0 db
Attenuation grilles to theatre air change 10 min (Table 5) outlet velocity 1000 fpm	22.0 db
Total natural attenuation	24.5 db

Difference between required and natural attenuation, 37 minus 24.5, is 12.5 db. This attenuation must be supplied by sound treatment in the duct, either in the form of duct lining or plate cells.

A similar analysis of the return duct system, shows that 15 db attenuation are to be furnished by absorptive material. An inspection of the installation shows that the lining of the plenum on the suction side of the fan would prove the most economical, where it would secure the dual function of heat insulation and sound absorption.

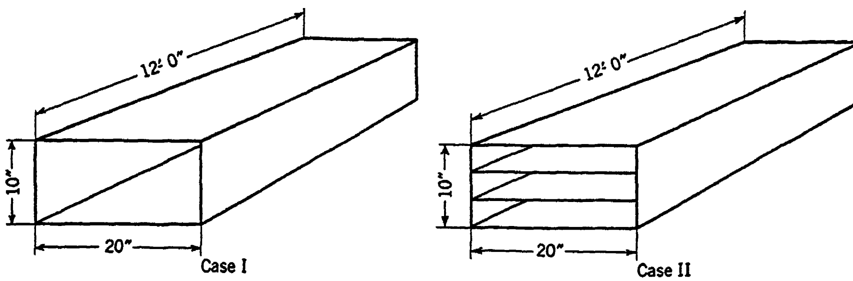

FIG. 4. DIAGRAM OF BRANCH DUCT TREATMENT WHERE LENGTH IS INSUFFICIENT FOR ADEQUATE ABSORPTION

Example 2. A 10 x 20 in. duct is connected to a private office space in a quiet location. Determine the length of lining necessary to attenuate average fan noise satisfactorily, using a lining material of a type to which Equation 7 applies, and having an absorption coefficient of 0.40 at 256 cycles. Assume that the duct is only 12 ft long, and that a 30 db reduction is required in this length.

Solution:

Case 1. (No splitters).

From Equation 7,

$$R_o = 12.6 \times 12 \times \frac{60}{200} \times 0.40^{1.4} = 12.6 \text{ db}$$

Case 2. (Two splitters, three channels).

From Equation 8,

$$R_s = 12.6 \times \frac{10 + (20 \times 3)}{10 + 20} = 29.6 \text{ db}$$

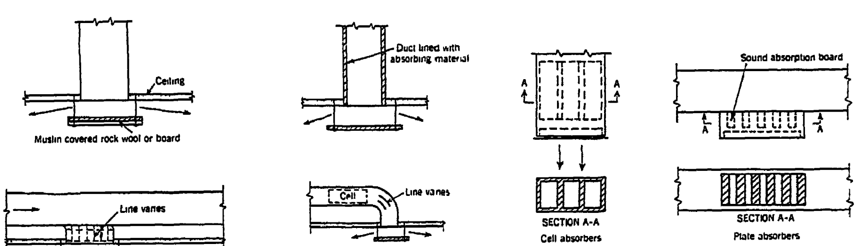

FIG. 5. OUTLET CELLS FOR PAN OUTLETS OR GRILLES

Outlet Sound Absorbers

Outlet sound absorbers are rectangular or plate cells installed directly behind an outlet or they may be the lining of a pan or plaque outlet. They are particularly effective in the elimination of high frequency whistles which are generated by air flow in the ducts. They are also

employed in large systems with long runs where only a few outlets near the fan require treatment. Frequently outlet cells are the only means of correcting existing noisy installations, as the duct sections directly behind the outlets may be the only sections acessible for treatment. (See Fig. 5.)

GRILLE NOISE

In the preceding discussion it is presupposed that the noise level generated at the face of the grille is less than the noise level of the fan minus the attenuation of the duct system up to the face of the grille. If this is not true the grille noise rather than fan noise then becomes the governing factor in room noise. Grille noise is similar in character to fan vortex noise. Knowing the noise level at the face of a grille for a given grille blade setting the noise will vary as:

$$db \text{ change} = 10 \log_{10} \left(\frac{V_{\text{new}}}{V_{\text{given}}}\right)^{5.5} \tag{9}$$

Where V is the velocity of the air through the grille. For a change in blade setting:

$$db \text{ (change)} = 10 \log_{10} \left[\frac{(\text{Total Pressure}_{\text{new}})}{(\text{Total Pressure}_{\text{given}})}\right]^{2.75} \tag{10}$$

The total pressure is measured directly behind the face of the grille. For a typical air conditioning grille the noise level at the grille face may be approximately 48 db with a total pressure behind the grille of 0.1 in. (Further discussion of grille noise is given in Chapter 31). If the grille noise at the face of the grille is more predominant than fan noise, then the resultant room noise level can be approximated by Equation 11.

$$\text{Room Level} = \left[\begin{matrix}\text{Noise Level at Face} \\ \text{of grille}\end{matrix}\right] - 10 \text{ Log}_{10} \frac{\text{Total Room Absorption in Sabines}}{\text{Total Grille Area}} \tag{11}$$

CROSS TRANSMISSION OF NOISE BETWEEN ROOMS

Ducts serving more than one room permit cross talk between the rooms and should be lined with acoustical material. Where the rooms are close together and the ducts short, the ducts should be sub-divided to provide ample acoustical treatment. Lagging material similar in character to acoustical board, when placed on the outside of ducts serves to prevent noise originating outside the ducts being carried inside the ducts and into the air stream.

A case where outside lagging is desirable occurs when ducts originate at the fan in the equipment room and pass through this room on the way to the room being conditioned or ventilated. Unless the ducts are lined some of the mechanical noise from the equipment room air may be transmitted through the wall of the duct, thus reaching the air stream and be carried into the room. In such cases, that portion of the duct which is exposed to the sounds in the equipment room should be lagged with material such as cork, pipe covering or other sound damping material to prevent the sound from entering the duct at this point. Numerical data are not available to permit a simple and practical calculating procedure to determine thickness of covering which should be used for this purpose.

Laboratory measurements have shown that the loss through a sheet of No. 22 gage metal is 24 db. When a sheet of rock wool insulation 1 in. thick and weighing 1.4 lb per square foot is added to this, the insulation value is increased to 29 db. In general, however, adding a layer of insulation or pipe covering does not materially increase the sound insulation value unless the material is dense, or unless it is surfaced with another sound impervious layer such as metal or board. Inside lining material used in the case previously mentioned would serve as an absorber of the sound transmitted through the duct walls, and thus act as a means of preventing the transfer of noise into the air stream. Inside lining may also be used in ducts to absorb noise which reaches the air stream from equipment such as fans, sprays and coils; noise due to eddying currents set up by elbows, dampers and similar obstructions; and noise transmitted from room to room where there is a common duct system.

GENERAL CONSIDERATIONS

Often in ventilating duct work the engineer feels that it will not be necessary to line ducts if the sound is traveling against the airflow. However, this is untrue since sound travels more rapidly than air in even high velocity systems, and it will travel as easily against the airflow as it does with it.

Sounds which are low in pitch are much harder to eliminate from a duct system than sound which is high in pitch, consequently equipment which produces low pitched sounds should be avoided as much as possible.

NOISE THROUGH BUILDING CONSTRUCTION

It is impossible to select equipment which will operate without producing some mechanical noise, and since the equipment must be mounted in a building, it is probable that a part of this noise will be transmitted to the building to such a degree as to make noisy conditions in the rooms which are to be air conditioned.

Controlling Vibration From Machine Mountings

Much of this noise may be transmitted by the duct if it is rigidly connected to the fan outlet. It is common practice to make the connection between the fan and the duct with a canvas sleeve which effectively restricts noise at this point. Noise may also enter the building through the mounting of the motor and the fan. Flexible mountings should be provided in all installations but these mountings must be carefully designed so that they will actually reduce the contact between the machinery and the supporting floor. If a flexible material is used, it is desirable to investigate the installation so that it is not short-circuited by through bolts which are improperly insulated and by electrical conduit which is not properly broken and is attached both to the equipment and to the building. The flexible mounting, if improperly engineered, may actually increase the contact between the equipment and the floor upon which it is supported.

In the proper isolation of vibration, which is the lower range of frequencies and does not include the *air borne* vibrations known as sound,

there is one basic law which is important in the solution of the problem. That is the law of transmissibility as governed by the equation:

$$T = \frac{1}{1 - \left(\frac{w}{w_n}\right)^2} \tag{12}$$

where

T = transmissibility of the support.
w = frequency of the vibratory force.
w_n = natural frequency of the machine unit on its support (Damping = 0).

Equation 12 shows that the transmissibility approaches unity for disturbing frequencies considerably lower than the natural frequency of the mounting. As the disturbing frequency is increased the transmissibility is also increased until at the resonant frequency, where $w = w_n$ the transmissibility becomes infinite. This is not true in practice because all materials have some internal damping effect. However, operating at or very close to the resonant frequency is always serious as forces and stresses may be multiplied 10 to 100 times. As the disturbing frequency becomes greater than the natural frequency the transmissibility becomes a smaller quantity and at the value of $w/w_n = \sqrt{2}$ it again has the value of unity. Beyond this point true isolation is first accomplished. At a ratio of 3 to 1 for w to w_n the isolation is effective enough for practical application, and experience and economical design has shown that a ratio of 5 to 1 is good. For high speeds, higher ratios for w to w_n are easily attained and give better results for effective vibration control but for the lower speeds as experienced with compressor work the higher ratios become uneconomical.

For a given installation the speed of the compressor is fixed by the specifications, therefore the value of w is fixed. That leaves only w_n to be determined and that is accomplished by the choice of mounting material and design for the support of the machine. It is well to keep in mind that when trying to isolate vibration, no attempt should be made to isolate the driving and driven piece of equipment separately. The two should be mounted on a rigid frame and then the entire assembly isolated according to the rules presented in this chapter.

The value of w_n can be controlled by the flexibility of the machine support, and when the deflection of the machine support is proportional to the load applied (such as springs or nearly so with rubber in shear) the value of w_n can be determined by Equation 13.

$$w_n = \sqrt{\frac{g}{d}} \tag{13}$$

where

g = gravitational constant.
d = static deflection of supporting material.
w = radiams per second and may be converted to frequency (f) expressed in cycles per second by Equation 14.

$$f = \frac{w}{2} = \frac{1}{2}\sqrt{\frac{g}{d}} \tag{14}$$

By the use of Equation 13 a set of curves may be plotted as shown in Fig. 6. The first line AB plotted as the critical frequencies for the various static deflections, is a curve showing the worst possible conditions or resonant conditions.

Plotting another curve CD, which is $\sqrt{2}$ times curve AB, shows the area $MCDN$ in which the resilient material or mounting does more harm than good. Plotting two more curves EF, 3 times curve AB, and GH, 5 times curve AB, shows area $EGHF$ which represents efficient and economical isolation. Area $GPOH$ is excellent isolation but for all except the highest speeds becomes rather uneconomical because of the large deflections required.

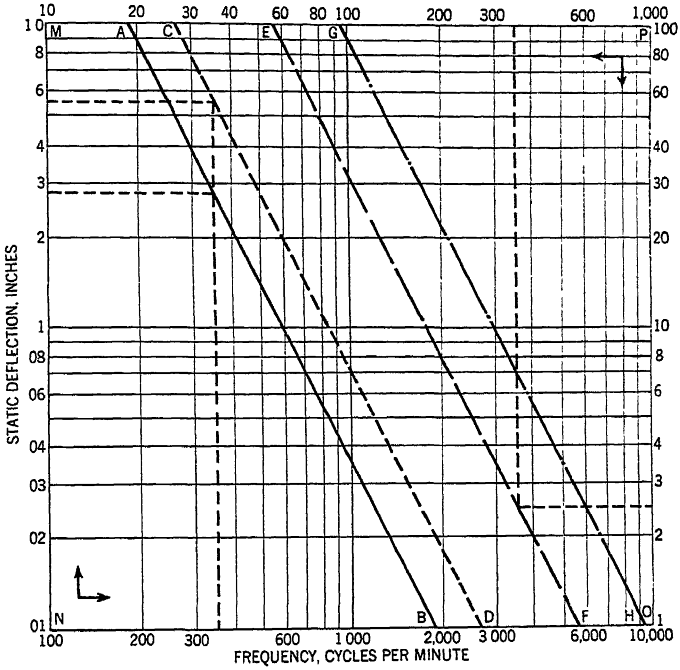

FIG. 6. STATIC DEFLECTION FOR VARIOUS FREQUENCIES

Example 3. An electric motor driven compressor unit is to be isolated. The compressor is partially balanced and operates at a speed of 360 rpm. The speed of the motor is 1160 rpm and is belt connected to the compressor. Total weight of the compressor and motor is 4500 lb.

Solution: The minimum disturbing frequency to be isolated is 360 cycles per minute. Assume that the desired ratio of forced to natural frequency is 3 as a minimum and that 5 is desired. The desired natural frequency of the mounting is 360 ÷ 5 = 72 cycles per minute.

From Fig. 6 a deflection of 7 in. is required to attain a natural frequency of 72 cycles per minute. This value may be obtained from critical curve AB for 72 cycles or from curve GH (5 times critical) for 360 cycles. For the minimum ratio of 3 the deflection would be 2.5 in.

The next step is to determine the total weight to be supported by the springs. For low speed partially balanced compressors, it has been found necessary to add a founda-

tion weighing 2 to 3 times the weight of the motor and compressor, in order to maintain the machine movement below 0.03 in.

Compressor and motor	4,500 lb.
Concrete foundation	9,000 lb.
Total	13,500 lb.

Practical application dictates the number of springs to be used, which is based on the design of the machine foundation and the supporting floor structure. However, it is desirable to design for at least 8 springs and one or two spares for cases of unknown weights. As many as 50 springs have been used on one installation. The distribution of the springs must be balanced against the masses to be supported, otherwise the foundation design and supporting structure determine the location of the springs.

The choice of the material used in the design of the resilient mounting is also important. For the slow-speed type compressor a common speed found in practice is 360 rpm. For speeds below this, isolation should not be attempted except under careful supervision. Referring to Fig. 6, it is found that for 360 rpm the static deflection required for a ratio of w/w_n of 3 to 1 (line *EF*) is 2.5 in. and for a ratio of 5 to 1 (line *GH*) it is 7 in. For these values of deflection the only choice of material is the coil spring. This is also true for speeds up to about 700 rpm. In consideration of the transverse spring constant (so as to maintain good ratios among the various degrees of freedom) experience has shown that the spring should be designed with a *working height* equal to 1.0 to 1.5 times the outside diameter. A long spring of small outside diameter has very low transverse rigidity and therefore requires some additional means of preventing side drift of the unit and on very sensitive applications this may tend to destroy the isolation efficiency. For speeds of 700 to 1200 rpm the required deflections range from 0.22 in. to 1.75 in. For these conditions rubber in shear serves as a rather satisfactory material if protected from oil. For speeds higher than 1200 rpm cork can be applied with good results. These limitations are by no means absolute because with careful and well engineered installations, especially, in consideration of all six degrees of freedom, certain liberties may be taken and still good results accomplished.

When a machine unit is properly isolated it will have a definite amount of movement which is determined by the ratio of the unbalanced forces to the total mass of the machine. If this resultant machine movement is too great for the necessary connections or the satisfaction of the customer it can be reduced only in two ways without destroying the quality of the isolation; first, adding mass or dead weight to the machine (such as concrete) common in the application of low speed, partially balanced machinery; second, accurately balancing (both statically and dynamically) all moving parts so as to eliminate the vibration at the source. This latter method is the best engineering practice and is the modern trend. However, even with well balanced machinery, installed in the vicinity of quiet offices it is usually necessary to properly isolate the equipment to prevent the transmission of vibration likely to cause complaints.

Where limitation of machine movement is desired during the starting and stopping periods, the application of friction or hydraulic damping will serve without seriously interfering with the efficiency of the isolation.

Chapter 34

AUTOMATIC CONTROL

Purpose of Automatic Control, Types of Control, Central Fan Systems, Limit Controls, Static Pressure Control, Unit Systems, Control of Automatic Fuel Appliances, Residential Control Systems, Control of Refrigeration Equipment

THIS chapter is prepared with the purpose of acquainting the engineer with the principles underlying the use of automatic control, the general types and varieties of control equipment available and their application.

Automatic control, properly applied to heating, ventilating and air conditioning systems, makes possible the maintenance of desired conditions with maximum operating economy. A properly designed and complete control system has the ability to interlock and coordinate the various functions of heating, ventilating and air conditioning in a manner impossible to accomplish with manual regulation.

Automatic control is an integral and essential part of a heating, ventilating or air conditioning installation and cannot be regarded as an accessory. In order to insure satisfactory results, the control should be designed with and incorporated in the heating, ventilating or air conditioning system. The control equipment should be given careful consideration in the planning of any installation in order that the entire system may operate together with satisfactory results.

In order that proper selection and application of controlling devices may be made it is important that a broad understanding exist as to the types of control available and their principles of operation. Improper selection and application of control equipment will result in unsatisfactory and inefficient operation. Specific control devices and systems are described in the *Catalog Data Section*.

PURPOSE OF AUTOMATIC CONTROL

Automatic control is normally applied to heating, ventilating or air conditioning systems:

1. To insure the maintenance of certain desired or required conditions of temperature, pressure, humidity, air motion or air distribution.
2. To serve a safety function, limiting pressures or temperatures within predetermined points, or preventing the operation of mechanical equipment unless it may function without hazard.
3. To produce economical results and thereby insure operation of the system at a minimum of expense.

TYPES OF AUTOMATIC CONTROL

Operating Medium or Source of Power Supply

Automatic control systems may be classified in three broad groups based upon their primary operating media or sources of power, as follows:

1. *Electric Control Systems.* In such control systems the primary medium utilized to provide for the operation is electricity, and the basic function of these controls consists of switching or otherwise adjusting electric circuits to govern electric motors, relays or solenoids. The individual units of this type of system are interconnected by line voltage or low voltage wiring, and this wiring serves to complete the circuits carrying the commands of the controllers to the controlled valves or damper motors.

2. *Pneumatic Control Systems.* In these systems the source of power for operation is compressed air, furnished by one or more centrally located compressors, and distributed in special piping to the controlling and controlled devices. The pressure is varied by the controlling instruments and this variation operates the controlled devices, which may be valves, damper motors, relays or electric switches.

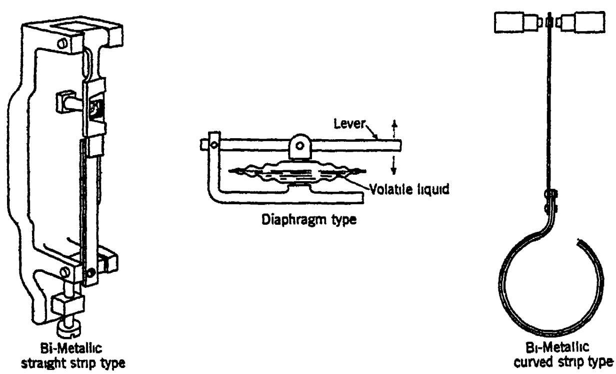

FIG. 1. TYPICAL THERMOSTATIC ELEMENTS

3. *Self-Contained Control Systems.* In self-contained control systems, the primary source of operation is the vapor pressure of a volatile liquid in the closed thermal system of the controller, which is increased or decreased in direct proportion to variation of the temperature in the controlled medium. These pressure changes are transmitted directly to the control valve or damper motor. Applications consist of valves or dampers to regulate the flow of heating or cooling media to coils, radiators, or liquid tanks, as determined by the controller element.

Typical thermostatic elements are shown in Fig. 1.

Motion of Controlled Equipment

Automatic control equipment can also be classified into two general types with respect to the characteristics of the motion imparted by the controls to the controlled equipment, such as two position or positive-acting control and modulating or graduated-action control.

In any control system it is necessary to choose the type of equipment of which the characteristics permit the type of control operation desired and in many cases both types of control are used in the same system to best meet various requirements.

1. *Two Position or Positive-acting Control.* This type of control operates positively between two positions such as *on and off* or *open and closed* with no intermediate positions or degrees of motion between the two extremes of operation. A simple thermostat which starts and stops an oil burner or a unit heater motor is an example of this type. As applied to a valve or a damper, the action of the controlling device would serve to fully open or fully close the valve or damper.

In some applications of this type of control, artificial heat is applied to the sensitive element of the room thermostat at the same time that heat is being added to the space under the control of the thermostat in order to produce more frequent operation. This usually results in more accurate control of the heat source.

2. *Modulating or Intermediate Control.* This type of control causes motion in the controlled device in proportion to motion caused in the controller by fractional degree variations in the medium to which the controller is responsive. After a fractional change has been measured at the controller and has effected a new position of the valve or damper in proportion to the amount of such change, the system stands by awaiting further change at the controller before any additional motion occurs. The extent of the motion is limited only by the limits of the controller and by the intensity of the change of conditions as measured. With this type of control, the damper or control valve may be operated in intermediate positions between its extreme limits in order to properly modulate or proportion the flow of air, steam or water, reacting with changes of conditions at the controller. Various modifications of this type of control are available, designed to meet special requirements and conditions, all based on operation of the controlled equipment in intermediate positions.

These controlling devices may be made to operate relatively faster or slower for any change in condition of the fluid being controlled. For example, a thermostat modulating a damper may move it from one extreme to the other in one degree temperature change, or many degrees change may be required to produce this same action. This characteristic is sometimes called the *sensitivity* of an instrument. The sensitivity may be fixed, or adjustable.

This type of control motion cannot be used on valves of one-pipe steam systems as the partial opening of the valves will not permit the condensate to escape against the flow of incoming steam. Where this type of control is used to control the flow of steam to a heater coil of a fan system which is in the direct path of untempered outdoor air at temperatures below freezing, care should be taken that the control point and operating characteristics of the regulator are such that the valve is open far enough, at air temperatures below freezing, to prevent the freezing of condensate in any part of the coil.

Control for Individual Rooms and Small Buildings

Control systems vary considerably with the type, size and occupancy of the building, and with the heating or cooling system, humidity supplying

equipment and ventilating means available for control. In the following paragraphs the general requirements of two types of control are discussed.

1. *Individual Room Control.* The most accurate and flexible form of control for any structure is that calling for the regulation of each individual room by control equipment reacting to conditions in that room only. Such control necessitates a thermostat in each room, located to properly measure the conditions of the room, controlling the radiator, unit heater, damper, unit air conditioner or other heating or cooling source, for that room. This arrangement permits the maintenance of any desired conditions in any room, entirely independent of any other room. In the case of large rooms, where one thermostat location will not serve to properly measure the conditions throughout the room, and where two or more sources of heating or cooling are provided in the room, additional thermostats may be used, each controlling its respective section. This form of control, due primarily to the number of control devices required over the entire building, normally is the most expensive. However, where maximum flexibility and the most accurate control are desired, individual room control should be used.

Room thermostats are available for various functions. *Dual* thermostats operate heating devices at normal temperatures during periods of normal occupancy but at lower temperatures, for economy, at other times. The change-over may be by clock or manual switches, and one or any number of thermostats may be on a single switch. *Summer-Winter* thermostats, as described for All Year Central Fan Systems, are used for reversing the operation of certain dampers or valves to make them function for both heating and cooling.

One precaution to be observed in the location of room instruments is to make sure that each is in control of all the heating and cooling devices that affect its temperature, except where two thermostats are used to operate at different temperatures.

2. *Single Thermostat Control.* A great majority of the buildings under automatic control have the comfort temperature maintained by a single thermostat operating directly on the source of heat or cooling for the entire building. In average size residences and in other small buildings, it is possible to select a thermostat location which will give entirely satisfactory results throughout the structure. This location must be one which truly represents average conditions and one which will not have unusual temperature effects. For example, a thermostat near an outside door may function improperly when the door is open. After the proper location is selected, the system is balanced to provide the proper temperature distribution.

Details of control by single thermostats will be found under the heading, Control of Automatic Fuel Appliances, in this chapter.

Zone Control

As the size of buildings increases, it becomes increasingly difficult to provide proper regulation for the entire structure from a single thermostat control. In such instances, where the advantages of individual room control are not obtainable by reason of its cost, an intermediate form of control system is available, commonly described as *zone control.* In this

form of control system a building is divided into areas or zones such that the general requirements and the general conditions through the areas are relatively constant as to exposure and occupancy, and then each zone is provided with control equipment which functions to regulate the conditions in that particular zone. As in the case of individual room control, each zone may be regulated to its own needs which may vary from the needs of other zones within the same structure.

The number of zones to be used is determined by several factors, such as:

1. Size of building.
2. Number and character of exposures.
3. Variation in occupancy or other inside conditions.
4. Cost of additional zones.

The greater the number of zones, the closer is the approach to the results and cost of individual room control. However, zone control has advantages even where individual room control is installed as it lightens the work of the room control. With room control, fewer zones are needed. In buildings of large floor area, it is usually desirable to have a separate zone for each exposure. If one wall is protected by an abutting building for half its height, two zones may be necessary. First floor conditions may vary enough from those of the rest of the building to justify a separate zone. In large buildings with several exposures toward any compass point, as occurs in wings and courts, all the northern exposures, for example, may be put on one zone control, or each north wall may have its own control. Court exposures are apt to be affected by surrounding walls and thus to require separate treatment.

In high buildings it is often important to consider zoning for stack or chimney effect in winter, caused by the difference in density between the warm air on the inside of a building and the colder air on the outside. Where the lower eight or ten stories are protected from winds by surrounding buildings, it may accentuate the need for zoning to correct the chimney effect, and on windy days there will be a marked difference in the heat requirements for the different horizontal sections at different elevations. An arrangement to provide for difference in heat requirement for exposure and chimney effect would give 12 zones; namely, north, east, south, and west lower, middle and top zones.

For steam heating systems the automatic control arrangement varies with the means of obtaining reduced temperatures. Some of the methods in common use are described in Chapter 14. From the control standpoint they are classified as follows:

1. Throttling steam to allow flow in proportion to the needs for heating.
2. Turning the steam on and off, leaving it on for longer or shorter periods as required.
3. Varying the pressure differential between supply and return lines, and varying the absolute pressures in both, so as to change the amount of steam passing through the radiators, due to the differences in pressure drop and due to the differences in volume per pound of steam.

The controlling thermostats may be inside or outside instruments or a combination of the two. Ordinary inside thermostats alone are likely to give disappointing results because an unusual condition at the thermostat upsets the whole zone, and because a slight temperature drop may allow

too much steam to pass before its heating effect reaches the thermostat. Therefore some device is needed to vary the flow in accordance with the weather. This may be a simple long range thermostat that restricts the flow as the weather moderates, or one that turns the steam off and on, on oftener and for longer periods in cold weather. One device is designed to directly control radiator temperatures at progressively lower points as the weather becomes warmer. Most outside thermostats have provision for sun and wind effect. They do not produce close control of indoor temperature, and are usually accompanied by hand switching devices for raising and lowering the control point, where individual room control is not included. They are, as stated previously, valuable adjuncts of room control.

For a hot water heating system, zone control consists of an outdoor thermostat varying the temperature of the water in accordance with the weather. This may be done by changing the amount of heat applied to the water, or by mixing hot water with recirculated water so as to produce the proper temperature. Inside zone thermostats may be used to correct improper action of weather thermostats, or, where only one outside instrument is used for a number of zones, to start and stop circulating pumps in accordance with the demand for heat in the various zones.

For both steam and hot water systems, zone control is primarily to reduce the general heating effect in moderate weather. Thus the term is used to describe a type of control system, though a building may have but a single zone.

In air conditioning systems, zone control may be applied to separate fan systems in different parts of a building or to two or more sections of the air distributing system from a single fan. The zone thermostat may be room type, or insertion type located in the return air duct from the zone. Where each zone has its own fan, the control may be the same as for an independent system. If one fan serves more than one zone, there will be heating and cooling coils for each, or a damper to mix air volumes of two temperatures to provide the proper conditions for the zone.

Zone control for an all-year air conditioning system presents problems that do not arise in either the heating or the cooling cycle alone. As a zone is normally selected for similarity of conditions, and the distribution of temperature effect to the various rooms adjusted so that one control point is sufficient, it is important that the similarity of conditions applies equally to heating and cooling. Two rooms that have like heating loads and that work well together in the heating season, may have entirely different cooling loads. This difficulty can be overcome by the use of sub-zones or individual room control where necessary.

CENTRAL FAN SYSTEMS

Central systems for air conditioning are described in Chapter 21. For explanation of the control problems for such systems, the various functions, such as heating, humidification, and cooling, are treated independently.

Ventilating Systems

A control system for a central fan ventilating system using all outdoor air and discharging air at a predetermined temperature is illustrated in

Fig. 2. Thermostat T_1 located in the outdoor air intake is set just above freezing, and controls valve V_1 on the first heating coil. This arrangement, where the valve is held completely open or closed to avoid danger of freezing, must be used where the coil is not specially designed for uniform steam distribution across its face. The by-pass damper around the heaters and the other two valves V_2 and V_3 are controlled by thermostat T_2 located in the discharge duct from the fan. When the temperature of the discharge air is too high, T_2 closes V_3 and V_2, gradually and in sequence, then if V_1 is open and supplying too much heat, T_2 opens the by-pass damper. The control of the damper and valves V_2 and V_3 must be gradual to prevent wide fluctuation in temperature.

In ventilating systems it is customary to supply air to the ventilated spaces at an inlet temperature approximately equal to the temperature maintained in the rooms. The radiators therefore are designed to take care of all the heat losses from the rooms and in order to maintain controlled room temperatures it is necessary to control the radiators independently of the ventilation control.

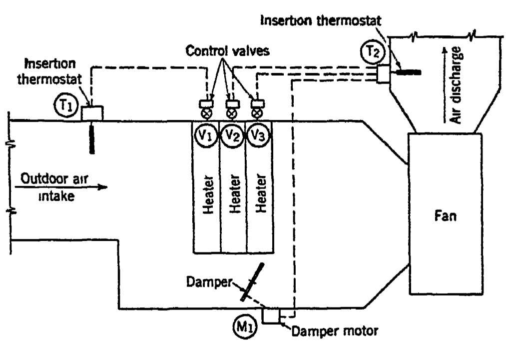

FIG. 2. CONTROL OF A CENTRAL SYSTEM FOR VENTILATION

In central fan systems, air washers are sometimes used and in such cases, due to the effect of temperatures on humidity, additional control is required. The heating coils are then divided, one or two at the inlet and usually two at the outlet, generally called preheaters and reheaters. There should be no by-pass under the former, because of the danger of a stratum of cold air freezing the water. To maintain relative humidity at a constant point a *dew-point* thermostat is inserted into the air stream between the two sets of coils, to control the preheaters. Cold air control of preheaters cannot be used because at temperatures just below freezing a standard heating coil, which will protect an air washer against freezing in zero weather, will give a too high dew-point temperature. Therefore, the one or two preheater coils must be controlled from the dew-point thermostat. This is preferably placed at the discharge side of the washer and set for about 40 or 45 F. As there is some cooling effect from the water, this provides a slightly higher temperature leaving the coils. In such cases, the throttled steam must be fairly uniformly distributed across the face of the coil, to prevent a stratum of cold air that would freeze water in the coil or in the washer.

Heating Cycle

Similar fan systems are used for heating, as well as for ventilating occupied spaces, by increasing the number of coils to four or five. Where they are all installed together the control remains the same as shown in Fig. 2, and the additional coils are controlled directly from a room thermostat which also causes T_2 to turn on full heat while the room is cool, but to function as described previously while the room is warm. This facilitates rapid heating of the room, after a vacant period.

An alternate plan is the use of a fan discharge thermostat whose control point can be automatically varied, and a room thermostat to reset it. Thus when the room is cold, air is delivered at a maximum temperature designed for rapid heating, and when the room is too warm, the air is kept as cool as can be safely introduced, or as the weather permits. The discharge temperature varies between these two extremes at the command of the room thermostat, until it finds the proper point for the existing conditions. This makes it unnecessary to vary the fan discharge thermostat manually to prevent overheating in moderate weather, or chilling in cold weather.

The heating coils are often separated into two groups, one at the suction side of the fan, controlled as shown in Fig. 2, and the other on the down stream side of T_2, controlled from the room. Control T_2 is then called the *tempered air* thermostat.

In all types of fan heating systems it is desirable to have the tempered air thermostat in the fan discharge where stratification has been broken up by the fan.

Where a fan system supplies heat to several rooms or zones that require separate treatment, the tempered air control can remain as in Fig. 2, and the variation can be supplied in any of the following ways:

1. By installing a separate duct to each zone, and using individual heating coils, each under control of its respective room thermostat.

2. By installing double chambers at the fan discharge, only one of which is supplied with additional heating coils. Individual room or zone ducts have mixing dampers which allow air to be taken from the warm air chamber, the tempered air chamber, or both, as demanded by their respective room thermostats. With this arrangement, precautions must be taken to prevent the warm air from being churned back and into the tempered air while a number of the mixing dampers are calling for the latter. If the warm air is controlled at a constant temperature under all conditions, the coils should be placed not less than 8 ft from the fan discharge and the dividing plate extended back several feet toward the fan. A good solution for the problem is to use an automatically adjusted thermostat in the warm air chamber, controlled by an outdoor thermostat so as to carry maximum warm air temperatures in the coldest weather and minimum in moderate weather.

3. By using a trunk duct, and varying the amounts of air delivered, by dampers at individual outlets or for the various zones. If a minimum amount of air is required for ventilation, the dampers must not close entirely. Thus to prevent over-heating, the trunk duct temperature must be varied according to the weather, as previously described. Also static pressure control may be needed. See a subsequent sub-head for a more detailed discussion of this subject.

Case 1 is illustrated in Fig. 3. Thermostat T in the fan discharge controls outside and return air through damper motors D_1 and D_2, face and by-pass dampers through damper motor D_3 and the steam supply to a heating coil through valve V_1. By having the face damper closed, and

the by-pass open, before steam is throttled, there can be no danger of freezing the coil. Thus the D_3 operation is completed before V_1 starts. However, the relationship between D_1 and D_2 controlling the amount of recirculation, and D_3 regulating the amount of steam heat added, depends on the design of the ventilation system. If the maximum amount of outside air is desired for ventilation, and return air is used only when insufficient steam is available, D_1 and D_2 operate to bring in all outside air before D_3 starts. On the other hand if greatest heating economy is desired and full outside air is to be used only to prevent overheating, D_3 completes its motion to close the face damper, before D_1 and D_2 start. Any relationship can be attained between these two extremes. Relay R prevents T from closing the outside damper completely, when a minimum of outdoor air is required during operation. Valves V_2 and V_3 control the steam supplied to booster coils for two zones, in accordance with the

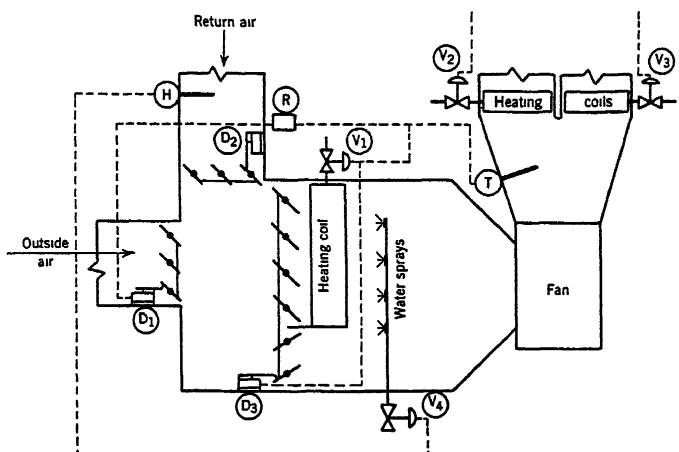

FIG. 3. CONTROL OF A CENTRAL SYSTEM FOR HEATING AND HUMIDIFICATION

requirements of room type zone thermostats, not shown in the diagram. Humidistat H, in the return air, regulates the amount of water supplied through V_4 to the spray heads.

Humidification

Humidification with air washers has been mentioned in connection with control of ventilating systems. Where ample room air change is provided, it is generally assumed that the dew-point of the conditioned space will soon equal that of the delivered air. This is due to exterior walls, especially glass, being less pervious to vapor flow than to heat transmission. With partial recirculation, dew-point control prevents over- as well as under-humidification, for ordinary installations.

Where air washers are not used, humidification may be accomplished by water sprays, preferably with heated water; by steam heated water in pans; or by steam jets, if their odors are not objectionable. Sub-atmospheric steam cannot be used, of course, for jets, and is not of much more value in coil heated pans. In all these cases the control is obtained by

humidistats, usually placed in rooms or in return air ducts. In ventilating systems the controlling instruments may be put in the fan discharge for results comparable with dew-point control. However, as hygroscopic elements are actuated by relative humidity they cannot be used with discharge temperatures that have been raised to supply heating effect.

Humidity control in cold weather is complicated by the danger of causing condensation or frost on windows and exterior walls, when otherwise desirable relative humidities are obtained. For satisfactory results in buildings that are not specially designed to prevent cold interior wall and glass surfaces, it is necessary to maintain lower humidities in very cold weather. This is done automatically either by an auxiliary humidistat mounted at a window to prevent condensation at that point, or by using a type of room or duct humidistat that is reset by an outdoor thermostat to maintain gradually drier conditions as the weather gets colder.

Cooling Cycle

Although central systems are occasionally used for cooling and dehumidifying only, the control features are essentially the same as for complete air conditioning systems. Where control of room conditions is obtained by varying the quantity of cooled air, as in a trunk duct system, individual room or zone thermostats operate volume dampers. It is customary to have these installed with a stop to prevent shutting off the air supply entirely, the reduction being from 40 to 60 per cent of the maximum delivery, depending on the design of the system. Later described control of the temperature of the air prevents over-cooling with the minimum supply, and the damper variation is normally sufficient to handle the distribution of the cooling effect throughout the area supplied by the fan or trunk duct.

In cases where systems and outlets are designed for particular velocities for proper room diffusion, volume dampers tend to produce undesirable results, by changing these velocities. Partially closed dampers reduce volumes in their ducts and increase volumes elsewhere. Trouble from too little air can be reduced by having the dampers close off, in one way or another, a part of the grille openings, thus maintaining approximately the same velocity through a smaller grille area. Trouble from increase of static pressure, due to reducing air volumes delivered, can be corrected by static pressure control, as described under a separate subheading in this chapter.

In installations where constant volumes of air are desired, and individual ducts are run to each room or zone, as shown in Figs. 3 and 5, of Chapter 21, air temperatures are varied as for the heating cycle, by room thermostats operating (1) mixing dampers which take air from either or both of two chambers, one of which has been cooled to the minimum temperature ever required; (2) booster cooling coils, one for each duct, in which the refrigerating medium can be turned on or off, or modulated; (3) individual by-pass dampers around booster cooling coils which are kept at a constant temperature; or (4) reheating coils which in times of light cooling load add heat to air that has been cooled to the minimum temperature required.

All these arrangements apply where one fan supplies more than one

room, or zone, and consequently the temperature-varying devices are downstream from the fan. The remainder of the control for the system is concerned with maintenance of conditions at the fan and is similar to what is used where a fan system is treated as a single zone.

Air washer cooling and dehumidification is commonly controlled by pumping the spray water through, or around a water cooler, with a dew-point thermostat operating a mixing valve which regulates the amount of water by-passing the cooler. An alternate scheme is to put cooling coils in the air washer spray or the pan, and to control the temperature of the coil. In both cases control of relative humidity is obtained by maintaining a constant dew-point temperature and thus a constant amount of water vapor per cubic foot of air handled. On account of this humidity factor, air leaving the washer must be reheated. As explained in Chapter 21, this is done, (1) by passing uncooled air around the washer[1] with thermostatic control of the proportion of uncooled air; (2) by adding heat by means of an automatically controlled coil; or (3) by allowing the room air to provide the heat by diffusion, in which case, still assuming a constant volume of air, the only means of dry-bulb control is the raising and lowering of the dew-point temperature, and hence the relative humidity.

Heat transfer surface coils, now more frequently used for cooling and dehumidification, are of either the direct expansion or cold water type. The former may be controlled by starting and stopping or unloading the compressor, by opening and closing a valve in the liquid line, by throttling the expansion valve, by throttling the suction line, or by raising and lowering the coil pressure, and temperature, through operation of a back pressure valve. The cold water type coils are controlled by valves to regulate the flow of water. They may throttle the flow, or they may be of the three-way type that allows a uniform flow but by-passes any necessary amount around the coil. Where well water pumps are operated only for cooling coils, control is added to stop them while cooling is not needed, but if they serve other purposes, they continue to run and the water is controlled by throttling valves.

The control with all types of coils may include a damper in an air by-pass[2] around the coils, with or without one over the face of the coils. If the installation is large enough to justify the use of two or more coils, side by side, the special air by-pass may be omitted and similar results obtained by closing the coils in sequence. The controlling instrument in all these cases is a thermostat in the room, return air, or fan discharge, whether the system serves one zone or several. In the latter case, a thermostat in the return air or in the fan discharge serves as a primary control, and the final control of room conditions is obtained with the zone thermostats.

Room or zone control in the cooling cycle is commonly provided by thermostats which operate at varying points depending on the weather. This takes care of the difference in optimum temperatures for the heating and cooling seasons and also of the objection to maintaining too high a differential between indoor and outdoor temperatures in hot weather.

[1]Patents exist covering the by-pass method.
[2]Loc. Cit. Note 1.

A thermostat sensing outdoor conditions is used to reset inside temperatures, raising them gradually to a point from 5 to 15 F below the highest outside temperature. This differential depends on the type of occupancy. Temperatures should be maintained so as to avoid too great a change for anyone entering or leaving. In large buildings, gradually lower temperatures at increasing distances from entrances and exits can be arranged. See Chapter 2 for general remarks about proper temperatures.

Except in the case of dehydrating systems, independent humidistatic control of dehumidification is seldom provided. Air washer systems as already described, are provided with dew-point thermostats. Cooling coils may be designed for proper proportion of sensible and latent heat removal so as to give satisfactory relative humidity when only the temperature is controlled. For a small installation, without by-pass or other reheat, a room thermostat and humidistat are sometimes arranged to provide cooling until both the temperature and humidity requirements

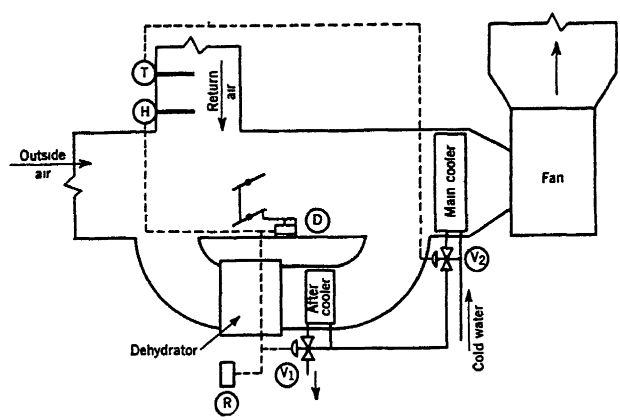

FIG. 4. CONTROL FOR A DEHYDRATING AND COOLING SYSTEM

have been satisfied, and a second thermostat is used to prevent excessive cooling by the humidistat. The cooling may be regulated by a combination of temperature and humidity that approaches *effective* temperature, by causing the relative humidity to gradually readjust the temperature control point, higher for dry air.

Control of Refrigeration. Room or duct conditions may start, stop and *unload* the refrigerant compressors directly, or may operate only at the evaporators. In either case other controlling instruments are used for the refrigeration, as described under the general heading, Control of Refrigeration Equipment.

Control of Direct Dehumidifiers—Dehydrators. Absorbent and adsorbent types of conditioning systems have the dehumidification controlled by room or return air humidistats. Since these processes are capable of producing relative humidities much below the desired point, only a portion of the air may be treated and a by-pass damper, controlled by the humidistat, used to vary this portion. The control of water cooling coils is similar to that previously described, except that the additional coil, used to extract

the sensible heat transformed from latent by the process, can use cooling water leaving another coil. That is, water leaves the main cooling coils at a low enough temperature to do the requisite cooling for the high temperature air. In order to have water available at both coils, the control valves at each are of the three-way type. As this allows free flow of water at all times, a normally closed valve can be installed in the water line and controlled by a thermostat varying the flow to maintain a suitable temperature. By connection to the fan motor circuit the valve can be kept closed while the system is not in use.

Some of these features are shown in Fig. 4. Humidistat H, on rising humidity, simultaneously starts the dehydrator and its fan through relay R, positions three-way valve V_1 to permit water to flow through the *aftercooler*, and closes damper D to increase the resistance in the main duct so as to reduce any tendency of the dehydrated air to short-circuit. Outside air and return air dampers, commonly used, are not shown. Their operation is as described for Fig. 3, except that for the summer cycle, the outside air is fully opened before V_2 turns on the main cooling

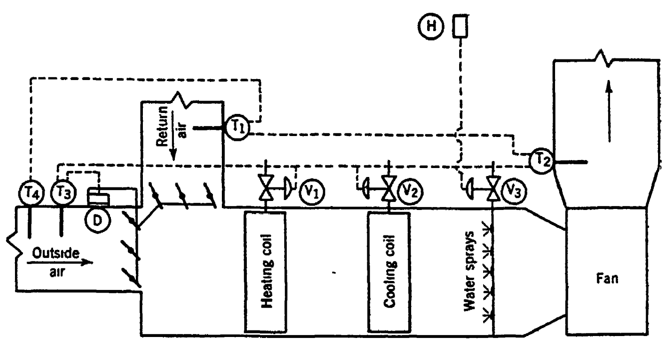

FIG. 5. CONTROL FOR AN ALL YEAR CENTRAL SYSTEM

coil, and a wet-bulb or similar thermostat in the intake cuts the outdoor air to a minimum when its wet-bulb temperature is greater than that of the return air.

All Year Systems

All year systems combine the features described for heating and cooling cycles, and have provisions for spring and fall conditions. Complete automatic control of all year systems incorporates an automatic change-over between the cooling and heating cycles. If the installation necessitates operation of a manual switch or other change-over device between the heating and cooling cycles, then the control system is semi-automatic. The full automatic change-over between cycles becomes particularly desirable in the early and late portions of the cooling and heating seasons, when heating and cooling may be required alternately.

For all year systems, a single thermostat may be used for both heating and cooling cycles, as shown in Fig. 5. In this diagram, T_2 regulates the amount of recirculation through damper motor D, the amount of steam by valve V_1 and the amount of chilled water by V_2. As the temperature rises, V_1 first operates completely to close off the steam; next, outdoor air

quantities are increased from a minimum, if the outdoor temperature as sensed by T_3 is low enough to provide cooling; and finally chilled water valve V_2 opens. Control T_2, however, operates, not at a constant temperature, but at a point varying from the minimum required in warm weather to the maximum required for heating. The variation is effected by T_1 in the return air, which raises and lowers the control point of T_2 until the proper return air temperature is obtained. During the heating and intermediate seasons, T_1 operates at a constant point, but in the cooling season it is readjusted by outdoor air thermostat T_4 to provide higher indoor temperatures. Room humidistat H opens valve V_3 on falling humidity to turn on the water sprays. As inside humidities in summer are normally higher than required in winter, the sprays are automatically kept closed.

Five diagrams of large central systems are shown in Chapter 21. The arrangement of coil and sprays diagrammed in Fig. 1 requires control as just described, except that if the cooling coil is of the direct expansion type, the refrigeration is controlled as explained in this chapter under

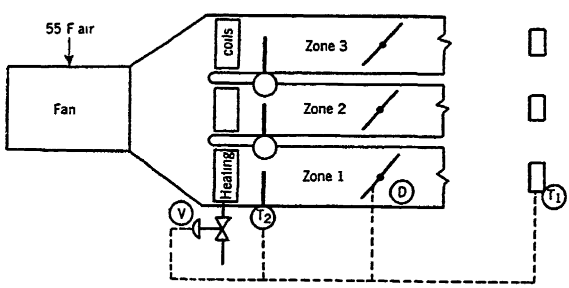

Fig. 6. All Year Zone Control with Booster Heating Coils and Volume Dampers

Cooling Cycle. In Fig. 2, a dew-point thermostat near the eliminator plates, on a rising temperature first turns off the preheater and then turns on the water cooler. A return air or fan discharge thermostat controls the reheater coil, and the return air and by-pass dampers. Assuming that the coil is not heated in summer, the by-pass damper is opened and the return air damper closed to provide reheat. In winter, provision must be made to keep the by-pass damper closed, or to reverse its operation to prevent by-passing the coil when heat is required.

The temperature at the primary fan in Fig. 4 is maintained 10 F or more below desired zone temperatures throughout the year, to allow correction of overheating in winter. In summer this setting is further reduced, to cut down the amount of outside air required for cooling and to provide sufficient dehumidification. The zone thermostats thus call for more return air for reheat.

Where the internal cooling load is great, an arrangement as shown in Fig. 6, of this chapter, has some advantages. Air entering the fan is controlled at about 55 F by operation of a steam valve and outside and return air dampers, so long as weather permits. In summer the refrigeration is turned on at a somewhat higher temperature, as required. Booster heating coils, low limit thermostats, volume dampers and room,

or return air, thermostats are installed for all zones, as shown. Volume dampers are adjusted with a minimum position that will supply sufficient air quantities for heating. While a zone is too cold, T_1 holds D in minimum position and steam valve V wide open. On rising zone temperature, the steam is first gradually turned off, and if internal heat sources cause the temperature to build up, D gradually opens to increase the amount of cool air delivered. Control T_2 is set for the minimum temperature at which air can be introduced into its zone.

If heating as well as cooling is supplied only by the fan system, and zone control is by volume dampers, special instruments known as *summer-winter* thermostats are required to open the dampers on falling temperature in winter and on rising temperature in summer. Such instruments are also used similarly to operate valves which supply hot water in winter and chilled water in summer.

Economizer Controls. Although the saving of fuel or power is one of the reasons for using any automatic control equipment, there are some applications where this is the sole reason. For example, central fan systems are usually designed to use all outdoor air, or as much as required, while it has suitable characteristics, for economical operation. Except for cases such as chemical laboratories where return air cannot be economically used, dampers are placed in both the return air and outdoor air ducts to regulate the amount of air used from each. These may or may not be mechanically interconnected but are arranged so that as one opens the other closes. Where a minimum amount of outdoor air is needed for ventilation requirements, the control of dampers may include a relay to prevent closing the outdoor damper beyond a certain adjustable point or this damper may be divided into two sections, only one of which operates with the return air damper. Another relay connected to the fan motor circuit operates the minimum outdoor opening, in either case, as the fan is started and stopped. Usually it also places the remainder of the dampers in recirculating position when the fan stops and leaves them under control of their thermostats while the fan is running.

The control of recirculation is from thermostats. Since the outdoor air, in excess of the minimum required for ventilation, is used for cooling, the dampers are commonly interconnected with other cooling devices, so as to gradually increase the amount of outdoor air, and no refrigeration is turned on until the possibilities of natural cooling are exhausted. So long as the wet-bulb temperature outside is lower than that inside the more outdoor air used during the cooling cycle, the lower the operating cost. However, as soon as the wet-bulb temperature of the outdoor air exceeds that inside, its use should be reduced to the minimum. This is done automatically by the conditions of the outdoor air as sensed by:

1. A wet-bulb thermostat.
2. A dry-bulb thermostat readjusted by a humidistat to produce operation approaching wet-bulb control.
3. A dry-bulb thermostat and a humidistat working together, either one of which may throw the dampers to recirculating position.
4. A dry-bulb thermostat, alone.

These items are listed in order of importance, from a theoretical standpoint, but practical considerations reverse the order. A wet-bulb ther-

mostat must be removed, or protected from damage, in sub-freezing weather. A dry-bulb thermostat is the most dependable under all conditions, and is generally sufficient for small installations. However, the considerably greater economy of wet-bulb or similar control justifies its use for the larger installations.

Limit Controls. There are certain limiting devices which are not concerned primarily with final room conditions but are necessary safety features. High and low limits for refrigeration pressures are described under Control of Refrigeration Equipment. Limit temperature controls are often used with heating coils exposed to sub-freezing air, to prevent freezing the condensate. Where there is danger of lack of steam pressure, a thermostat should be placed in the system to stop the fan or close the outdoor air damper when heat is not available.

The tempered air thermostat described under the Heating Cycle serves as a low limit for air introduced in winter. When cold outside air is used for cooling, this same thermostat is used to restrict the amount, while inside conditions call for full cooling. A low limit fan discharge thermostat is sometimes operated in conjunction with the refrigerating cycle, although this is usually unnecessary.

A thermostat can also be placed in the fan discharge to stop the fan in case of fire. The maximum temperature setting is often determined by local regulations, but the most protection comes from the lowest feasible setting and a point is recommended only a few degrees higher than the highest temperature of normal operation. Safety measures to prevent gravity as well as forced flow of air, in case of fire, often require various dampers throughout the fan distribution system to be closed by fusible links or by thermostats.

Static Pressure Control

As described and illustrated in Chapter 31, the discharge of air through outlets must be carefully studied for proper results. Control systems that depend upon varying the air quantities are apt to upset the design conditions. Even where the dampers are located so as to maintain proper outlet velocities, as well as possible, by closing off portions of the grilles, there is a general increase in static pressures when most of the dampers reach their minimum positions. This tends to defeat the operation of the damper and magnify the danger of noise.

Air filters, commonly used in central fan systems, vary the operating static pressure in two ways. Reduction of air quantity tends to reduce the static drop through them as through all other resistances to air flow, but accumulation of dust increases this static drop. Thus filters add to the need for static pressure control.

This control consists generally of a device operating one or more dampers. If filters are not used and the only function of the controller is to reduce high pressures caused by reduction in amounts of air delivered, the dampers may be in the side of the main duct downstream from the fan, and arranged for opening enough to relieve the excess pressure. In this case, if the controller is of the differential type, affected by ambient pressures, care must be used to prevent distortion due to slight building up of pressure in the room outside the duct.

Whether or not filters are used, dampers may be installed across the area of the duct on either side of the fan. One type is of special design for attachment to the fan intake. Closing such dampers reduces the pressure in the distribution ducts. When filters are used, the systems may be designed for operation with the dampers partially closed while the filters are clean, so the pressure controller can automatically open them to correct for the gradually increasing resistance caused by dust accumulation.

Air distribution systems designed for high velocities and consequent high pressure drops are not entirely corrected for action of volume dampers by static pressure control at the fan, because varying pressure drops through the ducts follow changes in quantities of air delivered. Therefore, where relatively constant pressures are important it may be necessary to use controllers at several carefully selected points.

Back pressure dampers, commonly used to prevent down drafts through vent flues, may be employed to relieve objectionable pressures in rooms or other spaces, under certain conditions.

UNIT SYSTEMS

A unit system provides for the same functions as a central fan system except that the actual conditioning is usually done within the space being conditioned instead of at some central location outside of the space. The automatic control problems, therefore, become exactly the same as for central fan conditioning systems except that compactness, ease of installation and control cost often assume somewhat more importance.

Because of the usual segregated location of unit equipment throughout a building and its consequent lack of competent supervision, complete automatic control is essential to its satisfactory operation.

Unit Heaters

In its simplest form, unit heater control consists of a room thermostat to start the unit heater motor when heat is required and shut it off when the demand is satisfied. With this limited control, it is possible in some instances that, with no steam available at the heater, the operation of the fan would cause objectionable drafts. To avoid this, limit controls are available which will prevent the operation of the fan at the command of the room thermostat except when steam is available, as determined by the temperature of the steam or return pipe or the pressure of the steam supply.

Where several unit heaters serve a limited area, they may be grouped for purposes of automatic control, and several heaters placed in operation at the command of one thermostat. By properly grouping the units which will operate together, the benefit of zone control can often be obtained with a minimum of control equipment. Where such group operation is utilized, the thermostat and limit control usually function through a relay, as the combined load of the several motors may exceed the current capacity of the thermostatic control device.

In some cases where cold drafts will not result, it is desirable to operate the unit heaters continuously for circulation of air. In such instances the

room thermostat regulates the supply of steam to the unit through a control valve in the steam supply line and the unit heater motor operation is manually controlled.

Unit heaters equipped with dampers arranged for by-passing air around the heating coils are controlled by room thermostats operating modulating damper motors attached to these dampers so that as the temperatures rise, a decreasing amount of air is heated. When the by-pass is wide open the heating effect is so much reduced that control of the steam supplied to the coil is not generally important. If valve control is added, the throttling of the steam may be concurrent with, or subsequent to, the opening of the by-pass.

Cooling Units

The recommended form of temperature control for a cooling unit contemplates the continuous operation of the fan, with automatic regulation of the compressor or cooling coil, or both, as determined by a thermostat in the room, or in the return air to the cooling unit. Such operation insures continuous circulation of air in the room, and in addition to providing the cooling effect of moving air, overcomes the tendency of the air to stratify. As the temperature begins to rise, the controller opens the valve to a cold water cooling coil, or for direct expansion coils, opens a valve in the refrigerant line, closes a by-pass around the coil or starts a compressor.

Cooling units may also be controlled by arranging the room thermostats to start and stop the fan motors or by a combination of motor and refrigerant control.

Unit Ventilators

There are various types of unit ventilators available but in general all types are designed to draw air from the outside or to mix outside and recirculated air, heat it and introduce it into the room under control of a thermostat.

The design of unit ventilators has to an extent been based on the requirements for automatic temperature control and the cycles of control have been developed to include other heating devices in the rooms with unit ventilators. Unit ventilators are frequently used in schools and other types of buildings where many states have laws or regulations governing the minimum amount of ventilation to be provided. The control of the amount of outdoor and recirculated air is designed to conform to the various laws. Usually the device circulates a constant amount of air and the amount automatically taken in from outdoors is controlled in one of these ways:

1. Full recirculation until the room temperature reaches a certain point, generally two degrees, below the desired room temperature; then a minimum amount of outdoor air for ventilation while the temperature is maintained by throttling steam; and if the room temperature rises with all steam shut off, the gradual increase in amount of outdoor air up to 100 per cent.

2. Full recirculation until the room reaches a set point below room temperature, after which all air is taken from outside.

3. Gravity recirculation while the fan motor is not running, with full outside air as soon as the fan starts, obtained by a relay in the motor circuit.

4. Full recirculation or all outdoor air as determined by a manual switch which can be operated at any time whether or not the fan is running. All the unit ventilators in a single building may be operated by one or many switches.

With arrangements 1 or 2, it is desirable to include a relay to prevent the intake dampers from opening while the fan is not running, regardless of room temperatures. With a dual system of control this is essential to prevent the thermostat keeping the outside damper open until the temperature falls to the reduced setting.

The intake and recirculated air quantities are determined by a single damper or by a pair of dampers working together, and operated by a damper motor. Although this affects the temperature of the air delivered, the main heat control comes from the throttling of the steam supplied to the heating coil, with or without by-pass damper control. To prevent air being delivered at too low a temperature, a low limit thermostat is commonly installed in the air stream and set at some point between 55 and 70 F. The lower settings may cause discomfort, the higher ones overheating, depending on circumstances. The air stream thermostat can be used to turn on steam, reduce the amount of outside air, or both.

Rooms with unit ventilators frequently have auxiliary heating devices, such as direct radiators, convectors or unit heaters, all under control of a single room thermostat. A common control cycle for such rooms is composed of the following functions, assuming that 72 F is the desired temperature:

1. Below 70 F the unit ventilator intake damper is in full recirculating position and all heat is turned on.
2. At 70 F the intake damper moves to a position that will admit a predetermined minimum amount of air from outdoors.
3. At 71 F the auxiliary heating devices are shut off.
4. From 71 to 72.5 F, the heating effect of the unit ventilator is throttled.
5. From 72.5 to 74 F, the intake damper is gradually moved to increase the amount of outside air from the set minimum to 100 per cent.
6. If the room thermostat calls for too much cooling, the air stream thermostat holds the delivery temperature at a proper minimum.

Other similar cycles may be used. One additional feature is the use of an air stream thermostat that has its control point reset by the room thermostat. Then as the room temperature rises, the delivery temperature is gradually reduced from a maximum to a minimum.

All Year Conditioning Units

It is desirable to provide for automatic change-over between the heating and cooling cycles in the control system for all year conditioning units because of the probable necessity of a change several times a year. In the fall season a period requiring cooling often follows one requiring heating, and the reverse is true in the spring. The automatic change-over is especially valuable where a large number of units is used.

A control system for an all year conditioning unit providing for the automatic change-over is shown in Fig. 7. Operation of the control equipment is as follows:

1. *During the Heating Cycle.* Combination controller T_1 measures the temperature in the space being conditioned and opens control valve V_2

so as to admit steam to the heating coil whenever heat is required so as to maintain a fixed temperature in the space. Combination controller T_1 also measures the relative humidity in the conditioned space and opens control valve V_1 so as to admit water to the sprays whenever moisture is required in the space.

2. *During the Cooling Cycle.* Combination controller T_2 measures the temperature and humidity in the conditioned space and opens refrigerant control valve V_3, thereby admitting refrigerant to the cooling coil whenever cooling is required to maintain the temperature or relative humidity within predetermined maximum limits.

The temperature control point of controller T_1 must be set at a lower point than that of controller T_2 in order to provide for the automatic change-over between the cooling and heating cycles. As an example, controller T_1 might be set at 72 F and 35 per cent and T_2 at 76 F and 60 per cent. As the room conditions rise above the settings of T_1, the

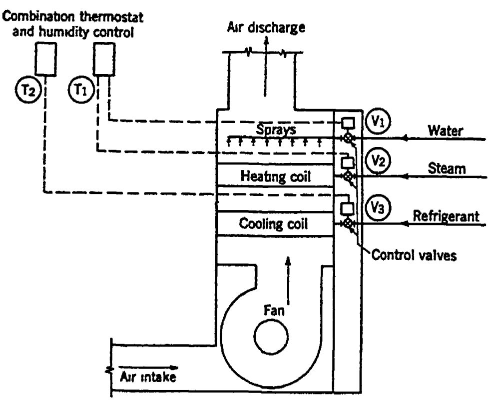

Fig. 7. All Year Air Conditioning Unit with Complete Automatic Control

heat and humidification are shut off and when they rise above the settings of T_2 the cooling and dehumidification are turned on.

CONTROL OF AUTOMATIC FUEL APPLIANCES

It is essential that automatic controls be used with oil burners, gas burners, and stokers in order to maintain even temperatures and provide safe and economical operation of the heating plant. There are many types of burners and many types of automatic control, and it is essential that the proper type of control equipment be selected to fulfill the requirements of the burner equipment and its application.

Combustion regulation equipment should be used on the larger commercial and industrial applications to control the secondary air supply and thereby provide for economical operation. This type of control will usually consist of a pressure regulator which measures and controls the pressure over the fire and which thereby indirectly regulates the carbon dioxide percentage in the flue gas.

On all automatically-fired steam boilers it is advisable to provide

control equipment which will stop the supply of fuel in case the boiler water line falls below a predetermined level of safety.

For hot water and warm air systems, control devices can be arranged to vary the water and air temperatures from outdoor thermostats. As the weather moderates, lower temperatures are maintained. Inside thermostats are usually installed to correct any improper results from the outside controls.

Thermostats used to control automatic fuel appliances may be supplied with clock mechanisms which will automatically shut off the heat or maintain lower temperatures during night hours for economy of fuel. For buildings that are not used every day of the week, clocks may be supplied to provide *night* conditions from Saturday noon or night to Monday morning.

Oil Burner Controls

In the normal oil burner installation as encountered in residential and small commercial installations, the burner operation is frequently regulated by electric controls and primarily governed by a room thermostat. It is essential that a limiting control be incorporated in the control system to prevent the temperature of the heating medium from exceeding any predetermined safe maximum. The type of limit control selected will depend on the type of the heating system. In a warm air furnace installation, a limit control would be used, reacting to the temperature of the heated air in the bonnet of the furnace; in a hot water system a control reacting to the temperature of the water in the boiler; and in a steam system a control reacting to the pressure of the steam in the boiler.

In addition to the normal control of the burner from the room thermostat and limit control, it is necessary that a combustion safety device be used to prevent operation of the burner under hazardous conditions. The oil fire is automatically ignited by means of gas, electric spark or incandescent element and the combustion safety control acting through a sequence device permits the burner operation only when the fire is properly established as the burner starts up. A further function of the combustion safety control is to react to any major disturbance in the flame during the running operation, shutting down the burner and preventing the discharge of unburned fuel if for any reason the flame is extinguished.

Gas Burner Controls

In the case of the domestic burner, full automatic operation is the normal requirement and the burner is started and stopped at the command of a room thermostat which, in turn, opens and closes a control valve in the gas supply line. Modulating controls and controls providing a high and low fire are also available for gas burners. For purposes of preventing abnormally high temperatures in the bonnet of gas-fired furnaces or in the temperature of the water in gas-fired hot water heating boilers or excessive pressures in gas-fired steam boilers, temperature and pressure limit controls are used. Ignition is normally secured through the use of a gas pilot flame and a safety device is provided, utilizing the heat of the pilot flame in such a manner that if the pilot light is extinguished for any reason, the main gas valve cannot be opened. For satisfactory

and economical operation, all automatically-fired gas burners should be equipped with pressure regulators on the gas supply line.

Stoker Controls

Domestic stokers are normally placed under command of a room thermostat for primary operation subject also to the command of a limit control to prevent their operation when conditions in the boiler or furnace exceed predetermined safe maximums. Utilizing coal as fuel, automatic ignition is not provided and the stokers, once ignited, maintain their fire, merely changing the rate of combustion by changing the draft and the rate at which the coal is fed. Thus, at the command of the room thermostat the stoker motor is started, driving a forced draft fan and fuel feeding mechanism. The rate of combustion is thus increased and this operation continues until the thermostat has been satisfied when the motor is stopped and the fuel in the combustion chamber continues to burn at a slow rate with reduced draft.

At certain seasons of the year, the operation of the stoker under the requirements of the thermostat may be so infrequent that there is a possibility of the fuel in the combustion chamber burning out or the fire going out between operations. To prevent this occurrence, automatic controls may be utilized to operate the stoker independently of thermostat requirements, sufficiently to sustain the fire either through a timing device functioning for short periods at predetermined intervals or through a temperature control device reacting to minimum stack or boiler temperatures. Control may also be utilized to prevent stoker operation and the delivery of coal into the combustion chamber in the event that the fire has gone completely out. This control is governed normally by the stack temperature and shuts down the stoker after a predetermined minimum stack temperature is reached.

RESIDENTIAL CONTROL SYSTEMS

The control installation in a residence may vary from the simple regulation of a coal-fired heating plant to the completely automatic all year air conditioning system. Residential installations with automatic fuel burning appliances, such as oil burners, gas burners or stokers, are normally equipped with single room thermostat, limit and safety controls as outlined previously under Control of Automatic Fuel Appliances.

Coal-Fired Heating Plant

Control in the normal coal-fired domestic heating plant consists of regulating the combustion rate in accordance with requirements. This function is accomplished by a spring or electric-driven damper motor which, under the command of a room thermostat and through chain linkage, operates the draft and check dampers of a boiler or warm air furnace. Such installation should be protected against excessive temperature or pressure by means of a limit control serving to check the fire when conditions at the boiler or furnace reach a predetermined maximum.

All Year Domestic Hot Water Supply

Hot water or steam heating boilers with automatic fuel burning appliances can be used for all year heating of domestic water supply. The

fuel burning appliance in this case is controlled from the temperature of water or pressure of steam in the boiler to maintain uniform boiler conditions and domestic hot water is heated by means of an indirect heater. The heating of the residence is normally governed by means of a thermostat which operates a control valve in the flow line of a gravity hot water or a steam system.

Air Conditioning Systems

Residential air conditioning systems normally include a heating source and a motor-driven fan for circulating air. In addition, such installations may involve spray-head equipment to supply humidity. Such installations distribute suitably heated and humidified air during the heating cycle, and during the summer or cooling cycle may be used effectively as conditioners if equipped with refrigeration means.

Regulation of the humidity during the heating cycle is normally accomplished by opening and closing a solenoid water valve supplying water to the spray-heads, the solenoid valve being under control of a room type humidity control. In the average installation the fan is permitted to run only during such intervals as the thermostat is calling for heat or at the command of a limit control to prevent the overheating of the bonnet of a warm air furnace. The limit control should also prevent the operation of the fan at the command of the thermostat until the circulating air temperature has increased to a predetermined point.

For the cooling equipment provided in such installations, control during the cooling cycle will be an adaptation of the control principles described for central fan systems selected for the type of cooling equipment utilized.

The selection of automatic control equipment for residential air conditioning systems is just as important as for commercial installations. Fewer controls are generally used and systems are usually less complicated except in the case of a very large residence installation when the control system may become as complete as the commercial installation.

CONTROL OF REFRIGERATION EQUIPMENT

The most common means of providing cooling for air conditioning may be divided into four general classifications as follows:

Compressor Type Refrigeration

Refrigeration compressors may furnish refrigerant to direct expansion cooling coils through which air is being passed, or to coils in cooling tanks through which water is passed which is then pumped to air washers or cooling coils through which the air is passed.

In either case the compressor motor may be started and stopped in order to meet the demand for refrigeration or a pressure controller may be used to regulate the low side or suction pressure of the compressor. When the latter method is used, the flow of refrigerant to cooling coils may be regulated by the opening and closing of a solenoid refrigerant valve at the command of a temperature controller or thermostat.

A high pressure cutout as an individual unit or in combination with

either a temperature or pressure controller provides a safety feature against excessive pressures on the high side of the compressor.

Many compressors may be *unloaded* by instruments sensing room or duct conditions, or by refrigerant pressures, thus reducing the frequency of starting and stopping. If two or more compressors are used for a single cooling system, *step controllers* are used to start them in sequence at intervals of a few seconds to avoid the large momentary electric input that simultaneous starting would demand.

When condensers are water cooled, thermostatic control to vary the quantity of water is needed for economical operation. Mechanical air condensers may be started and stopped with temperature demands.

Chilled water may be stored in tanks at temperatures slightly lower than required for air cooling coils. The control of temperature for the water distribution system is as described for Ice Cooling.

Ice Cooling

When ice is used for the cooling or dehumidification of air, it is usually placed in bunkers and water is sprayed over it. This water, after being cooled, may be used in air washers or surface cooling coils and is usually returned to the bunker for additional cooling after being used.

Control of the water temperature leaving the cold water tank may be maintained by a temperature controller, which measures the temperature of the water in the tank and modulates a control valve in a by-pass which permits a portion of the return water to return directly to the tank instead of passing through the sprays.

Vacuum Refrigeration

A vacuum refrigerating system consists of an evaporator, compressor, condenser and auxiliaries. The refrigerant used is water, and water vapor (steam) is the power medium.

Water which has been passed through an air washer or cooling coil is sprayed directly into the evaporator or water cooler where it is cooled by its own evaporation. A condenser is attached directly to the compressor discharge and its function is to recondense the water vapor drawn from the evaporator, plus the steam which supplies the energy for compression.

The temperature of the cold water leaving the flash chamber should be measured by a temperature controller which will in turn operate a two-position or positive-control valve installed in the steam line to the jet so as to permit steam to flow only when cooling is required. If city water is used in the condenser, the amount of water should be modulated according to the demand as measured at the condenser outlet by means of a temperature controller and control valve.

Refrigeration by Well Water

When well water is available in sufficient quantities at low temperatures during the cooling season, it may be pumped directly to air washers or cooling coils. Control is usually effected through control valves on the water supply to the cooling unit actuated by temperature or humidity controllers, or both, located either at the outlet of the conditioner or in the conditioned space.

Chapter 35

INSTRUMENTS AND TEST METHODS

Temperature Measurement, Pressure Measurement, Measurement of Air Movement, Air Change Measurements, Measurement of Relative Humidity, Dust Determination, Heat Transfer Through Building Materials, Measurement of Heat Exchange for Comfort Conditions, Combustion Analysis, Smoke Density Measurements, Carbon Monoxide Measurements

IN previous chapters, data from many tests and from much research on various divisions of heating, ventilating and air conditioning have been given. References have also been cited to a number of test codes adopted by the Society for the testing and rating of various types of equipment. This chapter presents a description of many test instruments, and discusses their use.

TEMPERATURE MEASUREMENT

Changes in the intensity of heat may be determined by several methods such as measuring the change in volume of a liquid, the change in internal pressure of a confined gas, the current set-up between dissimilar metals joined in a circuit, or the change in resistance of an electrical circuit.

Thermometers

The most common method used is the change in volume of a liquid such as mercury or alcohol enclosed in glass. Mercurial thermometers may be used for measuring temperatures from -40 F to approximately 1000 F. The lower limit is set by the freezing point of mercury. Since the boiling point of mercury is only about 675 F, the space above the mercury in thermometers designed for higher temperatures must be filled with an inert gas under pressure. Alcohol thermometers may be used for temperatures from -94 F to $+248$ F.

The more accurate thermometers are individually calibrated and have divisions etched on the stem. The two most common reference points are the freezing and boiling points of water. On the Fahrenheit scale, which is most commonly used in engineering work, there are 180 divisions between these points. On the Centigrade scale which is used by chemists and physicists, there are 100 divisions in this range. The temperature in degrees Fahrenheit equals $^9/_5$ of the temperature in degrees Centigrade, plus 32.

For permanent installations, glass thermometers are often protected by metal jackets and equipped with metal scales. Due to the heat

capacity and heat conductance of the jacket, it is more difficult to obtain the true temperature at a point with these than with the exposed etched stem type. The latter is usually preferred for test purposes. Where used to measure temperatures in a duct, it may be inserted through a cork or rubber plug. Care must be taken to locate the bulb at the point where the temperature is desired and in many cases several must be used to get a correct average.

Most mercury thermometers are calibrated for complete stem immersion. When incompletely immersed, a stem correction should be made for the most accurate determination. At ordinary atmospheric temperatures the correction is negligibly small, but it usually is important when measuring high temperatures such as those of steam and flue gas. The emergent stem correction may be calculated by the following equation:

$$K = 0.00009\, D\, (t_1 - t_2) \tag{1}$$

where

K = correction to be added, degrees Fahrenheit.
D = number of degrees on the thermometer scale which are not immersed.
t_1 = temperature indicated on the thermometer, degrees Fahrenheit.
t_2 = temperature of the non-immersed mercury column, degrees Fahrenheit.
0.00009 = difference in the coefficient of expansion of the mercury and glass.

In some cases, thermometers are calibrated for a certain depth of immersion indicated by an etched mark on the stem. Should such a thermometer be used for full immersion, a negative stem correction would be in order. In selecting a set of thermometers for a test, it is well to compare the group by immersion in a common bath and note variations. The more accurate ones can thus be selected for the more important positions. The interchanging of thermometers at inlet and outlet tends to cancel variations and therefore may result in greater accuracy. In extreme cases of small temperature differences involving large quantities of heat, it may be advisable to use thermometers graduated in tenths of degrees and mount magnifying glasses on them for accurate reading.

Since the bulb has considerable area, radiant energy may affect temperature readings[1]. In measuring room temperatures, care must be taken to locate thermometers away from hot surfaces such as radiators or cold surfaces such as walls or windows. Where this is impractical, shields should be used to screen the bulb from the radiant energy.

Thermocouple

When two dissimilar metals are joined at two points and a temperature difference exists between these junctions, an electromotive force will be developed. Its magnitude depends upon the metals used and the temperature difference of the two junctions. Often the cold junction is kept at 32 F by immersion in an ice bath. In other instances, a higher temperature such as that of the atmosphere is used for this junction. By proper selection of metals, any temperature up to 2900 F may be meas-

[1]Errors in the Measurement of the Temperature of Flue Gases, by P. Nicholls and W. E. Rice (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 473).

ured. Readings are obtained by means of a potentiometer or sensitive galvanometer which may be calibrated directly in degrees. A potentiometer balances the electromotive force against a known electromotive force with no current flowing, hence this method is independent of length and variations in resistance of leads. Calibration of thermocouples for high temperatures may be made against known melting points of metals. Radiation effects may be minimized by using the smallest size of wires consistent with mechanical strength. The use of small wires also makes the thermocouple sensitive to minute fluctuations in temperature.

Other advantages of thermocouples are: they are readable at remote points, they may easily and accurately be made recording, and an average temperature may be obtained readily by connecting many couples together in series.

Resistance thermometers depend for their operation upon the change of resistance of wire with change in temperature. Their use largely parallels that of thermocouples. Various metals may be used and the range is about the same as for thermocouples.

For measuring high temperatures, such as in furnaces, *pyrometers* are often used. Radiation pyrometers concentrate the radiant energy on a thermopile, and the reading is obtained on a galvanometer or potentiometer. Optical pyrometers match a narrow spectral band, usually red, emitted by the object with that from a standard electric lamp supplied with electric current.

PRESSURE MEASUREMENT

Barometer

The most accurate barometer for determining the atmospheric pressure is the mercurial type, consisting of a tube over 30 in. long closed at the top and standing in a mercury well. The barometric pressure is expressed as the height of the mercury column above the level of the mercury in the well. Such barometers are equipped with an adjustment to compensate for change in level of mercury in the well. The reading at the tube meniscus is obtained on a vernier scale. When extreme accuracy is required, as in determining the thermodynamic properties of vapors at very low absolute pressures, corrections for the variation of density of the mercury column with temperature should be made. Standard density of mercury is taken at 32 F and the conversion factor from inches of mercury to pounds per square inch is 0.491.

The following equation may be used to make corrections for temperature variations from 32 F for mercury columns:

$$h = h_1 [1 - 0.000101 (t_1 - 32)] \tag{2}$$

where

h = corrected column at 32 F, inches mercury.
h_1 = measured height of the column, inches mercury.
t_1 = observed temperature of the column, degrees Fahrenheit.

Standard atmospheric pressure at sea level is 29.921 in. mercury. Since normal atmospheric pressure decreases about 0.01 in. mercury for each 10 ft increase in elevation, it is important to make a correction if the

elevation of the barometer is not that of the test apparatus. In many cases the barometric reading may be obtained from a nearby weather bureau station. Inquiry should be made as to whether the value is as observed or corrected to sea level.

Atmospheric pressure may also be measured by an aneroid barometer which is easily portable. In this type, variations in atmospheric pressure bend the thin surface of a box or tube which contains a reduced pressure. The aneroid type is not as accurate as the mercurial and needs frequent calibration against one of the latter type.

Most of the pressure gages used in engineering work indicate the difference between the pressure being measured and the atmospheric pressure. Pressures as measured are called gage pressures. Absolute pressure may be obtained by adding barometer pressure and gage pressure algebraically.

Pressure Gages

The Bourdon type gage is a widely used device for measuring pressures. The Bourdon tube is elliptical in cross-section and circular in form, and is connected by suitable linkage to a hand which moves over a dial. An increase in pressure tends to straighten the tube and a decrease has the opposite effect. When used with high temperature steam, the tube must be protected by a water seal. When used with ammonia it must be made of steel or other material not attacked by this substance. When used for sub-atmospheric pressure, the gage is known as a vacuum gage, and is usually graduated in inches of mercury. For pressures above atmospheric, it is termed a pressure gage and is graduated in pounds per square inch. Some are made to read in both directions and are termed compound gages. Calibration is usually made with a dead weight tester, consisting of a platform and weights resting on a piston floating on oil. From the area of the piston and the total weight resting on the oil, the pressure at all points in the fluid is determined. Adjustments are provided in the gage linkage to make necessary corrections. A correction chart may also be made and used for accurate work.

For comparatively low gage pressures above and below atmospheric, the vertical *U* tube is a simple and accurate gage and is often used for test work with various fluids such as mercury, water, kerosene, or alcohol. Readings may be in inches of any of these fluids.

For measuring pressures within a few inches of water of atmospheric pressure, *U* gages are often made sloping for greater magnification of scale. In commercial gages of this type, commonly termed *draft gages*, only one tube of small bore is used and the other leg is replaced by a reservoir. Although the scale is calibrated to read in inches of water, a fluid having the density and characteristics of kerosene is often used. It is important, of course, to use a fluid having the same gravity as that for which the gage was originally calibrated, or to use a correction chart with some other fluid. Such gages may be checked one against another to detect errors in gravity of fluid. For more accurate calibration the gage may be checked against a calibrating device working on the *U* gage principle which uses hook gages and a micrometer screw. It is not considered desirable to use a slope of less than 1 to 10 in the design of these

gages. The accuracy of a draft gage is very dependent on the slope which is usually fixed by a built-in spirit level. If one side of a *U* gage is open to the atmosphere, the gage indicates pressure above or below atmospheric pressure. If both sides are connected, it indicates the difference in pressure existing between the two points of connection.

For measuring extremely low pressures accurately, very sensitive *micromanometers* of several types are available, such as the Chatelier, the Illinois or Wahlen and the Emswiler[2,3]. Calibration of these by a hook gage is impossible, and recourse must be made to fundamental calculations involving gravity of fluids and the principles involved. When proved accurate, a micromanometer is very useful for calibrating draft or slant gages.

MEASUREMENT OF AIR MOVEMENT

The problem of measuring air movement may be divided into three main parts: when confined in ducts, when circulating in free spaces, and when entering or leaving such space through openings such as grilles. Other gases might be measured by the same methods, but emphasis here will be on air measurements.

For determining the velocity, and therefore the volume of air flowing in a duct, such as in the test of a fan or a complete ventilating system, the Pitot tube as described in the A.S.H.V.E. Code[4] is probably most often used. The tube is a double tube 5/16 in. outside diameter with a rounded end up-stream. The inner tube is 1/8 in. inside diameter at the up-stream end, and the pressure in it is the sum of the velocity pressure and static pressure at its location in the duct. The outer tube, otherwise sealed, has 8 holes 0.04 in. in diameter and equally spaced around the circumference, and located eight diameters down-stream. A connection to this tube gives the static pressure. If both tubes are connected to opposite ends of a *U* gage, the gage indicates velocity pressure. At low velocities the resulting pressure head is so low that it becomes difficult to get accurate gage readings. The velocities used in many ducts are below the lower limit of determination with gages available. The relation between velocity and velocity pressure may be used to determine the range of gage required.

$$V = 1096.2\sqrt{\frac{h_V}{d}} \tag{3}$$

where

V = velocity, feet per minute.
h_V = velocity pressure, inches of water.
d = density of air, pounds per cubic foot.

Air flow in a round duct is seldom uniform. In general, the velocity is lowest near the edges, and maximum at or near the center. In order

[2]Illinois Micromanometer (*University of Illinois, Engineering Experiment Station Bulletin* No. 120, p. 91).

[3]The Weathertightness of Rolled Steel Windows, by J. E. Emswiler and W. C. Randall (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 527).

[4]Standard Test Code for Centrifugal and Axial Fans, Edition of 1938. See also Standard Code for the Testing of Centrifugal and Disc Fans (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 407; Vol. 37, 1931, p. 363).

to obtain higher velocities and more uniform flow across the measuring section, it is sometimes possible to reduce the duct to a smaller cross section at the Pitot station by use of a long transition piece. In any case, a large number of readings in two traverses should be taken, with 20 being quite desirable. These should be taken at the centers of equal areas for correct determination of volumes. For round pipe, these would be located from the center by multiplying the radius by the following factors: 0.316, 0.548, 0.707, 0.837 and 0.961. A fundamentally correct method of measurement is obtained with a Pitot tube and therefore it can be used without calibration. Pulsating or disturbed flow will give erroneous results and every effort should be made to remove disturbances in the Pitot tube section.

Many forms of Pitot tubes other than the one described have been used and calibrated[5]. A double-ended tube, one end pointing down-stream, and one up-stream, is sometimes used for low velocities, but it should be carefully calibrated for accurate results. A special form of this tube design consists of two straight ⅛ in. tubes soldered together, closed at the end, and with a 0.04 in. hole in each tube opposite the line of contact. This tube is useful in exploring velocities on exhaust inlets, such as on hoods placed around grinding wheels.

The rounded approach orifice or nozzle of the general type described in the A.S.H.V.E. Unit Heater[6] and Unit Ventilator[7] Codes is a very accurate air measuring device. When it is well made, the coefficient closely approaches unity. The velocity at the mouth is increased over that in the duct, and the resulting increased velocity pressures may be measured more accurately. The discharge from such a nozzle is very uniform and provides a good location for calibration of air velocity instruments[8].

The Venturi meter is somewhat like the nozzle except for the addition of a down-stream transition section that reduces the friction drop through the measuring apparatus. However, since a good one is expensive to make, the Venturi meter is seldom used with gases, although it is often used to measure liquids.

The thin-plate square-edged orifice has a decided advantage over the rounded approach orifice in cost. Its coefficient is approximately 0.60. The exact value depends on the location of the connections, the pressure drop, the diameter ratio of orifice to pipe, and the sharpness of the edge[9].

Another method of air measurement uses the thermal electric principle where by means of a measured amount of current, heat is put into the air stream. The temperature rise is measured, and with the specific heat of the air mixture known, the weight of air flowing may be calculated. Heat should be applied uniformly to the mass of air passing, and the small temperature difference must be determined accurately.

[5]Technical Notes No. 546, (*National Advisory Committee for Aeronautics*, November, 1935).

[6]Standard Code for Testing and Rating Steam Unit Heaters (A.S.H V.E. TRANSACTIONS, Vol. 36, 1930, p. 165).

[7]Standard Code for Testing and Rating Steam Unit Ventilators (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 25).

[8]Discharge Coefficients of Square Edged Orifices for Measuring the Flow of Air, by H. S. Bean, E. Buckingham and P. S. Murphy (*Bureau of Standards Journal Researcn*, Vol. 2, 1929, p. 561).

[9]The Flow of Air Through Circular Orifices in Thin Plates. by J. A. Polson and J. G. Lowther (*University of Illinois, Engineering Experiment Station Bulletin* No. 240).

Air Currents in Free Spaces

One of the instruments useful in determining the velocity of air currents in free spaces is the Kata-thermometer. It is essentially an alcohol thermometer with a large bulb. The stem has two marks, one corresponding to 95 F, and the other 100 F. The instrument is heated above 100 F, and the time in seconds required for it to cool from 100 to 93 when placed in the air current gives a measure of the non-directional velocity. The usual way of heating the bulb is in a water bath, and it is important to wipe the Kata-thermometer dry before taking the reading. A thermostatically controlled water bath is very convenient to use along with two instruments so one may be heating while the other is in use. For high atmospheric temperatures the high temperature Kata with a range of 130 to 135 F may be used. Usually several readings are taken in a given location and the average used. Each Kata has its own factor etched on the stem, and this factor must be used with the cooling formula or chart for obtaining the velocity. The Kata-thermometer is useful in exploring ventilated spaces to determine whether the proper air movement and distribution is being maintained. The Kata-thermometer also finds use in determining the cooling power of the atmosphere, since it loses heat by radiation and convection when dry, and by radiation, convection, and evaporation when the bulb is equipped with a wetted cloth covering[10].

Another instrument for measuring low velocity air currents is the heated thermometer anemometer[11]. This consists of an ordinary mercurial glass thermometer with a resistance winding on the bulb. Current is supplied from an external source in a measured amount. The temperature rise shown on this heated thermometer over that shown by an ordinary thermometer at the same location, and the current supplied, make it possible to calculate the non-directional velocity of the air stream. Since a smaller bulb is used than that on the Kata-thermometer, it is less affected by radiant heat sources.

Another instrument is the hot wire anemometer which is available in several patterns. In general, a measured current is supplied to raise the temperature of a fine bare wire above the temperature of the surrounding air. With the use of a very fine wire, minute fluctuations in velocity may be measured, and the area exposed to radiant exchange with heated or cooled surfaces is at a minimum. This instrument is easily made remote reading or recording. A group of them may be connected together to give the average velocity in a space, or the velocity at individual points within a test space, by suitable switching arrangements[12,13].

Deflecting Vane Anemometer

The deflecting vane anemometer consists of a pivoted vane enclosed in a case, against which air exerts a pressure as it passes through the instrument from an up-stream to a down-stream opening. The move-

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[10]Temperature, Humidity and Air Motion Effects in Ventilation, by O. W. Armspach and Margaret Ingels (A.S.H.V.E. TRANSACTIONS, Vol. 28, 1922, p. 103).

[11]The Heated Thermometer Anemometer, by C. P. Yaglou (*Journal Industrial Hygiene and Toxicology*, Vol. 20, October 1938, No. 8).

[12]Development of Testing Apparatus for Thermostats, by D. D. Wile (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 349).

[13]Linear Hot Wire Anemometer, Its Application to Technical Physics, by L. V. King (*Journal Franklin Institute*, 1916).

ment of the vane is resisted by a hair spring and a damping magnet. The instrument gives instantaneous readings of directional velocities on an indicating scale. When used in fluctuating velocities, it is necessary to average visually the swings of the needle to obtain average velocities. This instrument is very useful in locating and measuring peak velocities that may be objectionable in air conditioned spaces. Various attachments are available, such as a double tube arrangement for obtaining velocities in ducts, and a device to measure static pressures. Another attachment will be mentioned later under the measuring of velocities at inlets and outlets. Each instrument and the attachments for it must receive individual calibration.

Propeller or Revolving Vane Anemometer

The propeller or revolving vane anemometer consists of a light revolving wheel connected through a gear train to a set of recording dials that read the linear feet of air passing in a measured length of time. It is made in various sizes, 3 in., 4 in., and 6 in. being most common. Each instrument requires individual calibration. At low velocities the friction drag of the mechanism is considerable. In order to compensate for this, a gear train that overspeeds is commonly used. For this reason the correction is often additive at the lower range and subtractive at the upper range with the least correction in the middle range of velocities. Most of these are not sensitive enough for use below 200 fpm; therefore, they are not commercially available for the low velocity range met with in air conditioned spaces. Further discussion follows under air measurement at inlets and outlets.

Measurement of Velocities at Inlets and Outlets of Ducts

In the field it is often advisable to make volume measurements at the face of the supply openings. Often it is hard to get into the duct system, or it is difficult to find sections where the flow would be sufficiently uniform. The many types of approaches and grilles used make a high degree of accuracy almost impossible. For the best accuracy the instrument and its application should be checked on a similar approach and grille in the laboratory before use in the field. Where extreme accuracy is not required, such as in balancing a system, various instruments may be used at the face of the grille.

Tests have shown that the propeller type anemometer can be used successfully on the heavier type of supply grilles, such as square mesh of the cast, or pressed pattern[14]. The core area is divided into equal squares, and the anemometer is held against the face of the grille for the same length of time in each. To get the air volume in cubic feet per minute, the average corrected velocity in feet per minute thus obtained is multiplied by the average of the gross and net free area of the grille in square feet and by a correction coefficient determined as 0.97 at velocities from 150 to 600 fpm and as 1.00 at higher velocities.

On exhaust grilles, the anemometer traverse is made as described previously. The air volume may be determined by multiplying the corrected velocity in feet per minute by the gross or core area of the

[14]A.S.H.V.E. RESEARCH REPORTS Nos. 857, 911 and 966—Measurement of the Flow of Air Through Registers and Grilles, by L. E. Davies (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 201, Vol. 37, 1931, p. 619, and Vol. 39, 1933, p. 373).

grille in square feet and by a coefficient for average conditions of 0.80. This coefficient takes care of the interference of the bars of the grille and the effect on the anemometer of the air entering an exhaust grille through 180 deg[15].

When a propeller type anemometer is held in a stream of varying velocities, it tends to indicate higher than the true average, that is, the speed of the propeller is nearer to the top velocity in its area than it is to the minimum velocity. This is the main reason for the large difference in ratings of unit ventilators by the anemometer method and by air volume measurements in a duct approach to the inlet[16].

Any of the other anemometers described can be used within their range at the face of supply grilles when properly applied. In principle it is a case of finding the velocity at many points and using the average thus found with the correct discharge area at that cross-section. The deflecting vane anemometer equipped with a jet on the end of a rubber tube has been found especially convenient and accurate on supply grilles[17]. On modern air conditioning grilles the core area is used without a correction coefficient when the jet is held one inch away from the face of the grille. At this distance the constriction due to the thin bars has disappeared since the small air jets have reunited, and the air stream has not yet spread beyond the core dimensions. With deflecting grilles the exploring jet should be turned to the angle giving a maximum reading. This method of using this instrument is only applicable to supply grilles and cannot be used on exhaust grilles because of static pressure differences at the location of the jet and the instrument case.

While hardly a quantitative instrument, smoke is very useful in studying air streams and currents. The application of a more accurate instrument is often made more exact by a preliminary exploration with smoke. A mixture of potassium chlorate and powdered sugar in equal portions gives a very satisfactory non-irritating smoke. It is fired by a match, and since considerable heat is evolved, it should be placed in a pan away from inflammable objects.

AIR CHANGE MEASUREMENTS

Atmospheric air contains a certain amount of carbon dioxide. Its concentration is increased within enclosures by the carbon dioxide given off by occupants. The air changes through all means: open windows, infiltration, and mechanical ventilation, may be measured by the carbon dioxide concentration[18]. The Petterson-Palmquist apparatus has been accepted as the standard device for the determination of carbon dioxide in air. The principle used is absorption by caustic potash solution of the carbon dioxide in a known volume of air, and a remeasurement of the volume in a finely graduated capillary tube. Since the concentrations are in the order of 3 to 10 parts in 10,000, extreme care must be used to

[15]A.S.H.V.E. RESEARCH REPORT No. 1092—The Flow of Air Through Exhaust Grilles, by A. M. Greene, Jr., and M. H. Dean (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 387).

[16]A.S.H.V.E. RESEARCH REPORT No. 936—Investigation of Air Outlets in Class Room Ventilation, by G. L. Larson, D. W. Nelson and R. W. Kubasta (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 463).

[17]A.S.H.V.E. RESEARCH REPORT No. 1076—Air Distribution From Side Wall Outlets, by D. W. Nelson and D. J. Stewart (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 77).

[18]A.S.H.V.E. RESEARCH REPORT No. 959—Indices of Air Changes and Air Distribution, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 261).

obtain accurate determinations. Since occupants also give off moisture, the increase in humidity may also be used as an index of ventilation within a space. Humidity determinations are much simpler to make, but the accuracy may be affected slightly by absorption of moisture by hygroscopic materials such as fabrics and wood within the space. Measured amounts of either carbon dioxide or water vapor may be added for test purposes. However, neither method is used at the present time, and more direct methods of measuring air supply and air distribution are in favor.

MEASUREMENT OF RELATIVE HUMIDITY

Wet- and dry-bulb mercurial thermometers are usually used to determine relative humidity. The sling psychrometer is a common mounting of the thermometers to permit swinging. The wet-bulb wick and water for wetting it must be clean, and the temperature of the water should preferably be slightly above the wet-bulb temperature. An air stream velocity of 900 fpm is recommended, although velocities from 300 fpm to 1000 fpm have been found satisfactory for passage over the wet-bulb wick. The velocity may be obtained by whirling the thermometer or by aspirating air over the wet-bulb. In ducts, the air flow itself gives the proper evaporating conditions. Several observations should be made until the minimum temperature is reached. Relative humidity may be obtained from tables or psychrometric charts[19]. Although it is common practice to use the charts which are based on a barometric pressure of 29.92 in. mercury, a correction for barometric pressure is necessary for extreme accuracy. This correction is made by multiplying the relative humidity as determined from the chart by the ratio of the observed barometric pressure and the standard barometric pressure.

For temperatures below 32 F, the water on the wick is allowed to freeze, during which time the temperature will drop below the true wet-bulb. A thin film of ice is more desirable than a thick one, and it is satisfactory to remove the wick and freeze a thin film directly on the bulb. Care must be taken to read the temperatures accurately due to the slight wet-bulb depressions. Tables for ice conditions must be used[20].

The dew-point apparatus for humidity measurements consists of a polished plated container cooled by the evaporation of a volatile liquid within. The temperature at which the first slight water vapor forms is the dew-point. If the temperature is below 32 F, the deposit will appear as frost. Another method of determining humidity is by chemical means in which the water vapor is removed by a drying agent and weighed on a chemical balance. A thermal conductivity method is available for temperatures above 212 F or for extremely low humidities[21].

DUST DETERMINATION

The measurement of dust is complicated by the many kinds involved. Some of the collecting methods are impingement on viscous surfaces,

[19]Psychrometric Tables for Vapor Pressure, Relative Humidity and Temperatures of the Dew-Point, (*U. S. Department of Agriculture*, Weather Bureau, Washington, D. C.).

[20]A Review of Existing Psychrometric Data in Relation to Practical Engineering Problems, by W. H. Carrier and C. O. Mackey (*A.S.M.E. Transactions*, January, 1937, p. 33; Discussion, *A.S.M.E. Transactions*, August, 1937, p. 528).

[21]Gas Analysis by Measurement of Thermal Conductivity, by H. A. Daynes (*Cambridge Press*, 1933).

impingement at high velocity under water, collection on porous crucibles through which air passes, and electric precipitation. Determination may be by direct weighing of samples or by microscopic counting. The most commonly used methods are the modified Hill dust counter using microscopic count, the Smith-Greenburg impinger which collects samples in water and which are counted under a microscope in a Sedgwick cell[22], and the Lewis sampling tube with the analytical determination of the increase in weight of a porous crucible. All reports should state the method of sampling and counting. The A.S.H.V.E. Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work specifies the porous crucible method[23].

HEAT TRANSFER THROUGH BUILDING MATERIALS

The A.S.H.V.E. Standard Test Code for Heat Transmission Through Walls specifies the use of the guarded hot plate for tests on homogeneous materials. It further states that the mean temperature shall be at 60 F and the materials dried.

This code describes the construction and use of the guarded hot box for determining overall heat transmission coefficients of built-up sections. The standard temperature range through the test section is specified as 80 F and the mean temperature of the wall as 40 F.

The Nicholls heat meter is very useful for determining the heat flow through walls of buildings[24].

MEASUREMENT OF HEAT EXCHANGE FOR COMFORT CONDITIONS

Several instruments have been devised to measure the effect of various factors as they relate to the comfort of the body. The principle ones are the Kata-thermometer, Dufton's eupatheoscope, Vernon's globe thermometer, Winslow and Greenburg's thermo-integrator, and Yaglou's heated globe[25,26]. These instruments are attempts to stimulate and measure the heat exchanges between the human body and its environment. In order to stimulate conditions of hard physical labor, the entire surface of the device is covered with a wet cloth. At present special attention is being given the thermo-integrator as a means of measuring radiant effects of environmental conditions.

COMBUSTION ANALYSIS

The analysis of flue gases to determine completeness and efficiency of combustion is usually made chemically with the Orsat apparatus. This consists of a measuring burette, a leveling bottle, and three pipettes. Carbon dioxide is absorbed in the first pipette by potassium hydroxide,

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[22]Public Health Bulletin, No. 144, 1925, (*U. S. Public Health Service*).

[23]Testing and Rating of Air Cleaning Devices Used for General Ventilation Work, by S. R. Lewis (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 277).

[24]A.S.H.V.E. RESEARCH REPORT No. 685—Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 65).

[25]Instruments and Methods for Recording Thermal Factors Affecting Human Comfort, by C. P. Yaglou, A. P. Kratz and C.-E. A. Winslow (Year Book, *American Journal Public Health*, 36-37).

[26]The Thermo-Integrator—A New Instrument for the Observation of Thermal Interchanges, by C.-E. A. Winslow and Leonard Greenburg (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 149).

oxygen in the second by potassium pyrogallate, and carbon monoxide in the third by cuprous chloride. A known volume of gas is drawn in, and after each of the three absorptions the reduced volume is again measured in the burette. Pressure and temperature of the gas sample are kept constant while measuring. Several passes are made through each pipette which contains tubes or glass beads to increase the wetted surface. It is essential that each reaction be completed before the next reaction is started. Since the life of the reagents is limited, it is well to keep a record of the number of samples tested. Care is needed in operation to prevent the pulling of reagents out of the pipettes into the capillary tubing and burette. Many recording gas analyzers are available and are usually found in the larger plants.

SMOKE DENSITY MEASUREMENTS

A common method of determining the relative density of smoke issuing from chimneys is by visual comparison with the Ringelmann Smoke Charts. A sheet of four ruled charts with varying weights of black lines is used. The sheet is 12 by 26 in. overall on which are four charts, each consisting of 294 squares, 14 wide and 21 high. The width of line and spacings is given in Table 1.

TABLE 1. RINGELMANN SMOKE CHART SPACINGS

NUMBER OF CARD	THICKNESS OF LINES, MM	DISTANCE IN CLEAR BETWEEN LINES, MM
1	1.0	9.0
2	2.3	7.7
3	3.7	6.3
4	5.5	4.5

The charts are placed 50 ft from the observer and in line with the stack to be observed. At this distance the lines disappear and the charts appear as varying shades of gray. At times a white chart is added as No. 0 to the left of the four charts 1 to 4, and a black chart to the right as No. 5.

Apparatus using the photo-electric cell has been devised for recording smoke densities in large plants.

CARBON MONOXIDE MEASUREMENT

In garages and vehicular tunnels carbon monoxide is a constant potential danger. In small amounts it causes headaches and inefficiency, and in larger concentrations it causes collapse and death in rather short periods of exposure. A method of analyzing for carbon monoxide concentrations completes the oxidation of the carbon monoxide in a known volume of sample, in the presence of a catalyst. The heat resulting is measured by a thermocouple calibrated in parts per 10,000 of carbon monoxide[27].

[27]A Carbon Monoxide Recorder, by S. H. Katz, D. A. Reynolds, H. W. Frevert and J. J. Bloomfield (*U. S. Bureau of Mines, Technical Paper* No. 355, 1926).

Chapter 36

MOTORS AND MOTOR CONTROLS

Direct Current Motors, Alternating Current Motors for Single Phase and Polyphase, Special Applications, Classification of Motors, Manual Control, Automatic Control, Pilot Controls, Direct Current Motor Control, Squirrel Cage Motor Control, Multispeed Motor Control, Slip Ring Motor Control, Single Phase Motor Control

THE electric motor, available in many different types suitable for various services, is now the most widely used form of prime mover. The equipment for starting, controlling and protecting these motors varies with the type and with the functions it is desired to attain. Motors used for heating, ventilating and air conditioning applications may be divided into two general classifications as follows:

1. For use with direct current.
2. For use with alternating current.

DIRECT CURRENT MOTORS

There are three types of direct current motors available:

1. Shunt Wound.
2. Compound Wound.
3. Series Wound.

Shunt Wound motors, being suitable for application to fans, centrifugal pumps, or similar equipment, where the amount of starting torque required is relatively small, are used for the majority of applications in the field of heating, ventilating and air conditioning. They may be used on reciprocating pumps and compressors, if started under unloaded conditions.

Compound Wound motors are required for application to compressors, stokers, reciprocating pumps when started under loaded conditions, and also when applied to similar equipment where high starting torque is required. Whenever frequent starting makes high starting and accelerating torque desirable, or where sudden changes of load are encountered, compound wound motors are used.

Series Wound motors find only limited application in a few special cases and are available in only a limited range of sizes.

Speed Characteristics

Direct current motors are available with speed characteristics of four types:

1. Constant speed.
2. Adjustable speed.
3. Adjustable varying speed.
4. Varying speed.

Constant Speed motors may be shunt wound or compound wound. Shunt wound motors have a nearly flat speed-load characteristic, with a regulation of 15 per cent for up to ¾ hp, 12 per cent for one to 5 hp and 10 per cent for 7½ hp and larger, based on full load speed.

Compound wound motors have a speed regulation over the range from full load to no load of not more than 25 per cent, based on full load speed.

Adjustable Speed motors are usually shunt wound since it is impractical to maintain the proper relation between the shunt and series fields of compound wound motors when wide variations of the field strength are required to obtain the speed adjustment.

Adjustment of the speed of shunt wound motors is obtained by field control on motors rated at ¾ hp and larger, with the minimum or base speed at full field strength and higher speeds at reduced field strength (obtained by adding resistance in the field circuit). The speed regulation from no load to full load will not exceed 22 per cent for 2 to 5 hp; nor 15 per cent for 7½ hp and larger. Below 2 hp, the regulation may exceed 22 per cent. If closer speed regulation is required, specifically wound motors must be obtained.

Practically constant horsepower output is obtained at all speeds up to a ratio of 2 to 1. For higher speed ratios, the horsepower rating at the minimum speed is less than at the maximum speed, this difference varying with the speed ratio. High efficiency is maintained over the entire speed range. Most listed constant speed motors are suitable for operation up to a speed ratio of 2 to 1 by the use of proper control equipment.

Adjustable Varying Speed motors may be either shunt or compound wound and speed adjustment is obtained by adding resistance in series with the armature. The speed thus obtained is always below the rated full-field speed. Any standard shunt or compound wound constant speed motor may be used in conjunction with the proper armature resistor. The usual range of speed reduction is 50 per cent. The speed obtained for any setting of the resistor depends on the load of the motor and will vary with this load.

The speed regulation at high speed is comparable to a constant speed motor, but becomes poorer as the speed is decreased.

When operating at reduced speed, an increased torque requirement which the motor could easily handle at rated speed is easily sufficient to stall the motor; for example, a motor operating at two-thirds speed would be stalled by a torque about 50 per cent in excess of the normal requirement.

The efficiency of the motor is reduced as the speed is reduced, since the

loss in the resistor is greater at lower speeds. Speed reduction by armature control is usually selected where:

1. A wide speed range is not required.
2. Close speed regulation is not necessary.
3. Operating time at reduced speed is short.
4. Operating load at reduced speed is small so that the reduced efficiency can be ignored.
5. The rating is less than 1 hp.

Varying Speed motors are series wound and the speed varies with the load on the motor. They should be used where:

1. The load is practically constant or increases with speed.
2. The motor can easily be controlled by hand.

They should not be used where there is a possibility of operation without load or at a reduced load, as the speed of the motor may become dangerously high.

For shunt wound motors with full field strength, the starting torque varies almost directly with the starting current, which is dependent on the resistance in the armature circuit. With varying positions of the starting rheostat, it is possible to obtain a wide range of starting torque, within the limits of starting current permitted by the power company.

A compound wound motor requires somewhat less current for the same starting torque. The maximum torque of shunt, series, and compound wound motors is limited by commutation.

ALTERNATING CURRENT MOTORS

Alternating current motors may be divided into two main groups, namely, (1) those operating on single phase current, and (2) those operating on polyphase current.

1. Single phase motors are available in four common types:
   a. Capacitor motors.
      1. Full capacitor.
      2. Capacitor start induction run.
   b. Repulsion induction motors.
   c. Repulsion start, induction run motors.
   d. Split phase motors.

2. Polyphase (2 or 3 phase) motors are available in four common types:
   a. Squirrel cage induction motor.
   b. Automatic start induction motor.
   c. Slip ring, wound rotor induction motor.
   d. Synchronous motor.

Where the public utility supplying the current determines that a particular installation should be served with polyphase current, it is generally understood that the major portion of the motors will be for polyphase current, although it is commonly acceptable for the smaller motors to be single phase. This will limit the use of single phase current to the smaller motor ratings and the polyphase to the larger motors. Domestic and semi-commercial installations will invariably be single phase.

Single Phase Motors

Capacitor type motors are available in ratings up to 10 or 15 hp for general purposes. These motors are recommended for pumps, compressors and fan duty including housed centrifugal fans and propeller fans. The general purpose motor is commonly known as a high torque capacitor motor having approximately 300 per cent starting torque with normal current and having a different value of capacitance for starting and running which is automatically changed over by a mechanical or electrical means.

Capacitor motors for *fan duty* are usually divided into the open high torque type for belted fans and the totally inclosed non-ventilated low torque type for propeller fans mounted directly on the motor shaft. The open low torque capacitor motor may be used with small centrifugal fans mounted on the motor shaft.

Although the motors for *belted fans* are called high torque, the available starting torque is somewhat less than the torque of the general purpose motor and the slip at full load is approximately 8 per cent. With this larger amount of slip, adjustable speed down to 60 or 70 per cent of rated speed may be obtained by line voltage variation. Motors for *propeller fan* drive may be supplied with sleeve bearings to obtain greater quietness in the smaller sizes where the fan thrust does not exceed approximately 25 lb. For larger fans, thrust ball bearing motors should be used. Low torque capacitor motors have approximately 50 per cent starting torque and do not change the value of capacitance from start to run.

Capacitor motors with *high slip* may have taps brought out from the main winding which, when connected to the line, give a second speed of from 65 to 70 per cent of the normal speed. This type of motor must be specially designed for the individual fan, otherwise the correct low speed will not be obtained. Care should be exercised in applying it to centrifugal fans where restriction to the air flow through the use of adjustable dampers changes the motor load and consequently the speed. This same effect is also found in transformer speed controllers, however, a series of transformer taps allows for a selection which partially overcomes the effect of change in motor load.

Capacitor start-induction run motors are usually confined to the smaller horsepower ratings and differ from the capacitor motors by having no running capacitor. The value of starting capacitance used may vary with the different types of applications involved. These motors may be used for practically any of the applications met in air conditioning. However, consideration should be given to the fact that they are not as quiet as a capacitor motor.

Repulsion induction motors start as repulsion motors and operate under full speed as combined repulsion and induction motors through the inherent characteristics of the motor which has, in addition to the wire winding with commutator, a buried squirrel cage winding. No additional switching devices are required to change over from start to run. This and the repulsion motor described later may be used for constant speed drives where high starting torque is required and where commutator and brush noise is not a factor.

The *repulsion start-induction run* motor starts as a repulsion motor, has a switching means for transferring from start to run which short circuits the commutator and permits operation under full speed as a wound induction motor. This motor is suitable for applications similar to those for which the repulsion induction motor is used.

The *split phase* motor has a high resistance auxiliary winding in the circuit during starting which is disconnected through the action of a centrifugal switch as the motor comes up to speed. Under running conditions, it operates as a single phase induction motor with one winding in the circuit. These units are available for the lower horsepower ratings and when equipped with a high slip rotor may be used for adjustable varying speed through line voltage control.

Polyphase Motors

Squirrel cage induction motors are available in three types and a full range of sizes:

1. The normal torque, normal starting current squirrel cage motor has close speed regulation, high efficiency, high power factor, medium starting torque, high pull-out torque, and is suitable for general purpose applications. This motor has a large current inrush and a low starting current power factor. It operates with these characteristics only when started directly across the line on full voltage. When central stations require current limiting starting equipment on such motors, the starting torque is less. Current limiting hand operated starters are standard equipment.

2. The normal torque, low starting current squirrel cage motor has approximately the same torque as the normal current motor, but the starting current is about 20 per cent less than the normal torque motor on full voltage and ordinarily within the *Edison Electric Institute* locked rotor current limits on sizes up to 30 hp.

   This motor lends itself to automatic or remote control because no current limiting starting equipment is necessary up to and including 30 hp. A magnetic starter with low voltage and thermal relay overload protection gives the most satisfactory service.

3. The high torque, low current squirrel cage motor has a starting torque approximately 25 to 50 per cent greater than the normal torque motor on full voltage with starting current approximately 10 per cent less than the normal torque motor started on full voltage, but within the required limits on 30 hp sizes and smaller. These motors are also started directly across the line on full voltage through a magnetic starter or other approved starting device.

These three types of motors are also available in two, three, or four speed designs with variable torque, or constant torque characteristics. Two speed motors may be either single, or two winding; three speed motors are single, two, or three winding; and four speed motors are two, three, or four winding. When a motor is wound with a winding for each speed, better operating characteristics may be obtained because no sacrifice is made for the other speed and operating characteristics approaching single winding motors may be expected.

Frequently, multispeed motors lend flexibility to an installation that cannot be obtained in any other way.

Multispeed motors are started directly across the line through magnetic starting equipment with overload and low voltage protection and compelling relays to insure starting on low speed regardless of the ultimate running speed. Starting on low speed limits the starting current to the starting current of the low speed winding and consequently lowers the maximum demand.

TABLE 1. CLASSIFICATION OF MOTORS

Current	Type	Speed Characteristics	Full Voltage Starting Torque	Full Voltage Starting Current	Hp Range	Type of Application See Footnote*
		Constant Speed Drives				
Direct	1. Shunt	Constant	Medium	Medium	All	(*a*) Fans and (*c*) Centrifugal Pumps
Direct	2. Compound	Constant or Variable	High	Medium	All	(*b*) (*c*) (*e*) Reciprocating Pumps and frequent or hand starting
Direct	3. Series	Variable	High	Medium	Small	(*d*) Fans direct connected
Polyphase	4. Squirrel Cage General Purpose	Constant	Normal	High 6-8 Times	All	(*a*) Fans and (*c*) Centrifugal Pumps
Polyphase	5. Squirrel Cage Medium Torque	Constant	Normal	Medium 5-6 Times	Medium Small	(*a*) Fans and Centrifugal Pumps
Polyphase	6. Squirrel Cage High Torque	Constant	High	Medium 5-6 Times	Medium Small	(*b*) Reciprocating Pumps (*e*) and Compressors started loaded
Polyphase	7. Automatic Start High Torque	Constant	High	Low 3 Times	Medium	(*b*) Reciprocating Pumps (*e*) and Compressors started loaded
Polyphase	8. Slip Ring Wound Rotor	Constant	High	Low 1-3 Times with secondary control	All	(*a*) and Hoists (*b*) Reciprocating Pumps (*c*) and Frequent (*e*) or Hand Start
Polyphase	9. Synchronous High Speed	Constant	Medium	Medium 5-7 Times	Medium Large	(*a*) Fans and Centrifugal Pumps
Polyphase	10. Synchronous Low Speed	Constant	Low	Low 3-4 Times	Medium Large	(*a*) Reciprocating Compressors starting unloaded
Single phase	11. Capacitor	Constant	High	Normal	Medium Small	(*b*) Pumps and Compressors

*Applications:
a. Drives having medium or low starting torque and inertia (WR^2) such as fans and centrifugal pumps or reciprocating pumps and compressors started unloaded
b. Drives having high starting torques, such as reciprocating pumps and compressors started loaded.
c. Similar to (*a*) except where frequent or hand starting (large WR^2) requires a higher starting and accelerating torque.
d. Fans direct connected.
e. Stoker drives.

TABLE 1. CLASSIFICATION OF MOTORS—(Continued)

CURRENT	TYPE	SPEED CHARACTERISTICS	FULL VOLTAGE		HP RANGE	TYPE OF APPLICATION SEE FOOTNOTE*
			STARTING TORQUE	STARTING CURRENT		
SINGLE PHASE	12. Capacitor Fan	Constant	High	Medium	Medium Small	(a) Fans—belted
	13. Capacitor Fan	Constant	Low	Medium	Medium Small	(d) Fans—direct
	14. Capacitor Start Induction Run	Constant	Any	Medium	Medium Small	(a) Fans (b) Pumps and Compressors
	15. Repulsion Induction	Constant	High	Medium	Medium Small	(a) Fans (b) Pumps and Compressors
	16. Repulsion Start Induction Run	Constant	High	Medium	Medium Small	(a) Fans (b) Pumps and Compressors
	17. Split Phase	Constant and Adjustable	Medium	Medium	Fractional	(a) Fans (b) Pumps and Compressors
		Adjustable Speed Drives				
DIRECT	18. Shunt Field Adjustment	Constant	Medium	Medium	All	(a) Fans and (e) Centrifugal Pumps
	19. Shunt Armature Resistor	Variable	Medium	Medium	All	(a) Fans and (e) Centrifugal Pumps
POLYPHASE	20. Squirrel Cage High Slip, Tapped Winding	Variable	Medium	Medium	Medium Small	(a) Fans
	21. Squirrel Cage High Slip, Transformer Adjustment	Variable	Medium	Medium	Medium Small	(a) Fans
	22. Squirrel Cage Separate Winding or Regrouped Poles	Constant Multi-Speed	Medium or High	Low	All	(a) Fans (b) Pumps and (c) Compressors

TABLE 1. CLASSIFICATION OF MOTORS—(Concluded)

Current	Type	Speed Characteristics	Full Voltage		Hp Range	Type of Application See Footnote*
			Starting Torque	Starting Current		
Poly-phase	23. Wound Rotor, Slip, Ring, External Secondary Resistance	Variable	High	Low	All	(a) Fans and (b) Centrifugal Pumps
Single phase	24. Capacitor High Torque Tapped Winding	Variable	High	Normal	Medium Low	(a) Fans, belt
	25. Capacitor Low Torque Tapped Winding	Variable	Low	Medium	Medium Low	(d) Fans, direct
	26. Capacitor High Torque Transformer Adjustment	Variable	Low	Low	Fractional	(d) Fans
	27. Capacitor Low Torque Transformer Adjustment	Variable	Low	Low	Fractional	(d) Fans
	28. Split Phase Regrouped Poles	Constant	Normal	Normal	Fractional	(d) Fans

Often where the central station requires current limiting starting equipment for the normal torque, normal starting current motor, it is advisable to use the normal torque low starting current multispeed motor.

High slip polyphase motors may be used for adjustable varying speed drives in a manner similar to that described for capacitor motors, with either a transformer speed regulator or tapped motor windings.

It is apparent from these motor characteristics that a squirrel cage motor may be selected for operating any air conditioning and allied equipment.

Automatic start induction motors are constructed with two windings on the rotor, one of which is a high resistance, squirrel cage winding used in starting and gives a high starting torque approximately the same as the high torque, squirrel cage. A centrifugal mechanism within the motor switches to the second low resistance winding when the motor comes up to speed, thus obtaining running characteristics equal to the normal torque, normal current squirrel cage motor. The power factor of the starting current is high.

Slip ring wound rotor motors are built for two classes of service, constant speed and adjustable variable speed. The motors are identical in each case and use the same primary control, the only difference being in the secondary control.

Slip ring motors for constant speed service are used where high starting torque with low starting current is required for bringing heavy loads up to speed. The resistance is in the secondary or rotor circuit, only when starting, and is short circuited when the motor is up to speed.

For adjustable varying speed service, part or all of the secondary controller resistance is in the circuit whenever the motor is operating below full speed. The speed obtained with a given resistance in the secondary circuit is dependent on, and changes with the load on the motor. The horsepower developed by the motor is approximately proportional to the speed, whereas the power required by the motor is practically the same at reduced speed as at full speed, hence the efficiency at reduced speeds is much lower than at full speed.

Synchronous motors are ordinarily used only where there is a need for, or advantage in, obtaining power factor correction. It is necessary to consider each application as a special case which must be individually engineered, since for satisfactory operation, the combined moment of inertia of the compressor fly wheel and motor rotor must be correctly established.

The general classification of motors used for heating, ventilation and air conditioning is shown in Table 1.

SPECIAL APPLICATIONS

A few applications of motors may require special constructions such as splash proof, explosion proof, fully enclosed, and self-ventilated to meet hazardous or special duty conditions. These requirements are frequently encountered in certain industrial applications, in which cases it is necessary to select the motors from the viewpoint of service conditions, as well as the required operating characteristics to meet the demands of the machines being driven.

CONTROL EQUIPMENT FOR MOTORS

In selecting control for alternating and direct current motors it is necessary to determine whether the installation is to be operated by manual or automatic control. The available controls and the function of each group of apparatus may be outlined as follows:

1. Manual Control:
   a. To establish current.
      (1) Snap switch.
      (2) Knife switch.
      (3) Manually operated contactor.
      (4) Drum switch.

   b. Establish current and add overload protective device.
      (1) Snap switch with overload element.
      (2) Knife switch with fuse or thermal cutout.
      (3) Manual contactor with overload protective device; also reduced voltage starting compensator.
      (4) Drum switch with overload protection.

   c. Establish current and add overload and low voltage protective devices.
      (1) Not used.
      (2) Not used.

(3) Manual contactor or reduced voltage compensator with overload and low voltage release.
(4) Drum switch equipped with latch coil to give low voltage release.

2. Automatic Control:
   a. To start on full voltage.
      (1) Without overload device.
      (2) With overload device.
      (3) With combination overload device and knife switch.
   b. Reduced voltage starting.
      (1) Primary resistance type starter.
      (2) Auto compensator type.
      (3) Reactance type.

PILOT CONTROLS

In selecting pilot control devices to operate in conjunction with either manual or automatic motor control, it is necessary that they be classified as follows:

1. *Two Wire Control.* Most thermostats, float switches, and pressure regulators provide two wire control which gives low voltage release. A three position pilot switch can be used in connection with this method and thus provide manual control. With a low voltage (12 or 20 volt) control circuit it is desirable to use a low voltage thermostat. When this type of thermostat is used it will be found that a saving in the wiring cost results. When using the low voltage thermostat on a control circuit a relay and transformer panel should be used instead of the low voltage coil on the starter.

2. *Three Wire Control.* Momentary contact start and stop push button stations are usually furnished as standard accessories with automatic starters, which give low voltage protection. This control cannot be used in combination with two wire pilot devices.

In selecting manual control for an alternating or a direct current motor, the common practice is to locate the control near the motor. When the control is installed at the motor, an operator must be present to start and stop or change the speed of the motor by operating the control mechanism. Frequently manual control is employed only as a device to give overload protection and another device is employed to start and stop the motor. Manual control is used particularly on small motors which operate unit heaters, small blowers, and room coolers in an air conditioning system. In other cases manual control in the form of drums, when used with multispeed motors, is only used as a speed setting device with the starting and stopping functions operated automatically through thermostats, and pressure switches.

Because of the increasing complexity of air conditioning systems, heating, ventilating and air conditioning equipment is being operated on automatic control with less dependence on manual operation and regulation.

Automatic control of motor starters may be accomplished by the use of remote push button stations, by a thermostat, float switch, pressure regulator or other similar pilot devices. An added advantage of automatic control is that the main wiring for the starter may be installed near the motor, while the starter may be operated by a control device located elsewhere. In the majority of air conditioning installations, requiring motors 1 hp and larger, two or three phase alternating current is usually supplied.

DIRECT CURRENT MOTOR CONTROLS

Air conditioning installations using direct current power are now only used where alternating current is not available. Direct current motors are always started through starters, which are devices using a resistance to be put in series with the armature circuit during starting only, the resistance being gradually cut out as the motor comes up to speed. The starting current is held within safe limits by the use of the resistance.

The speed of a direct current motor may be regulated by the following methods:

1. Speed regulation by field control—by using a device with resistance to be put in series with the field winding. After the motor has been started to be used to increase the speed of the motor above full field speed.

2. Speed regulation by armature control—by using devices with resistance to be put in series with the armature circuit to be used to reduce the speed of the motor below full field or normal speed.

3. Combinations of field and armature control, so that the starting, field control, or armature control may be combined in a single unit.

Field control is usually preferred, depending on the size of the installation. For example, if a direct current motor were required with speed regulation between 1200 and 600 rpm, a choice of supplying a 1200 rpm motor with armature control or a 600 rpm motor with field control, both giving the same speed variation would be possible. While the 1200 rpm motor with armature control is lower in first cost than the 600 rpm motor with field control, the cost of operating the 600 rpm motor with field control is less and will save the difference in first cost over a period of time depending on the size of installation. A wide speed variation can be easily obtained in a direct current motor by using a combination of field and armature control.

SQUIRREL CAGE MOTOR CONTROL

To meet the requirements of various drives of an air conditioning system, three types of squirrel cage, two or three phase motors may be used:

1. Normal torque, normal starting current.
2. Normal torque, low starting current.
3. High torque, low starting current.

Because of the large current inrush of the normal torque, normal starting current motor, central stations usually require current limiting starting equipment on such motors above 5 hp. To meet the starting current requirements, manual or automatic current limiting starting compensators are used. These compensators are equipped with 50, 65 and 80 per cent voltage taps, the 65 per cent tap being regularly furnished when the compensator leaves the factory. Motors 5 hp and smaller have starting currents within the requirements of central stations and manual or magnetic, full voltage control may be used.

The normal torque, low starting current motor has a starting current which is approximately 20 per cent less than the normal current motor on

full voltage and well within the required current limits on 30 hp sizes and smaller. This motor, therefore, lends itself to across-the-line control because no current limiting equipment is necessary. In selecting motors for fans, pumps, or blowers, it should be noted that while the cost of the normal starting torque, low starting current motor is higher, the cost of full voltage control is lower, so that the total cost of low starting current motors with across-the-line control is lower.

A magnetic starter with low voltage and thermal overload protection gives the most satisfactory service. These switches may be controlled by remote push button stations, thermostats, or pressure switches to meet the requirements of any particular installation.

The high torque, low starting current motor has a starting current approximately 10 per cent less than the normal torque, low starting current motor when started on full voltage. These motors, most commonly used on compressor drive, can be started directly across-the-line with manual or magnetic starters.

Adjustable varying speed motor control by terminal voltage regulation requires a tap-changing switch manually or magnetically operated. Such a control switch operates to alter the voltage applied to the motor by contacting different auto-transformer voltage-ratio taps or by changing the amount of resistance inserted in the primary or line circuit.

MULTISPEED MOTOR CONTROL

To make an installation more flexible, multispeed motors are available with two, three or four speed designs, with variable torque, constant torque or constant horsepower characteristics. Multispeed may be started by means of manual or magnetic starting equipment.

When using automatic magnetic control with two, three, and four speed separate winding or consequent pole motors, control is obtained from a remote point by means of a push button master switch. The various speeds of the motor are obtained from the master switch by simply depressing the correct push button, which is known as selective speed control. It is commonly used in the smaller theater installations where the fan and motor are located backstage and the speed control is located in the lobby.

Magnetic multispeed motor controllers may also be provided with a compelling relay which makes it necessary that the operator press the first speed button before regulating the motor to the desired speed. This assures the operator that the motor is always started at low speed before the motor is adjusted to one of the higher speeds. Starting on low speed limits the starting current to the starting current of the low speed winding, and therefore, permits the use of motors in sizes larger than ordinarily permitted by central stations for full voltage starting.

Timing relays, which provide for automatic acceleration, may be used for control. With the automatic acceleration feature, it is only necessary to press the button for the desired speed. The motor will always start in low speed and automatically step up to the desired speed.

Where the change of speeds does not occur at regular intervals, and where it is only necessary to change from one speed to another to take

care of seasonal requirements, a manual drum speed selector may be used. This drum is used to select the proper motor speed while an automatic starter is used to start and stop the motor.

The smaller size speed selector drums rated 10 hp at 220 volts and smaller may also be used as a motor starter to make and break the current, as well as serving as a speed selector device. Reversible or non-reversible drums may be supplied depending on the requirements of the installation.

In the large size drums, a separate contactor must be provided to make and break the current. The contactor may be any approved starter. Overload and low voltage protection may be accomplished by using a magnetic starter. No push button station is required, the handle switch on the drum having the same characteristics as a three wire push button station.

In selecting two speed motors for fan, pump, blower, or compressor drive it will be found that the two winding motors are more expensive than the single winding. The control for two speed, two winding motors is more economical and the combined price of the motor and contactor is only slightly higher. Because of the better performance of the two speed motor and the factor of safety in having two independent motor windings, the increased cost is considered worth the difference.

SLIP RING MOTOR CONTROL

When close speed regulation and low starting current are required slip ring or wound rotor motors are used. Slip ring motors are built for two classes of service, constant speed and adjustable varying speed. The motors for the two classes of service are identical, the only difference being in the secondary control used with the motors. Control for both primary and secondary of a slip ring motor is required.

The primary control for a constant or adjustable speed is the same type as used with squirrel cage motors. Manual or magnetic starters, across-the-line type, may be used depending on the installation.

The starting current and starting torque of a slip ring motor are almost entirely dependent on the amount of resistance in the secondary control and in the manner in which the secondary control is operated. The *National Electric Manufacturers Association* has adopted service classifications which allow a selection of resistors permitting a starting current on the first contact of resistance varying from approximately 25 per cent of full load current to approximately 200 per cent of full load current or more, and permitting the resistor to remain in the secondary circuit of the motor for a period varying from not more than 15 seconds during an interval of operation from 4 minutes to continuous.

Speed regulation of a slip ring motor is obtained by inserting resistance in the secondary circuit and usually provides for a 50 per cent speed reduction when the motor takes its full rated current at normal speed. As resistors are supplied for both fan duty and constant torque duty, care should be taken in selecting the proper resistors.

Slip ring motors when used with centrifugal pumps and fans should have fan duty resistors. Because of the low current inrush of the fan and pump

load a starting resistor *NEMA* classification No. 15 may be used. For speed regulation resistor, classification No. 93 should be selected. On a compressor drive using an unloader, a constant torque resistor classification No. 15 should be used. If the compressor is started under load, *NEMA* classification No. 56 or 76 is used. For constant torque speed regulation, resistor No. 95 is used.

SINGLE PHASE MOTOR CONTROL

Where three phase current is not available or where single phase operation is preferred, then single phase repulsion induction, capacitor type or multispeed single phase motors may be used. Since the starting currents of all single phase motors are required to be within the starting-current limits established by the local power supply company, a suitable type of starter may be chosen from the following selection:

1. Enclosed two pole manually operated motor starters with thermal overload protection.
2. Enclosed two pole automatic motor starter operated by a push button, thermostat or similar device, with thermal overload relay and low voltage protection.
3. A manual or magnetic resistance type starter with low voltage protection.
4. A manual or magnetic control for pole changing motors and for adjustable varying speed motors using an auto-transformer or resistance in the primary circuit to obtain line (or terminal) voltage drop.

In selecting across-the-line control for single phase capacitor type motors it is usually very desirable to use three pole across-the-line starters. Control for multispeed, single phase capacitor motors may be selected from tables on three phase rating when consideration is given to the increased current and the necessary switching of connections.

Chapter 37

AIR CONDITIONING IN THE TREATMENT OF DISEASE

Operating Rooms, Reducing Explosion Hazards, Nurseries for Premature Infants, Fever Therapy, Cold Therapy, High Temperature Hazards, Control of Allergic Disorders, Oxygen Therapy, General Hospital Air Conditioning

IN the past few years air conditioning has made considerable progress as an adjunct in the treatment of various diseases. Among the important applications are those in operating rooms, nurseries for premature infants, maternity and delivery rooms, children's wards, clinics for arthritic patients, heat therapy, cold therapy, oxygen therapy, X-ray rooms, the control of allergic disorders, and for the physiological effects in industry.

OPERATING ROOMS

The widest application of air conditioning in hospitals is in operating rooms. Complete air conditioning of operating wards is important because winter humidification helps reduce the danger of anesthetic gases; summer cooling with some dehumidification is needed to eliminate excessive fatigue and to protect the patient and operating personnel; and finally, filtering for the removal of allergens from the operating room air.

Reducing Explosion Hazard

Explosion hazards in operating rooms began with the introduction of modern anesthetic gases and apparatus. Ether administered by the old drop method is still regarded as comparatively safe; but when mixed with pure oxygen or with nitrous oxide in certain concentrations the explosion hazard may be as great as with ethylene-oxygen, or cyclopropane-oxygen mixtures[1]. (See Table 1.)

During the course of ethylene anesthesia, the mixture, usually 80 per cent ethylene and 20 per cent oxygen, is so rich that the danger of explosion is slight in the immediate vicinity of the face mask, but leakage of ethylene into the air may accumulate to any lower concentration, and thus introduce a serious hazard. The most dangerous period is at the end of the operation when the patient's lungs and the anesthesia apparatus are customarily *washed out* with oxygen with or without the addition of

[1]Safeguarding the Operating Room Against Explosions, by Victor B. Phillips (*Modern Hospital*, 46, April and May, 1936).

carbon dioxide. Even when this procedure is omitted, it is difficult in practice to avoid dilution of the anesthetic gas with air during the normal course of breathing following the administration. In either case the mixture would pass through the explosion range and extraordinary precaution is necessary for the safety of the patient and operating personnel.

Copious ventilation from 6 to 12 air changes per hour reduces to some extent the danger from the open drop method but is of little value in the closed system type of anesthetic machine now in common use. However, this abundant circulation reduces the concentration of anesthetic gases to below the physiologic threshold so that the surgeon and his personnel will not be affected.

The most important cause of accidents is probably static sparks which may result from accumulation of frictional charges on the rubber surfaces of the anesthesia apparatus, on woolen blankets, and on the bodies of the operators as they walk on insulated floors, when the humidity is low. Grounding the various parts of the anesthesia apparatus is not entirely

TABLE 1. EXPLOSIVE PROPERTIES OF ANESTHETICS[a]

Anesthetic	Formula	Density Air = 1	Limits of Inflammability			
			In Air		In Oxygen	
			Lower	Upper	Lower	Upper
Ethylene	C_2H_4	0.97	2.75	28.6	2.90	79.9
Propylene	C_3H_6	1.45	2.00	11.1	2.10	52.8
Cyclopropane	C_3H_6	1.45	2.40	10.3	2.45	63.1
Nitrous Oxide	N_2O	1.52	-----	-----	Not Inflammable	
Ethyl Chloride	C_2H_5Cl	2.23	4.00	14.8	-----	-----
Ether-divinyl	$(C_2H_3)_2O$	2.42	1.70	27.0	1.85	85.5
Ether-diethyl	$(C_2H_5)_2O$	2.56	1.85	36.5	2.10	82.0
Chloroform	$CHCl_3$	4.12	-----	-----	Not Inflammable	

[a]Explosion and Fire Hazards of Combustible Anesthetics (*U. S. Bureau of Mines*, Report of Investigations No. 3443, April, 1939).

effective, so long as rubber remains in use in the conventional equipment. Some form of protective grounding within the apparatus may be a partial solution.

A comprehensive study of the explosion problem and of the general causes and prevention of operating room hazards is being conducted by the *University of Pittsburgh*, the A.S.H.V.E. Research Laboratory, and the *U. S. Bureau of Mines*. The first result of this investigation has been a fruitful attempt to eliminate the explosive range of cyclopropane, one of the best but most difficult gases to handle. The use of helium as a diluent in the total gaseous mixture controls the oxygen concentration by replacement and since its flame quenching qualities are known it is the ideal gas for this purpose. In addition, a gaseous mixture containing helium is more difficult to ignite by electric discharges and this quality also increases the safety factor of anesthetic administration. A more general idea of the mixtures containing cyclopropane, oxygen and helium necessary to produce satisfactory anesthesia is given in Table 2. Clinically and with slight variation, the non-inflammable mixtures of Table 2

have produced satisfactory results and samples of gas taken during operation show no tendencies to explosion.

In the absence of more understanding, no single safeguard can be given, but desirable precautions may be classed as follows: (1) to limit the region of the explosive gas mixtures; (2) to make all electric contacts explosion-proof; (3) to avoid building up static charges; (4) to ground those surfaces where charges may be built up; and (5) to discourage accumulation of static electrical charges by humidity control.

Operating Room Conditions

Little is known about optimum air conditions for maintaining normal body temperatures during anesthesia and the immediate post-operative period. An anesthetized patient displays dilatation of blood vessels in the skin resulting in profuse sweating and (it has been believed) inability to regulate body temperature. From this it was concluded that all anesthetized patients suffered considerable heat loss. In spite of this a recent paper[2] reports little more than 0.8 F variation in the rectal temperature during the course of the operation. The severe physiological effects, such as excessive sweating and rapid pulse, of high operating room temperatures on attendants and patients during the hot months signify the need for proper cooling. A comparison of surgeons' statements who operate in both air conditioned and non air conditioned rooms strongly indicates lesser fatigue; and the greater recuperative power of the patient is confirmed by the previously referred to study[3].

TABLE 2. NON-INFLAMMABLE MIXTURES FOR ANESTHETIC USE[a]

Mixture No.	Composition, Per Cent by Volume		
	Cyclopropane	Oxygen	Helium
1	15	20	65
2	20	20	60
3	25	25	50
4	30	30	40

[a]Explosive Properties of Cyclopropane: Prevention of Explosions by Dilution with Inert Gases (*U. S. Bureau of Mines*, Report of Investigations No. 3511, May, 1940).

Although the comfortable air conditions for the operatives are not identical with those for the patient a compromise is as a rule not difficult; with a relative humidity of 55 to 60 per cent, temperatures from 72 to 80 F are used. The work just cited, reported that 68 to 70 deg effective temperature not only furnished comfort for the operating room workers but apparently prevented exhaustion of the patient as evidenced by rapid convalescence in the recovery ward. Additional heat may be furnished to the patient locally or by suitable covering according to body temperature in individual cases.

[2]A.S.H.V.E. RESEARCH REPORT No. 1111—Air Conditioning Requirements of an Operating Room and Recovery Ward, by F. C. Houghten and W. Leigh Cook, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 161).

[3]Loc. Cit. Note 2.

In an investigation recently conducted at the University of Pittsburgh, in a cooperative research program with the Society, comparative studies were made on the bacterial content of conditioned and non-conditioned operating rooms. From these studies[4] it was concluded that the bacterial content of conditioned operating rooms was considerably less than that of non-conditioned rooms. Although this difference may not be great it is sufficient to demonstrate that properly conditioned spaces with adequate filtration can definitely reduce the bacterial and other foreign substance content in an enclosure.

The increasing incidence of allergies or of their recognition is becoming a factor in the operating room. Operations may be postponed on allergic patients during asthmatic manifestations through fear of complications. The removal of the allergens, therefore, is in some cases an important function of the air conditioning system.

Central system air conditioning plants and unit air conditioners prove satisfactory in operating rooms when producing between 8 and 15 air changes per hour of filtered and properly conditioned air without recirculation during the course of anesthesia. A separate exhaust fan system is as a rule necessary to confine and remove the gases and odors. Double windows are desirable and often necessary to prevent condensation and frosting on the glass in cold weather and to minimize drafts. The high air flow of 8 to 15 air changes in operating rooms is desirable for three reasons: (1) to reduce the concentration of the anesthetic to well below the physiologic threshold in the vicinity of the operating personnel, (2) to remove the great amounts of heat and sometimes moisture, from sterilizing equipment if inside the operating room, from the powerful surgical lights, from solar heat, and from the bodies of the operatives, and (3) to provide extra capacity for quickly preparing the room for emergency operations. Much can be gained by careful insulation of sterilizing equipment and by thorough exhaust ventilation of sterilizing rooms adjoining the operating rooms.

A very common complication presumably traceable to operations is pneumonia. The difference in conditions between the operating room and the final hospital destination of the patient, including corridors and elevators, is conducive to post-operative pneumonia. A suggested remedy is a recovery ward where conditions closely approximate those of the operating room and in which the patients remain from one to four days. Satisfactory conditions in the recovery ward not only hasten convalescence, but dispel the fear frequently found in patients who must undergo operations during the hot seasons[5].

Sterilization of Air in Operating Room

Of considerable significance to operating rooms and contagious wards is the use of ultra-violet radiation for sterilizing the air[6]. Results reported[7]

[4]Report on Air Conditioning in Surgery, by W. Leigh Cook, Jr. (Department of Industrial Hygiene, School of Medicine, University of Pittsburgh, 1940).

[5]Report of the Committee on Air Conditioning (*The American Hospital Association*, 1937, p. 2).

[6]Air-Borne Infection and Sanitary Air Control, by W. F. Wells (*Journal Industrial Hygiene*, 17:253, 1925).

[7]Sterilization of the Air in the Operating Room by Special Bactericidal Radiant Energy, by Deryl Hart (*Journal Thoracic Surgery*, 6:45, 1936).

would indicate that the post-operative temperature rise of patients during the first few days is in most instances caused more by bacterial contamination of the operative wound than by the absorption of blood and traumatized tissues. Operating room infections, which were quite frequent before the installation of special ultra-violet lamps, are apparently being reduced.

Direct ultra-violet radiation is distinctly advantageous in sterilizing not only the site of operation but also wounds to prevent the spread of infection. In infants' wards, contagious disease wards, and even in school rooms, the sterilizing effects are definitely known. Whether an air conditioning system with ultra-violet installations in the ducts is a feasible procedure is controversial; but it would appear that this indirect method is not as satisfactory as the direct in the light of present reported knowledge.

TABLE 3. NET MORTALITY OF PREMATURE INFANTS ACCORDING TO HUMIDITY[a]
Infants Hospital, Boston, Mass.

Cause of Death	Unconditioned Nurseries (1923-1925)	Conditioned Nurseries (1926-1929)	
	Natural Humidity	Relative Humidity	
		25-49 Per Cent	50-75 Per Cent
	Per Cent Mortality	Per Cent Mortality	Per Cent Mortality
Acute and chronic infections	26.5	9.7	0.0
Congenital deformities	1.2	0.0	0.7
Unclassified	1.2	4.8	0.0
All causes	28.9	14.5	0.7

[a]Excluding cases with multiple congenital anomalies incompatible with life, and also deaths occurring within 48 hours after admission to the hospital.

NURSERIES FOR PREMATURE INFANTS

One of the most important requirements in the care of premature infants is the stabilization of body temperature. This is because their heat regulating systems are not fully developed; the metabolism is low and the infants generally exhibit marked inability to maintain normal body temperatures. The resistance to infection is low and mortality rate high.

Air Conditioning Requirements

The optimum air conditions for the growth and development of these infants were determined by extensive research[8] at the Infants Hospital, Boston, Mass., using four valid criteria, namely, stability of body temperature, gain in weight, incidence of digestive syndromes, and mortality. Individual temperature requirements varied widely (from 72 to

[8]The Premature Infant: A Study of the Effects of Atmospheric Conditions on Growth and on Development, by K. D. Blackfan, C. P. Yaglou and K. McKenzie (*American Journal Diseases of Children*, 46: 1175, 1933).

100 F) according to the constitutional state of the infants and body weights. The optimum relative humidity was about 65 per cent, and the air movement less than 20 fpm.

A single nursery conditioned to 77 F and 65 per cent relative humidity was found to fulfill satisfactorily the requirements of the majority of premature infants. Additional heat for weak (or debilitated) infants may be furnished in the cribs or by means of electric incubators placed inside the conditioned nursery, and the temperature adjusted according to individual requirements. In this way multiplicity of chambers and of air conditioning apparatus is obviated; the infants in the heated beds derive the benefit of breathing cool humid air, and the nurses and doctors need not expose themselves to extreme conditions.

Importance of Humidity: Although external heat is an important factor in the maintenance of normal body temperature, humidity appears to be of equal or greater importance. When the premature nurseries at the Infants Hospital were kept at relative humidity between 25 and 50 per cent for two weeks or longer, the body temperature became unstable, gain in weight diminished, the incidence of gastro-intestinal disturbances increased, and the mortality rose. On the other hand, continuous exposure to air conditions with 55 to 65 per cent relative humidity gave satisfactory results over a period of years. The effect of humidity on mortality is shown in Table 3. The initial physiologic loss of body weight (loss occurring within first four days of life) was found to vary inversely with the humidity. In the old nurseries with natural humidity it averaged 12.4 per cent of the birth weight; in the conditioned nurseries it was 8.9 per cent with 25 to 49 per cent relative humidity, and 6.0 per cent with 50 to 75 per cent relative humidity. The number of days required to regain the birth weight was correspondingly maximum in the old nursery and minimum in the conditioned nurseries under high humidity.

Maximum gains in body weight occurred in the conditioned nurseries under high humidity (55 to 65 per cent) in infants weighing less than 5 lb. The gains were less under low humidity (25 to 50 per cent) in the same nurseries, and in the old nurseries prior to the installation of air conditioning apparatus.

The incidence and severity of digestive syndromes, with diarrhea, persistent vomiting, diminishing gain or loss of body weight, and other symptoms, were generally from two to three times as high under low than under high humidity.

Summarizing, the best chances for life in premature infants are created by maintaining a relative humidity of 65 per cent in the nursery and by providing a uniform environmental temperature just sufficiently high to keep the body temperature within normal limits. Medical and nursing care are, of course, factors of equal and sometimes of greater importance.

Air Conditioning Equipment

Most of the installations now in use are of the central system type providing for filtration, for humidification and heating in cold weather, and for cooling and dehumidification in hot weather. A high ventilation rate, between 15 and 25 air changes, is desirable to remove odors and

maintain uniformity of temperatures in extremes of weather. Recirculation is not used extensively in these wards owing to odors and the possibility of infection.

FEVER THERAPY

Artificial production of fever in man is an imitation of nature's way of overcoming invading pathogenic organisms. The action may be direct and specific by destruction of the invading organism within the safe limit of human temperatures, or indirect in the case of heat resistant organisms, by general mobilization of the defensive mechanisms of the body, which retard or neutralize the activity of pathogenic bacteria and their toxins.

The limits of induced systemic fever are usually between 104 and 107 F (rectal), and the duration from 3 to 8 hours at a time. The total period of fever treatment varies with the type of the organism involved from a few hours to 50 or more.

The diseases which respond favorably to artificial fever therapy are gonorrhea and its complications, (which include arthritis, pelvic infections in women, and involvement of the eye), syphilis, chorea, infectious arthritis (non-gonorrheal), encephalitis, and some forms of asthma. There are other conditions which show promise under this treatment; but the most striking results are seen in gonorrhea and syphilis, since the causative organisms can be destroyed at temperatures compatible with human life[9].

Equipment for Production of Fever

Various means have been tried for producing artificial fever, including injections of various crystalloid or colloid substances, bacterial products of typhoid and malarial organisms; a number of physical methods, such as hot baths, radiant heat, diathermy, radiothermy, and in the last few years, air conditioned chambers. The relative advantages and disadvantages of various methods have been discussed in a paper[10]. The results by the use of air conditioned cabinets have not been fully explored, and it is therefore difficult to determine all the advantages and disadvantages of the value of air conditioning at this time.

In the earlier studies of the Society[11], temperatures were elevated more easily using saturated atmospheres. A fever therapy apparatus[12] using these same principles has proved efficient as a means of inducing and maintaining fever in a body with small likelihood of burns because of the comparatively low dry-bulb temperatures. This saturation factor is in great use today where fever is created by induction currents by

[9]Report of the First Year of Fever Therapy Research by the Department of Industrial Hygiene, School of Medicine, University of Pittsburgh, 1938.

[10]Fever Therapy by Physical Means, by Frank H. Krusen and E. C. Elkins (*Journal American Medical Association*, 112: 1689-1696, April 29, 1939).

[11]A.S.H.V.E. Research Report No. 654—Some Physiological Reactions of High Temperatures and Humidities, by W. J. McConnell and F. C. Houghten (A.S.H.V.E. Transactions, Vol. 29, 1923, p. 129).

[12]A.S.H.V.E Research Report No. 1054—Fever Therapy Induced by Conditioned Air, by F. C. Houghten, M. B. Ferderber and Carl Gutberlet (A.S.H.V.E. Transactions, Vol. 43, 1937, p. 131). A.S.H.V.E. Research Report No. 1162—Fever Therapy Locally Induced by Conditioned Air, by M. B. Ferderber, F. C. Houghten and Carl Gutberlet (A.S.H.V.E. Transactions, Vol. 46, 1940).

placing the body in an electrical field. When the optimum body temperature has been reached by electrical induction, the atmosphere of the enclosure is kept at saturation to prevent heat loss, thus maintaining the patient's temperature at the desired point. Other apparatus[13] which uses electric heaters, centrifugal fans, and a water container for humidification has been used in the past, but the more recent trend is toward saturation with a lower dry-bulb temperature.

When heat is necessary in treating legs or arms, such media as short or long wave diathermy, infra-red, water baths, etc. have been used extensively. A recent development, a saturated atmosphere heating unit, similar to one previously described[14] has proven satisfactory, because heat may be administered over longer periods which render deep heating possible without fear of burns or shocks[15]. Local heating has been somewhat satisfactory in relieving the painful symptoms of peripheral vascular disease.

The final criteria for the use of fever therapy may be changed because of the introduction of certain drugs which appear prominent in the experimental treatment of some diseases for which fever therapy has been efficacious.

COLD THERAPY

In contrast to fever therapy the use of cold as a means of treatment is being investigated. From the available literature[16] the chief virtues of cold therapy (cryotherapy) are the reduction of pain due to extensive cancer and the possibility that the process may be arrested. For a localized lesion, ice water between 36 to 48 F is circulated through tubing at the site of the disease for periods ranging from 4 to 48 hours. A later development was the principle of hibernation during which time the patient is kept in an air conditioned space with an environmental temperature between 50 to 60 F for five days. The body temperature is reduced below the critical level of 95 F to as low as 80 F. Most of the vital processes of life are at very low ebb and this period simulates the hibernation of the wild animals. Although relief of pain is reported it remains to be seen to what extent this form of treatment will be used.

HIGH TEMPERATURE HAZARDS

Heat disease is now classified as heat exhaustion, heat cramps, and heat stroke[17]. The last was formerly thought to be the result of high temperatures, but in the light of present knowledge is considered a neurologic defect. This inability to control temperature is most frequently seen in diseases of the nervous system, and heat stroke, therefore, is not so much a result of environmental temperatures as it is some intrinsic defect in the mechanism itself. The hazards of high temperatures are

[13]Artificial Fever Therapy of Syphilis, by W. M. Simpson (*Journal American Medical Association*, 105: 2132, 1935).

[14]Loc. Cit. Note 11.

[15]Saturated Atmospheres in the Treatment of Injuries, by M. B. Ferderber (*Industrial Medicine*, 8: 256-259, June, 1939).

[16]Temperature Factors in Cancer and Embryonal Cell Growth, by L. W. Smith, and Temple Fay (*Journal American Medical Association*, Vol. 113: 653-660, August, 1939).

[17]Heat Disease: Clinical and Laboratory Studies, by M. W. Heilman and E. S. Montgomery (*Journal of Industrial Hygiene and Toxicology*, 18: 651-666, November, 1936).

not easily understood. It is difficult to say whether a repeated rise of 1 or 2 deg of body temperature is dangerous or whether short exposures at high temperatures are more harmful than longer exposures at lower temperatures. A new concept is evident in finding an increase in leucocytes (white cells) of the blood in workers subjected to high temperatures. These leucocytes are defensive factors which are increased when infection invades a body. A rise in temperature and leucocyte count indicates body defense in the presence of disease. Since a recent study[18] showed that both temperature and cell count were increased, the question arises whether long exposures to very high temperatures might not cause exhaustion of these defense mechanisms.

ALLERGIC DISORDERS

Although there is some division of opinion over the ultimate cause of allergy, the prevailing belief is that it is due to an inherited or acquired hypersensitiveness to pollen or other foreign proteins in certain individuals who react abnormally to the offending substance. The reaction may be induced by inhalation, eating, or absorption (through the skin) of the allergens. Some of the clinical manifestations are hay fever, asthma, eczema, and contact dermatitis.

Symptoms of Hay Fever and Asthma

The respiratory tract is the site of probably the most usual allergic manifestations, the so-called hay fevers and asthma. In hay fevers, the nose and eyes are red and itchy, and there is considerable discharge. Nasal obstruction is the most common and most distressing symptom. The severity of the symptoms varies widely from day to day depending chiefly on the amount of pollen in the air.

Seasonal asthma comes in attacks. The most popular theory concerning the mechanism of action is that the offending substance irritates the nerve endings in mucous membranes of the respiratory tract, causing spasmodic contraction of the small bronchioles of the lungs, which interferes with breathing, particularly with expiration. Non-seasonal allergic disturbances are sometimes attributed to house or street dusts, fungi, odors, animal dander, irritating gases, and heat or cold, particularly sudden temperature changes. It is often stated in the literature that heat regulation in asthmatic individuals is likely unstable, with a tendency toward the subnormal. Many allergic cases who are apparently well, develop their attacks when cold weather appears, or upon changing from warm to cool outdoor air.

Air Conditioning Apparatus

In recent years considerable effort has been directed toward the elimination of the principal cause of allergy from the air of enclosures by filtration or other air conditioning processes capable of removing pollens, in the hope of providing relief to individuals who fail to respond to medical treatment (desensitization or immunization).

Paper or cloth filters, mounted in inexpensive window or floor units, prove quite satisfactory, but since dust and smoke frequently cause

[18]A.S.H.V.E. RESEARCH REPORT No. 1106—Air Conditioning in Industry, by W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten and M. B. Ferderber (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 59).

asthmatic attacks, it is necessary that an air filter, to be of full value in the treatment of asthma, must remove all dusts and pollens regardless of size or amount. An electrostatic cleaner has proved extremely efficient in removing particles of 15 to 20 microns and smaller, besides dusts and smoke[19].

Although the chief remedial factor in the treatment by conditioned air is the filtration of pollen, a certain amount of cooling and dehumidification appears to be desirable. A comfortable temperature between 70 and 75 F and a relative humidity well below 50 per cent proved satisfactory[20]. Direct drafts, overcooling or overheating are apt to initiate or aggrevate the symptoms.

Limitations of Air Conditioning Methods

The results obtained with air filtration or other air conditioning processes in the control of allergic conditions are fairly comparable to those obtained by desensitization treatment so long as the patients remain in the pollen free atmosphere. But while specific desensitization is preventive and in a few instances curative, for all practical purposes filtration gives only temporary relief. With rare exceptions, the symptoms recur on exposure to pollen laden air. Moreover the usefulness of air conditioning methods is limited because all cases are not caused by air-borne substances. Cases of bacterial asthma do not respond at all to the treatment with filtered air.

Despite these limitations air conditioning methods possess definite advantages in the simplicity of treatment, convenience, and under certain conditions almost immediate relief. Pollen cases are usually relieved of most of their symptoms within 1 to 3 hours after exposure to properly filtered air.

A pollen-free atmosphere is especially valuable in cases where desensitization has given little or no relief, and where desensitization is not advisable owing to intercurrent illness. On the whole, conditioning methods are considered to be a valuable adjunct in medical diagnosis and treatment of allergic disorders.

OXYGEN THERAPY

Oxygen therapy is the principal measure employed for preventing and relieving the distressing symptoms of anoxemia, which is a deficiency in the oxygen content of the blood. Some of the more important conditions in which oxygen treatment is believed to be beneficial are pneumonias, anemia, heart affections, post-operative pulmonary disturbances, certain mental disturbances, asphyxia, asthma and atelectasis in new-born infants.

The necessity of air conditioning in oxygen therapy arises from the fact that oxygen is too expensive a gas to waste in the ventilation of oxygen tents and oxygen chambers. The oxygen rich atmosphere in these enclo-

[19]Air Cleaning as an Aid in the Treatment of Hay Fever and Bronchial Asthma, by Leo H. Criep and M. A. Green (*Journal of Allergy*, 7: 120, January, 1936).

[20]The Effect of Low Relative Humidity and Constant Temperature on Pollen Asthma, by B. Z. Rappaport, T. Nelson and W. H. Welker (*Journal of Allergy*, 6: 111, 1935).

sures is therefore reconditioned in a closed circuit by removal of excess heat, moisture, and carbon dioxide given off from the occupants being treated.

Oxygen Tents

In oxygen tents the air enriched with oxygen is usually circulated by means of a small motor blower which sends the air over soda lime to remove carbon dioxide and then over ice to remove excess heat and moisture. The concentration of oxygen in the tent is regulated by means of a pressure reducing valve and flow meter. In an inadequately cooled tent, high temperatures and humidities are inevitable, increasing the discomfort of the patient and imposing an added strain on an already overburdened heart. Oxygen therapy under such conditions may do more harm than good. An ice melting rate of approximately 10 lb per hour gives satisfactory results in patients with fever in a medium size oxygen tent.

Oxygen tents are somewhat confining to the patient; the restless type of person is difficult to control, and the delirious, impossible to control. Medical and nursing care is complicated, as the tent must be opened or removed with attendant loss of oxygen. Oxygen concentrations of 50 per cent or more are difficult to maintain, and it is a problem to keep the temperature and humidity low enough in hot weather. The direct advantages are portability and low cost.

Oxygen Chambers

The conventional oxygen chamber is an air-tight sheet metal enclosure of fire-proof construction, large enough to accommodate one or two patients. Trap doors or curtains are provided for the personnel, food and service, to avoid loss of oxygen. Glass windows in the ceiling and walls admit light from outside the chamber.

The air conditioning system may be of the gravity type, or of the fan type using mechanical refrigeration or air drying agents. The gravity system includes a bank of cooling coils controlled thermostatically, which dehumidify and cool the air. The cool air falls over trays of soda lime at the bottom of the coils, to remove the carbon dioxide given off by the occupants. A heater at the base of the opposite wall warms the air to the desired temperature. Ordinary industrial oxygen is introduced from storage tanks outside the chamber and the concentration is regulated according to the prescription of the physician. The only change of air in the chamber is that taking place by air leakage through the trap doors.

The chief objections to the gravity circulation system are stratification of cold air near the floor and accumulation of odors, which may require the use of activated charcoal, or an excess of oxygen for dilution of the air in the chamber.

The fan circulation systems include compact extended surface coolers, heaters, and sometimes air-drying beds installed outside the chamber for the removal of moisture.

The temperature and humidity requirement in oxygen therapy depend primarily upon the physical condition of the patient, and secondarily

upon the type of disease. In pneumonias, according to Bullowa[21], prescribed conditions should be an effective temperature of 66-68 F, humidity of 50 per cent, air movement of not less than 50 linear feet per minute, oxygen concentration of 50 per cent, and carbon dioxide of less than 1 per cent.

Oxygen chambers are more comfortable than oxygen tents. The patients receive unhampered medical and nursing care, and the oxygen concentration, the temperature and humidity can be adequately controlled at any desired level. The chief disadvantages are high initial and operating costs in comparison with oxygen tents or with the nasal catheter method of oxygen administration. The nasal catheter method is the simplest and most inexpensive of all but it may cause considerable discomfort to the patient and it is not satisfactory for continuous administration nor for restless or delirious patients. Moreover, oxygen concentrations greater than 40 per cent in the inspired air are difficult to maintain, although concentrations as high as 48 per cent have been obtained.

GENERAL HOSPITAL AIR CONDITIONING

Complete conditioning of a large hospital involves a capital investment and running expense which may not be justified. In clean and quiet districts, the requirements of almost all general and private wards during the cool season of the year can be satisfactorily fulfilled by the use of usual heating in conjunction with window air supply and gravity or mechanical exhaust. Insulation against heat and sound is much more important than humidification in winter; it will also help in keeping the building cool in warm weather. Excessive outside noise and dust may require the use of silencers and air filters in the window openings.

Cooling and dehumidification in warm weather are important. In new hospitals particularly, the desirability of cooling certain sections of the building should be given serious consideration. Financial reasons may preclude the cooling of the entire building, but the needs of the average hospital can be met by the use of built-in room coolers and a few portable units which can be wheeled from ward to ward when needed. Objectionable noise is an important drawback to the use of self-contained units, but the difficulty is gradually being overcome by improvements in design.

In the North and certain sections of the Pacific Coast, cooling is needed but a few days during summer, while in the South, it can be used to advantage from May to October, and in tropical climates almost continuously throughout the year.

Aside from comfort and recuperative power of the patients, cooling is of great assistance in the treatment of fevers in the new-born and in post-operative cases, in enteric disorders, fevers, heat stroke, heart failure, and in a variety of other ailments which often accompany summer heat waves.

Considerable research is in progress on the influence of air conditioning upon a wide variety of diseases such as pneumonia, upper respiratory diseases, tuberculosis, arthritis, nervous instability, hyper-thyroidism, essential hypertension, skin diseases, and vascular disorders.

[21]The Management of Pneumonias, by J. G. M. Bullowa, 1937, p. 260.

Chapter 38

TRANSPORTATION AIR CONDITIONING

Railway Passenger Car Ventilation, Method of Air Distribution, Air Cleaning, Winter and Summer Air Conditioning, Humidity and Temperature Control, Summer Air Conditioning for Buses and Automobiles

THE principles of air conditioning used in connection with stationary applications such as stores, restaurants, hospitals, theaters, and homes are in general applicable to such mobile applications as railway passenger cars, passenger buses, automobiles, and ships. However, the equipment used for these mobile applications, with the possible exception of those on board ship, differs from that used for stationary purposes in that it must meet additional requirements. Especially important are the features of compactness with the retention of ready accessibility for quick inspection and servicing, and low weight. Freedom from vibration which could be transmitted to the supporting vehicle and thus to the passengers is essential.

RAILWAY PASSENGER CAR VENTILATION

In non air-conditioned cars, ventilation is accomplished by exhaust fans, roof ventilators, and open doors and windows. This practice provides an ample supply of outside air but does not prevent the entrance of smoke, cinders, and dirt.

An average passenger car contains approximately 5000 cu ft of air and may seat as many as 80 passengers. The occupants are continually liberating heat, carbon dioxide, moisture, odors, and some organic matter from their breath, skin and clothing. The heat and moisture can be removed by cooling and dehumidification, but the other constituents can be successfully handled only by proper ventilation and air cleansing. In the average car from 2000 to 2500 cfm should be circulated by the air conditioning unit. Some of this air may be recirculated, but a portion of it should always be brought in from the outside. The amount of outside air required depends upon the type of car, number of passengers, air temperature, humidity, odors, and whether or not occupants are smoking, and will vary from 15 to 90 per cent of the total air circulated.

Careful attention must be exercised in specifying the rate of outside air taken in so as to fit the type of service adequately and yet not to supply more ventilation than is necessary. Conditioning this outside air is a major factor in determining the size of both summer and winter conditioning equipment. With present average ventilation requirements, about 30 per cent of the cooling equipment and sometimes as high as 50

per cent of the heating equipment is necessary to handle only the outside air load.

For normal conditions, 10 cfm of outside air per passenger is sufficient. When smoking is permitted, at least 15 cfm should be admitted. In some of the dining cars and deluxe sleeping cars, outside air rates as high as 20 and 30 cfm per occupant are used.

Method of Air Distribution

The fact that the amount of space devoted to railway passengers may be as low as 60 cu ft per person (ranging as high as 190 cu ft per person), coupled with the high air flow rates made necessary by severe ventilation and sun loads, makes the problems of air distribution and air delivery in railway cars critical ones.

Various methods may be used to distribute the air delivered to the interior of the car by the circulating fan or blower. The methods commonly used are:

1. A duct lengthwise along the center of the car.
2. One or two side ducts built on the outside of monitor-roofed cars, or on the inside of turtle-backed or arched-roofed cars.
3. Free discharge at the end bulkheads, or by free discharge from a unit placed overhead in the center of the car, discharging toward the ends. This bulkhead delivery system, while inexpensive, is apt to cause complaints due to drafts, and, accordingly, is not being favored.

Delivery grilles and plaques are used, and are often designed to give considerable entrainment and mixing to avoid cool drafts.

Smoking rooms present a special problem. The cloud of smoke that usually hangs near the ceiling can be broken up by having the incoming air directed along the ceiling in all directions at a velocity somewhat higher than that used for the rest of the car. The air should be exhausted from the room by a fan or through a grille to the washroom or lavatory, and then outside by a fan in a ventilator.

For compartments an adjustable supply duct outlet grille of suitable size and design should be provided and provisions made in the door or partition for the removal of the air to be recirculated.

Lower berths in sleeping cars and office cars should be provided with an adjustable air outlet which will discharge the amount of air desired at low velocity in any direction so that the occupant can regulate the ventilation to meet his own requirements.

In cars containing but one or two rooms or compartments, satisfactory results may be obtained by discharging the air directly from the conditioning unit into the upper part of the car. Care must be taken to have a proper discharge velocity. If the velocity is too low, the air will drop before reaching the end of the car and if too high it will discharge against the end bulkhead and be reflected back. Care must be exercised to secure proper circulation, otherwise objectionable drafts will be experienced.

The recirculating air grilles are usually of the straight flow type, and should be located so that objectionable drafts will not be created by the return air. The outside air intakes, located in the car vestibule, on the side of the car, or on the roof of the car, depending upon the location of the cooling coils, should be of ample size to permit the entrance of suf-

ficient outside air. On many of the recently air-conditioned cars, there are no dampers or shutters at the outside air intakes, the percentage of outside air being controlled by blocking the flow through the recirculating grille.

Air Cleaning

All of the air circulated by the blower is filtered before passing over the cooling coils. In some cars the outside and recirculated air are filtered separately before mixing, while on others the air from the two sources is mixed before passing through a common filter. Filters in use are made of metal, wool, cloth, spun glass, hemp, paper, hair, and wire screen. Most filters have a viscous coating of oil for greater cleaning efficiency. Some types may be cleaned, retreated, and returned to service while other types are discarded when dirty.

RAILWAY PASSENGER CAR WINTER AIR CONDITIONING

The majority of cars in service use steam from the locomotive or from a head-end, oil-fired boiler as a source of energy for winter heating. In some instances electrical energy from either a head-end generating set or motive power supply is utilized for resistance heating. In still other cases electrical energy and waste heat from individual car engine-generator sets is employed. The peak heating loads which depend largely upon the amount of insulation used in the car, the type of windows (whether single or double glazed), and the ventilation rate, may vary from 150,000 to 250,000 Btu per hour.

In order to temper the cold outside air, about 30 to 50 per cent of the total heat energy required is distributed by means of finned coils or resistance heaters located in the outside air duct. The remainder is usually transmitted to the car air by finned tubing located along the sides of the car near the floor, thus preventing cold convection currents falling from the car windows from reaching the feet of the passengers.

RAILWAY PASSENGER CAR SUMMER AIR CONDITIONING

Three general types of cooling or refrigerating equipment are being used in the 11,700 railway cars which are now air conditioned in the United States. Of these 3,900 are ice-activated, 1,900 use steam jet systems, and 5,900 employ mechanical compression schemes. These systems which functionally are identical with those used for stationary applications (see Chapter 25) are modified in design to meet the requirements of mobile service. Contrasted with stationary applications of summer conditioning equipment, the use of water as a final means of heat disposal from condensers cannot be resorted to because water in such quantities cannot be transported economically. Accordingly, air cooled or evaporative condensers are always used, with the result that mobile cooling equipments operate at higher temperature, pressure, and power requirement levels than stationary equipment.

The maximum cooling and dehumidifying load which depends largely upon the amount of insulation, the type of windows, the ventilation rate, the sun intensity, and the number of passengers may vary from 60,000 to 96,000 Btu per hour.

An average ice-activated system for such capacities uses about 500 pounds of ice and 1.2 kw per hour. The increase in car weight due to such a system is approximately 8500 lb.

The same service from a steam jet system is obtained with the expenditure of 230 lb of steam and 3.3 kw per hour, with an added weight per car of 11,000 lb.

The mechanical compression systems, all of which use dichlorodifluoromethane as a refrigerant, may be classified by several types depending on the method of driving the compressor. The source of power for driving the compressor (approximately 10 hp) is complicated by the necessity of obtaining this power at all times whether the car is in motion or standing still on the right-of-way or in a terminal where auxiliary power plug-ins are available. In those cases where compressors are driven from car axles, additional refinements in the drive are necessary in order that a nearly constant cooling capacity may be obtained from a variable speed power source. Numerous combinations of electrical generating schemes for generating sufficient electrical energy from the car axle for lighting, ventilation, and summer air conditioning are in use, and their operation is closely interlocked with compressor demands, need for pre-cooling, battery charging, etc. It is difficult therefore to state the additional weight imposed on a car because of such a compression air conditioning system, but it is probably in the vicinity of 6000 lb. These systems, depending mostly upon the locomotive for supplying power for operation, impose a load which may amount to 10 per cent of the capacity of the locomotive.

Several schemes for relieving the locomotive of this compression load are used. Some of the articulated trains, which run as unit equipment—the same cars always in the same train—employ a head-end, engine-generator combination for supplying power to compressor motors. In other cases engine-alternators on individual cars are used to supply alternating current power to compressor motors, as well as to supply all power for car lighting and auxiliaries. Engine-compressor combinations on individual cars provide attractive low weight equipment where continuous engine operation is permissible under all circumstances. Diesel engines and propane engines are used for these purposes, and such engine-driven units have the additional advantage of being able to use waste engine heat either for modulating refrigeration with a reheat cycle or for car heating purposes.

RAILWAY PASSENGER CAR HUMIDITY AND TEMPERATURE CONTROL

The temperature to be maintained in a car depends upon the outside temperature and the humidity desired inside the car. With a low humidity it is necessary to maintain a higher temperature to establish a desirable comfort condition. Little humidity control has been attempted on cars up to the present time. A certain degree of automatic humidity control is secured with cooling, but the relative humidity obtained depends largely upon the temperature of the evaporator, which should be below the dew-point temperature of the air. With certain outside atmospheric conditions it may not be possible to operate the conventional equipment with a sufficiently low evaporator temperature to reduce the humidity

without dropping the temperature too low. One method has been developed whereby the evaporator temperature is carried below the dew-point a sufficient amount to insure dehumidification and then the cold air is heated to the proper temperature by passing it over coils through which part of the high temperature liquid from the condenser is by-passed. Such a system is costly and has not been generally applied. The reheat cycle obtainable from waste engine heat may be used to good advantage in reducing the humidity without reducing the dry-bulb temperature.

During the heating season humidification is desirable from a comfort standpoint, but unless properly controlled, condensation will appear on the windows. A steam or water spray controlled by a humidistat will provide the necessary moisture for humidification. There are several cars with this feature now in use.

Temperature control for the most part obtained by rugged thermostats and relays capable of withstanding vibrations attendant with mobile service is usual equipment.

Manual zone control for varying outdoor conditions, as well as controls which regulate the car temperature automatically in accordance with outdoor conditions, are employed.

Simplified controls from the standpoint of operation by train crews and especially from the servicing viewpoint are very desirable. The control of summer temperatures is accomplished mainly by cycling the complete cooling system; however, modulation is being effected by using multiple evaporators in which a fixed portion may be cut out of the system. In the engine-driven equipments, modulation is obtained by changing engine speed.

For further information on controls, see Chapter 34.

PASSENGER BUS SUMMER AIR CONDITIONING AND VENTILATION

The highways in the United States are now traveled by about 1000 summer air conditioned passenger buses. Many of the facts stressed in connection with the design and installation of summer conditioning equipment in railway cars are even more important in these newer vehicles. Weight and space limitations are more stringent, and the problem of circulating from 900 to 1200 cfm of air in coaches carrying from 25 to 40 passengers with about 35 cu ft of space per passenger without drafts is no easy one.

Some bulkhead delivery systems have been used, and while the over-head package racks have served to break up drafts to some extent, these installations are not gaining in popularity. Longitudinal ducts in the corners above the package racks are sometimes used to carry conditioned air to a series of outlet louvers along the top of the windows. Other designs provide for false spaces below the package racks which serve as ducts to distribute air to either entrainment grilles in the bottom of the racks or distributing slots at the edges of the package racks. Some coaches employ a false ceiling to provide a duct, with delivery taking place from numerous perforations in the ceiling.

Return air grilles and filters are usually located near the rear ceiling where the evaporator is placed. Outside air intakes and filters are located

preferably near the front of the vehicle so as not to contaminate this supply with exhaust fumes and road dust. Of the 30 cfm circulated per person, about 8 to 10 cfm are outside air and the remainder is recirculated. Power for the motor driving the centrifugal fans is obtained from the bus battery.

More recently a coach design has been brought out which provides for a number of return air outlets below the seats; these permit return air to enter a longitudinal duct below the floor. The filters and evaporator are located in this duct near the front of the vehicle. In this instance a central heating coil utilizing waste heat from the coach engine is also located in this duct. Conditioned air is delivered through a pair of vertical ducts to a package rack distribution scheme.

Summer conditioning systems for these vehicles range in cooling capacity from 36,000 to 48,000 Btu per hour. Mechanical compression systems using dichlorodifluoromethane are used, and are powered by water cooled, gasoline engines of approximately 14 hp.

Complete systems add from 800 to 1300 lb to the weight of a coach. Sometimes an auxiliary generator driven by the air conditioning engine is used which serves to help charge the bus battery and thus offsets the power drain imposed by the ventilating blower. Belted reciprocating compressors and direct driven V-type and rotary compressors are used, with engine speeds up to about 1800 rpm. Air cooled condensers for this service require about 5000 cfm of outdoor air, and this is provided by either centrifugal or propellor type fans belted or direct driven by the air conditioning engine. Preventing noise and vibration from affecting passengers is of vital importance. Installations must be made so that quick daily servicing of the engine is possible. In all cases fuel is obtained from the main bus tanks, and in some cases the main engine jacket water cooling system is used to cool the air conditioning engine.

In the more deluxe equipment after the driver has started the air conditioning engine by means of its own cranking motor, the engine speed is modulated automatically as the refrigeration demand is partially met, and if this demand is then fully met, the engine is stopped thermostatically. Restarting when the cooling thermostat is no longer satisfied is accomplished either automatically or manually. The various protective and automatic devices on the refrigerant and engine systems make some of the bus air conditioning control systems quite complicated.

AUTOMOBILE SUMMER AIR CONDITIONING

Recently summer air conditioning has been applied to automobiles. The average present day automobile with little insulation, large, single glazed window areas, and high infiltration and exfiltration losses requires about 15,000 Btu per hour of cooling capacity. One system utilizes a reciprocating compressor belted from the main engine fan shaft thus operating at varying speeds up to 3000 rpm. The resulting refrigeration capacity varies from about 6000 Btu per hour at idling speed to 24,000 Btu per hour at maximum car speed.

A dry air condenser is placed in front of the engine radiator, and the liquid and suction refrigerant lines run back under the car floor to the

evaporator which is located in back of the rear seat. Conditioned air is delivered into the car just above the shelf near the back of the rear seat. A return grille is provided under the rear seat, and the recirculated air is filtered. Outdoor air is provided by infiltration. Power for the air circulating blowers is obtained from the car storage battery. Equipment of this nature increases the car weight approximately 200 lb.

REFERENCES

RAILWAY PASSENGER CARS

Summary Report on Air Conditioning of Railroad Passenger Cars, by Division of Equipment Research, *Association of American Railroads*, November 24, 1936.

Engineering Report on Air Conditioning of Railroad Passenger Cars, by Division of Equipment Research, *Association of American Railroads*, April 15, 1937.

Report on Performance and Cost of Operation of 1937 Internal Combustion Engine Mechanical Compression Equipment for Air Conditioning Railroad Passenger Cars, by Division of Equipment Research, *Association of American Railroads*, May 1, 1937.

Report on Relative Performance of Air Filters, by Mechanical Division, *Association of American Railroads*, January 15, 1938.

Air Conditioning of Railroad Passenger Cars, by L. W. Wallace and G. G. Early, Jr. (*A.S.M.E. Transactions*, November, 1937).

Improved Methods of Railroad Air Conditioning, by A. E. Voigt (*Refrigerating Engineering*, July, 1940, p. 16).

BUSES AND AUTOMOBILES

Iso-Thermal Manifold and the Conditioned-Air Unit Used on Nash Cars, by E. Monson (*Society of Automotive Engineers Journal*, January, 1938).

Summer and Winter Air Conditioning of Automobiles and Buses, by L. W. Child (*Heating and Ventilating*, March, 1938, p. 38).

Air Conditioning Takes to the Highways (*Heating, Piping and Air Conditioning*, May, 1938, p. 314).

Fleet of Air Conditioned Buses West Carries Demountable Units (*Heating and Ventilating*, May, 1938, p. 30).

Conditioned Air for Automobiles (*Heating and Ventilating*, May, 1938, p. 44).

Air Conditioning of Automobiles and Buses, by L. W. Child (*Society of Automotive Engineers Journal*, June, 1938).

Bus Air Conditioning, by Jerry Hicke (*Heating, Piping and Air Conditioning*, October, 1938, p. 639).

Bus Air Conditioning, by A. J. Mallinckrodt and Lars Hanson (*Refrigerating Engineering*, June, 1939, p. 388).

Problems in Air Conditioning Automobiles, by F. J. Linsenmeyer (*Society of Automotive Engineers Journal*, July, 1939).

Packard Offers Automobile Air Conditioning System (*Heating, Piping and Air Conditioning*, December, 1939, p. 756).

AIRPLANES

Flight in the Sub-stratosphere, by J. C. Edgerton (*Society of Automotive Engineers Journal*, 39:484-6 December, 1936).

Comfort Aloft (*Heating and Ventilating*, October, 1937, p. 40).

Portable Climate, American Buys 12 Plane Air Conditioning Units (*Aviation*, 36:59, September, 1937).

Steam Heating in the Sky (*Heating, Piping and Air Conditioning*, October, 1938, p. 631).

Steam Heating for Modern Air Transports, by Jack McQuire (*Heating and Ventilating*, February, 1939, p. 37).

Altitude Conditioning (*Heating and Ventilating*, August, 1940, p. 16).

SHIPS

Ventilation and Air Conditioning of the S. S. Panama (*Heating and Ventilating*, September, 1939, p. 47).

Air Conditioning the New Mauretania (*Heating, Piping and Air Conditioning*, July, 1939, p. 431).

Heating, Ventilating and Air Conditioning on Shipboard, by J. H. Clarke (*Heating, Piping and Air Conditioning*, August, p. 467; September, p. 529; October, p. 610, 1940).

Chapter 39

INDUSTRIAL AIR CONDITIONING

Atmospheric Conditions Required, General Requirements, Classification of Problems, Control of Regain, Moisture Content and Regain, Conditioning and Drying, Control of Rate of Chemical Reaction, Control of Rate of Biochemical Reactions, Control Rate of Crystallization, Elimination of Static Electricity

IN the application of air conditioning to industrial processes, too much stress cannot be laid upon a thorough understanding by the air conditioning engineer of the problems involved. A complete knowledge of these problems is necessary before a satisfactory design can be made. Individual processes and machines are changing rapidly and air conditions must be constantly revised to meet the new conditions.

ATMOSPHERIC CONDITIONS REQUIRED

The most desirable relative humidity during processing depends upon the product and the nature of the process. As far as the behavior of the material itself and its desired final condition are concerned, each material and process presents a different problem. The best relative humidity may range up to 100 per cent. Similarly the most desirable temperature may range between wide limits for different materials and treatments. Extremes in either relative humidity or temperature require relatively expensive equipment for maintaining these conditions automatically. In departments where people are working, their health, comfort, and productive efficiency must be considered and often a compromise between the optimum conditions for processing and those required for the comfort of the worker is desirable.

It is generally considered that relative humidities below 40 per cent are on the dry side, conducive to low regains, a brittle condition of fibrous materials, prevalence of static electricity, and a tendency toward dryness of the skin and membranes of human beings. At the other end of the scale, humidities above 80 per cent are relatively damp, conducive to high regains, extreme softness, and pliability.

Table 1 lists desirable temperatures and humidities for industrial processing. In using this table, care must be taken in qualifying the process. In preparing many materials, conditions are not maintained constantly, but different temperatures and humidities are held for varying lengths of time.

TABLE 1. DESIRABLE TEMPERATURES AND HUMIDITIES FOR INDUSTRIAL PROCESSING

INDUSTRY	PROCESS	TEMPERATURE DEGREES FAHRENHEIT	RELATIVE HUMIDITY PER CENT
AUTOMOBILE	Assembly line	65	40
BAKING	Cake icing	70	50
	Cake mixing	75	65
	Dough fermentation room	80	76 to 80
	Loaf cooling	70	60 to 70
	Make-up room	75 to 80	55 to 70
	Mixing room	75 to 80	55 to 70
	Paraffin paper wrapping	80	55
	Proof boxes	80 to 90	80 to 95
	Storage of flour	70 to 80	60
	Storage of yeast	28 to 40	60 to 75
BIOLOGICAL PRODUCTS	Vaccines	below 32	
	Antitoxins	38 to 42	
BREWING	Fermentation in vat room	44 to 50	50
	Storage of grains	60	30 to 45
CERAMIC	Drying of auger machine brick	180 to 200	
	Drying of refractory shapes	110 to 150	50 to 60
	Molding room	80	60
	Storage of clay	60	35
CHEMICAL	General storage	60 to 80	35 to 50
CONFECTIONERY	Chewing gum rolling	75	50
	Chewing gum wrapping	70	45
	Chocolate covering	62 to 65	50 to 55
	Hard candy making	70 to 80	30 to 50
	Packing	65	50
	Starch room	75 to 85	50
	Storage	60 to 68	50 to 65
DISTILLERY	General manufacture	60	45
	Storage of grains	60	30 to 45
DRUG	Storage of powders and tablets	70 to 80	30 to 35
ELECTRICAL	Insulation winding	104	5
	Manufacture of cotton covered wire	60 to 80	60 to 70
	Manufacture of electrical windings	60 to 80	35 to 50
	Storage of electrical goods	60 to 80	35 to 50
FOOD	Butter making	60	60
	Dairy chill room	40	60
	Preparation of cereals	60 to 70	38
	Preparation of macaroni	70 to 80	38
	Ripening of meats	40	80
	Slicing of bacon	60	45
	Storage of apples	31 to 34	75 to 85
	Storage of citrus fruit	32	80
	Storage of eggs in shell	30	80
	Storage of meats	0 to 10	50
	Storage of sugar	80	35
FUR	Drying of furs	110	
	Storage of furs	28 to 40	25 to 40

TABLE 1. DESIRABLE TEMPERATURES AND HUMIDITIES FOR INDUSTRIAL PROCESSING (Concluded)

Industry	Process	Temperature Degrees Fahrenheit	Relative Humidity Per Cent
Incubators	Chicken	99 to 102	55 to 75
Laboratory	General analytical and physical	60 to 70	60 to 70
	Storage of materials	60 to 70	35 to 50
Leather	Drying of hides	90	
	Mulling	95 to 100	95
Library	Book storage (see discussion in this chapter)	65 to 70	38 to 50
Linoleum	Printing	80	40
Matches	Manufacturing	72 to 74	50
	Storage of matches	60	
Munitions	Fuse loading	70	55
Paint	Air drying lacquers	70 to 90	25 to 50
	Baking lacquers	180 to 300	
	Air drying of oil paints	60 to 90	25 to 50
Paper	Binding, cutting, drying, folding, gluing	60 to 80	40 to 60
	Storage of paper	60 to 80	40 to 60
	Testing Laboratory	60 to 80	55 to 65
Photographic	Development of film	70 to 75	60
	Drying	75 to 80	50
	Printing	70	70
	Cutting	72	65
Printing	Binding	70	45
	Folding	77	65
	Press room (general)	75	60 to 78
	Press room (lithographic)	60 to 75	20 to 60
	Storage of rollers	70 to 90	50 to 55
Rubber	Manufacturing	90	
	Dipping of surgical rubber articles	75 to 80	25 to 30
	Standard laboratory tests	80 to 84	42 to 48
Soap	Drying	110	70
Textile	Cotton— carding	75 to 80	50 to 55
	combing	75 to 80	60 to 65
	roving	75 to 80	50 to 60
	spinning	60 to 80	50 to 70
	weaving	68 to 75	85
	Rayon— spinning	70	85
	throwing	70	60
	weaving	75 to 88	60 to 75
	Silk— dressing	75 to 80	60 to 65
	spinning	75 to 80	65 to 70
	throwing	75 to 80	65 to 70
	weaving	75 to 80	60 to 70
	Wool— carding	75 to 80	65 to 70
	spinning	75 to 80	55 to 60
	weaving	75 to 80	50 to 55
	Testing Laboratory	70	65
Tobacco	Cigar and cigarette making	70 to 75	55 to 75
	Softening	90	85
	Stemming or stripping	75 to 85	70

GENERAL REQUIREMENTS

In general, air conditioning apparatus for industrial purposes must be capable of absorbing heat from various sources such as machinery power, electric lights, people, sunlight and chemical reaction; of warming or cooling to any desired temperature, and of providing ample air supply at all times. Refrigeration may or may not be required, depending upon natural conditions, the required relative humidity and the maximum permissible temperature. Washing, purifying and recirculating of the air may be desirable. Good distribution is essential for the control of air motion and for the prevention of uneven conditions. Accurate, sensitive and reliable automatic control of humidity or temperature, or both, is vital in most cases.

Ordinarily, outside weather conditions and the ventilation required for workers are of secondary importance in relation to the total work to be done by the air conditioning system. In extreme cases of high concentration of industrial heat from machinery and ovens the error of entirely omitting the heat gain through the building structure would not be serious. At the other extreme, where low temperatures must be produced with refrigeration and where comparatively little power is used for driving the machinery, the heat gain through the building structure will become the major factor in determining the size of equipment and in this case the ventilation requirement assumes a normal degree of importance.

Buildings which are to be air conditioned should therefore be designed with careful consideration of overall cost and efficiency. Condensation resulting from high humidities must be prevented by suitable materials and construction, or else collected and drained to prevent loss of product or quick deterioration of the structure. Air leakage or filtration may add greatly to operating costs or make the maintenance of low humidities (relative or absolute) wholly impossible. Low temperatures require good insulation.

It is apparent that the subject of air conditioning for industrial processes is extensive and greatly involved, and that a detailed treatment is therefore beyond the scope of this book. A few of the salient points of the general subject are covered in this chapter.

CLASSIFICATION OF PROBLEMS

In general, any industrial air conditioning problem may be listed under one or more of the following five classes:

1. Control of Regain.
2. Control of Rate of Chemical Reactions.
3. Control of Rate of Biochemical Reactions.
4. Control of Rate of Crystallization.
5. Elimination of Static Electricity.

CONTROL OF REGAIN

In the manufacture or processing of hygroscopic materials such as textiles, paper, wood, leather, tobacco and foodstuffs, the temperature and relative humidity of the air have a marked influence upon the rate of production and upon the weight, strength, appearance and general

quality of the product. This influence is due to the fact that the moisture content of materials having a vegetable or animal origin, and to a lesser extent minerals in certain forms, comes to equilibrium with the moisture of the surrounding air.

In industries where the physical properties of a product affect its value, the percentage of moisture is of special importance. With increase in moisture content, hygroscopic materials ordinarily become softer and more pliable. Standards of regain are firmly fixed in trade with fair penalties for excesses. Deficiencies result in loss of revenue to seller and loss of desirable quality to buyer.

Manufacturing economy therefore requires that the moisture content be maintained at a percentage favorable to rapid and satisfactory manipulation and to a minimum loss of material through breakage. A uniform condition is desirable in order that high speed machinery may be adjusted permanently for the desired production with a minimum loss from delays, wastage of raw material and defective product.

In the processing of hygroscopic materials, it is usually necessary to secure a final moisture content suitable for the goods as shipped. Where the goods are sold by weight, it is proper that they contain a normal or standard moisture content.

MOISTURE CONTENT AND REGAIN

The terms *moisture content* and *regain* refer to the amount of moisture in hygroscopic materials. *Moisture content* is the more general term and refers either to free moisture (as in a sponge) or to hygroscopic moisture (which varies with atmospheric conditions). It is usually expressed as a percentage of the total weight of material. *Regain* is more specific and refers only to hygroscopic moisture. It is expressed as a percentage of the *bone-dry* weight of material. For example, if a sample of cloth weighing 100.0 grains is dried to a constant weight of 93.0 grains, the loss in weight, or 7.0 grains, represents the weight of moisture originally contained. This expressed as a percentage of the total weight (100.0 grains) gives the moisture content or 7 per cent. The regain, which is expressed as a percentage of the bone-dry weight, is $\frac{7.0}{93.0}$ or 7.5 per cent.

The use of the term *regain* does not imply that the material as a whole has been completely dried out and has re-absorbed moisture. During the processing of certain textiles, for instance, complete drying during manufacturing is avoided as it might appreciably reduce the ability of the material to re-absorb moisture. A basis for calculating the regain of textiles is obtained by drying under standard conditions a sample from the lot and the dry weight thus obtained is used as a basis in the calculations to determine the regain.

The moisture content of an hygroscopic material at any time depends upon the nature of the material and upon the temperature and especially the relative humidity of the air to which it has been exposed. Not only do different materials acquire different percentages of moisture after prolonged exposure to a given atmosphere, but the rate of absorption or drying out varies with the nature of the material, its thickness and density.

TABLE 2. REGAIN OF HYGROSCOPIC MATERIALS

Moisture Content Expressed in Per Cent of Dry Weight of the Substance at Various Relative Humidities—Temperature, 75 F

Classification	Material	Description	Relative Humidity—Per Cent 10	20	30	40	50	60	70	80	90	Authority
Natural Textile Fibers	Cotton	Sea island—roving	2.5	3.7	4.6	5.5	6.6	7.9	9.5	11.5	14.1	Hartshorne
	Cotton	American—cloth	2 6	3.7	4.4	5.2	5.9	6.8	8.1	10.0	14.3	Schloesing
	Cotton	Absorbent	4 8	9 0	12.5	15.7	18.5	20.8	22.8	24.3	25.8	Fuwa
	Wool	Australian merino—skein	4.7	7.0	8.9	10.8	12 8	14.9	17.2	19.9	23.4	Hartshorne
	Silk	Raw chevennes—skein	3.2	5.5	6.9	8.0	8 9	10.2	11.9	14.3	18.8	Schloesing
	Linen	Table cloth	1.9	2.9	3.6	4 3	5.1	6.1	7.0	8.4	10.2	Atkinson
	Linen	Dry spun—yarn	3.6	5.4	6.5	7.3	8.1	8.9	9.8	11.2	13.8	Sommer
	Jute	Average of several grades	3.1	5.2	6.9	8.5	10.2	12.2	14.4	17.1	20.2	Storch
	Hemp	Manila and sisal—rope	2.7	4.7	6.0	7.2	8.5	9.9	11.6	13.6	15.7	Fuwa
Rayons	Viscose Nitrocellulose Cupramonium	Average skein	4.0	5.7	6.8	7.9	9.2	10.8	12.4	14.2	16 0	Robertson
	Cellulose Acetate	Fibre	0.8	1.1	1.4	1.9	2.4	3.0	3.6	4.3	5 3	Robertson
Paper	M. F. Newsprint	Wood pulp—24% ash	2.1	3.2	4.0	4.7	5.3	6.1	7.2	8.7	10.6	U. S. B. of S.
	H. M. F. Writing	Wood pulp—3% ash	3.0	4.2	5.2	6.2	7.2	8.3	9.9	11.9	14.2	U. S. B. of S.
	White Bond	Rag—1% ash	2.4	3.7	4.7	5.5	6.5	7.5	8.8	10.8	13.2	U. S. B. of S.
	Com. Ledger	75% rag—1% ash	3.2	4.2	5.0	5.6	6.2	6.9	8.1	10.3	13.9	U. S. B. of S.
	Kraft Wrapping	Coniferous	3.2	4.6	5.7	6.6	7.6	8.9	10.5	12.6	14.9	U. S. B. of S.
Misc. Organic Materials	Leather	Sole oak—tanned	5.0	8.5	11 2	13.6	16.0	18.3	20.6	24.0	29.2	Phelps
	Catgut	Racquet strings	4.6	7 2	8.6	10.2	12.0	14.3	17.3	19.8	21.7	Fuwa
	Glue	Hide	3.4	4.8	5.8	6.6	7 6	9.0	10.7	11.8	12.5	Fuwa
	Rubber	Solid tire	0.11	0.21	0.32	0.44	0.54	0.66	0.76	0.88	0.99	Fuwa
	Wood	Timber (average)	3.0	4.4	5.9	7.6	9.3	11.3	14.0	17.5	22.0	Forest P. Lab.
	Soap	White	1.9	3.8	5.7	7.6	10.0	12.9	16.1	19.8	23.8	Fuwa
	Tobacco	Cigarette	5.4	8.6	11.0	13.3	16.0	19.5	25.0	33.5	50.0	Ford
Foodstuffs	White Bread		0.5	1.7	3.1	4.5	6.2	8.5	11.1	14.5	19.0	Atkinson
	Crackers		2.1	2.8	3.3	3.9	5.0	6.5	8.3	10.9	14.9	Atkinson
	Macaroni		5.1	7.4	8.8	10.2	11.7	13.7	16.2	19.0	22.1	Atkinson
	Flour		2.6	4.1	5.3	6.5	8.0	9.9	12.4	15.4	19.1	Bailey
	Starch		2.2	3.8	5.2	6.4	7.4	8.3	9.2	10.6	12.7	Atkinson
	Gelatin		0.7	1.6	2.8	3.8	4.9	6.1	7.6	9.3	11.4	Atkinson
Misc. Inorganic Materials	Asbestos Fiber	Finely divided	0.16	0.24	0.26	0.32	0.41	0.51	0.62	0.73	0.84	Fuwa
	Silica Gel		5.7	9.8	12.7	15.2	17.2	18.8	20.2	21.5	22.6	Fuwa
	Domestic Coke		0.20	0.40	0.61	0.81	1.03	1.24	1.46	1.67	1.89	Selvig
	Activated Charcoal	Steam activated	7.1	14.3	22.8	26.2	28.3	29.2	30.0	31.1	32.7	Fuwa
	Sulphuric Acid	H_2SO_4	33.0	41.0	47.5	52.5	57.0	61.5	67.0	73.5	82.5	Mason

Table 2 shows the regain or hygroscopic moisture content of several organic and inorganic materials when in equilibrium at a dry-bulb temperature of 75 F and various relative humidities. The effect of relative humidity on regain of hygroscopic substances is clearly indicated. The effect of temperature is comparatively unimportant. In the case of cotton, for instance, an increase in temperature of 10 deg has the same effect on regain as a decrease in relative humidity of one per cent. Changes in temperature do, however, affect the rate of absorption or drying. Sudden changes in temperature cause temporary fluctuations in regain even when the relative humidity remains stationary.

The regain or moisture content affects the physical properties of textiles to a marked degree, changing the strength, pliability and elasticity.

The fact that the regain of textiles will come into equilibrium with the conditions of the surrounding air and vary with its temperature and relative humidity is the fundamental basis for the control of physical qualities during manufacture. During the preparation processes in a cotton mill, the cotton fibers should be in a condition to be easily carded.

These preliminary processes are carried out best in a relative humidity of 50 to 55 per cent. As the cotton fiber comes to the spinning operation, more flexibility is needed and the relative humidity is increased in this department. For many years, 65 per cent relative humidity was considered the optimum. To offset the extra work performed on the fiber as the spindle speed is increased, many cotton mills now carry 70 per cent relative humidity in the spinning rooms.[1] Winding, warping and weaving are all processes calling for great flexibility and a consequent need for higher humidity.

Other textile fibers, due to their different natural characteristics, are processed under relative humidities and temperatures applicable to each.

Rayons, on account of great loss of strength with the higher regains, should be processed in a relative humidity of 55 to 70 per cent. Acetate silk, another chemical fiber, with approximately 50 per cent of the regain of rayon, may be processed between 60 and 65 per cent relative humidity.

All hygroscopic materials release sensible heat equivalent to the latent heat of the moisture absorbed by the material, all of which may account for a small percentage of the total heat load.

CONDITIONING AND DRYING

In general, the exposure of materials to desirable conditions for treatment may be coincidental with the manufacture or processing of the materials, or they may be treated separately in special enclosures. This latter treatment may be classified as conditioning or drying. The purpose of conditioning or drying is usually to establish a desired condition of moisture content and to regulate the physical properties of the material.

When the final moisture content is lower than the initial one, the term *drying* is applied. If the final moisture content is to be higher, the process is termed *conditioning*. In the case of some textile products and tobacco,

[1]The Present Status of Textile Regain Data, by A. E. Stacey, Jr. (*National Association of Cotton Manufacturers*, 1927).

for example, drying and conditioning may be combined in one process for the dual purpose of removing undesirable moisture and accurately regulating the final moisture content. Either conditioning or drying are frequently made continuous processes in which the material is conveyed through an elongated compartment by suitable means and subjected to controlled atmospheric conditions.

CONTROL OF RATE OF CHEMICAL REACTIONS

A typical example of the second general classification, that is the control of the rate of chemical reactions, occurs in the manufacture of rayon. The pulp sheets are conditioned, cut to size, and passed through a mercerizing process. It is essential that during this process close control of both temperature and relative humidity should be maintained. Temperature controls the rate of reaction directly, while the relative humidity maintains a constant rate of evaporation from the surface of the solution and gives a solution of known strength throughout the mercerizing period.

Another well-known example of this class is the *drying* of varnish which is an oxidizing process dependent upon temperature. High relative humidities have a retarding action on the rate of oxidization at the surface and allow the gases to escape as the chemical oxidizers *cure* the varnish film from the bottom. This produces a surface free from bubbles and a film homogeneous throughout.

Desirable temperatures for *drying* varnish vary with the quality. A relative humidity of 65 per cent is beneficial for obtaining the best processing results.

CONTROL OF RATE OF BIOCHEMICAL REACTIONS

In the field of biochemical control, industrial air conditioning has been applied to many different and well-known products. All problems involving fermentation are classed under this heading. As biochemistry is a subdivision of chemistry, subject to the same laws, the rate of reaction may be controlled by temperature. An example of this is the dough room of the modern bakery. Yeast develops best at a temperature of 80 F. A relative humidity of 65 per cent is maintained so as to hold the surface of the dough open to allow the CO_2 gases formed by the fermentation to pass through and produce a loaf of bread, when baked, of even, fine texture without large voids.

Another example of a similar process is found in the curing of macaroni. The flour and water mixture is fermented and dried. As it is necessary to have a definite amount of water present to carry on a fermentation process, the moisture must be removed in a relatively short period to stop fermentation and prevent souring and in such a manner as to avoid setting up internal strains in the mixture. Best results are obtained with the correct cycles of both temperature and humidity.

The curing of fruits, such as bananas and lemons, also comes under this classification. Bananas are treated somewhat differently and to accomplish the required results, a cycle of temperatures and relative humidities is used. The starches in the pulp of the fruit must be changed and the

skin cured and colored, after which the fruit is cooled to maintain as low a rate of metabolism as possible. Ideal conditions range between 55 to 57 F and in no case should the temperature go below 49 F, as the starches then become fixed and are indigestible.

The curing of lemons is an entirely different problem. Bananas are cured for a quick market, while lemons are held for a future market. The process, therefore, varies in the temperature used. Temperatures from 54 to 59 F have been found to be best suited for this process. A high relative humidity of 88 to 90 per cent is necessary to hold shrinkage to a minimum and, at the same time, develop the rind so it will be sufficiently tough to permit handling.

Tobacco from the field to the finished cigar, cigarette, plug or pipe tobacco, offers another interesting example of what may be done by industrial air conditioning in the control of color, texture and flavor. In the processing of tobacco, the first three classifications of air conditioning are involved, and only through close atmospheric control can the best quality of the leaf be developed.

CONTROL RATE OF CRYSTALLIZATION

The rate of cooling of a saturated solution determines the size of the crystals formed. Both temperature and relative humidity are of importance, as the one controls the rate of cooling, while the other, through evaporation, changes the density of the solution.

In the coating pans for pills, gum and nuts, a heavy sugar solution is added to the tumbling mass. As the water evaporates, each separate piece is covered with crystals of sugar. A smooth, opaque coating is only accomplished by blowing into the kettle the proper amount of air at the right temperature and relative humidity. If the cooling and drying is too slow, the coating will be rough and semi-translucent, and the appearance unsightly; if too fast, the coating will chip through to the interior. Only by balancing temperature, relative humidity, and volume of air to the sugar solution, can the proper rate be obtained and a perfect coating assured.

ELIMINATION OF STATIC ELECTRICITY

The presence of static electricity is very detrimental to the satisfactory and economical processing of many light materials, such as textile fibers, paper, etc. It is also extremely dangerous where explosive atmospheres or materials are present. Fortunately, this hazard is easily eliminated by increasing the relative humidity.

In attempting to eliminate static electricity, it must be borne in mind that for successful elimination the air that actually comes in contact with the material in the machine must be at a relative humidity of 45 per cent or more. As some machines consume a great deal of power which is converted directly into heat, the temperature in the machine may be considerably higher than the temperature adjacent to the machine where the relative humidity is normally measured. In such cases, the relative humidity in the machine will be appreciably lower than that elsewhere in the room, and it may be necessary to maintain a room

relative humidity of 65 per cent, or even more, before the desired results can be obtained.

CALCULATIONS

The methods for determining the proper heating and cooling loads for the various industrial processes are similar to those outlined in Chapters 6 and 7. Because of the large number of motors and heat processing units usually prevalent in an industrial application, it is particularly important that operating allowances for the latent and sensible heat loads be definitely ascertained and used in the calculations to determine the total design load.

REFERENCES

Effect of Air Conditioning upon Munitions, by J. I. Lyle (A.S.H.V.E. TRANSACTIONS, Vol. 23, 1917, p. 383).

Air Conditioning for Sausage Manufacturing Plants, by M. G. Harbula (A.S.H.V.E. TRANSACTIONS, Vol. 28, 1922, p. 343).

Air Conditioning and Refrigerating Large Bakeries, by W. L. Fleisher (*Heating, Piping and Air Conditioning*, December, 1929, p. 621; January, 1930, p. 24).

Air Conditioning for Textile Plants Making and Using Synthetic Yarns, by L. L. Lewis (*Rayon Textile Monthly*, July, August, September, 1930).

Air Conditioning in the Bakery, by W. L. Fleisher (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, February, 1931, p. 158).

Air Conditioning in Industrial Processes (Match Factory), by W. L. Fleisher (*Heating, Piping and Air Conditioning*, March, 1931, p. 196).

Air Conditioning of Press Rooms, by I. C. Baker (*Heating, Piping and Air Conditioning*, July, 1931, p. 553).

Pre-Cooling Fruits and Vegetables with Circulating Air, by C. E. Baker (*Heating, Piping and Air Conditioning*, January, 1932, p. 42).

Air Conditioning Maintains Quality of Fruits and Vegetables, by C. E. Baker (*Heating, Piping and Air Conditioning*, August, 1935, p. 369).

Air Condition the Bakery Throughout, by W. W. Reece (*Heating, Piping and Air Conditioning*, August, 1936, p. 149).

Air Conditioning as Applied in Theatres and Film Laboratories, by D. C. Lindsay (*Transactions Society of Motion Picture Engineers*, April, 1927, Vol. XI, No. 30, p. 335-365).

Air Conditioning Requirements of Multicolor Offset Printing, by C. G. Weber (*Refrigerating Engineering*, December, 1936, p. 6).

Banana Ripening Manual, Circular No. 14, Equipment Department, Fruit Dispatch Co., New York, N. Y.

The Commercial Storage of Fruits, Vegetables and Florists' Stocks, by D. H. Rose, R. C. Wright and T. M. Whiteman (*U. S. Department of Agriculture*, Circular No. 278).

Reactions of Lithographic Papers to Variations in Humidity and Temperature, by C. G. Weber and L. W. Snyder (*U. S. Bureau Standards Journal Research*, January, 1934).

Relation of Air Conditions to Tobacco Curing, by J. Johnson and W. B. Ogden (*Wisconsin Agricultural Research Bureau* 110: 1-48, 1931).

Temperature Studies of Some Tomato Pathogens, by Alice A. Nightingale and G. W. Ramsey (*U. S. Department of Agriculture Technical Bulletin* No. 520, August, 1936).

The Treatment of Offset Papers for Optimum Register, by C. G. Weber and M. N. V. Geib (*U. S. Bureau Standards Journal Research*, February, 1936).

Summary Report of National Bureau of Standards Research on Preservation of Records, by A. E. Kimberly and B. W. Scribner (*U. S. Bureau of Standards Miscellaneous Publication*, 154, March 16, 1937).

Chapter 40

INDUSTRIAL EXHAUST SYSTEMS

Classification of Systems, Design Procedure, Requirements for Suction and Velocity, Hoods, Design of Duct Systems, Collectors, Resistance of Systems, Efficiency of Exhaust Systems, Selection of Fans and Motors, Corrosion

IN almost every industry some type of exhaust or collecting system is essential to achieve efficient and economical control of dusts and fumes. General design information is included in this chapter which is intended to relate primarily to factory exhaust systems.

CLASSIFICATION OF SYSTEMS

There are two general arrangements, the central and the group systems. In the central system a single or double fan is located near the center of the shop with a piping system radiating to the various machines to be served. In the group system, which is sometimes employed where the machines to be served are widely scattered, small individual exhaust fans are located at the center of the machine groups. The group arrangement has the advantage of flexibility.

Exhaust systems are also classified by the means employed to collect dust or other material handled. The dust or refuse may be collected and controlled by enclosing hoods, open hoods, inward air leakage, or by exhausting the general air of the room.

With some classes of machinery it is not feasible to closely hood the machines and in these cases open hoods over or adjacent to the machines are provided to collect as much as possible of the dust and fumes. This class includes such machines as rubber mills, package filling machinery, sand blast, crushers, forges, pickling tanks, melting furnaces, and the unloading points of various types of conveyors.

The open hoods should be placed as close to the source of dust or fumes as possible, with due regard to the movements of the operator. When the hood must be placed at some distance above the machine it should be large enough to encompass an area of considerable extent as diffusion is usually quite rapid.

Consideration must also be given to the natural movement of the fumes. For those that are lighter than air the hood should be over or above the machine and where a heavy vapor or dust-laden air at ordinary temperature is to be removed, horizontal or floor connections are required. If it is attempted to remove heavy dust such as lead oxides by an overhead hood the conditions may be worse than if no exhaust were used at all, owing to the rising air current carrying the dust up through the

breathing zones. The objective to keep in mind in all cases is to take advantage of the natural tendency of the material to move upward or downward.

In another class of operation the main objective is to prevent the escape of dust into the surrounding atmosphere, the removal of some dust from the machine or enclosure being merely incidental. The dust-creating apparatus is enclosed within a housing which is made as tight as practicable, and sufficient suction is applied to the enclosure to maintain an inward air leakage, thus preventing escape of the dust. While the exhaust system is required to handle only the air which leaks in through the crevices and openings in the enclosure, yet in many installations leakages are very high and great care is required to obtain satisfactory results with a system of this kind. The inward-leakage principle is utilized for controlling dust in the operating of tumbling barrels, grinding, screening, elevating, and similar processes.

Certain dust and fume producing operations are best carried on by isolating the process in a separate compartment or room and then applying general ventilation to this space. The compartment or room in which the work is performed should be as small as is consistent with convenience in handling the work. The ventilating system should be designed so that a strong current of clean air is drawn across the operator, and away from him toward the work, where the dust is picked up and carried from the room.

DESIGN PROCEDURE

The first step in the design of an exhaust system is to determine the number and size of the hoods and their connections. No general rules, however, can be given since hood and duct dimensions are determined by the characteristics of the operations to which they are applied. When a tentative decision regarding the set-up has been made, it is then necessary to obtain the suction and air velocities required to effect control. At this point the designer must rely upon the prevailing practice and on such physical data relating to hoods, duct systems and collectors as are available. Finally, in choosing the fan, the area of the intake should be equal to or greater than the sum of the areas of the branch ducts. The speed, of course, must be sufficient to maintain the estimated suction and air velocities in the system. In general, the most important requirements of an efficient exhaust and collecting system are as follows[1]:

1. Hoods, ducts, fans and collectors should be of adequate size.
2. The air velocities should be sufficient to control and convey the materials collected.
3. The hoods and ducts should not interfere with the operation of a machine or any working part.
4. The system should do the required work with a minimum power consumption.
5. When inflammable dusts and fumes are conveyed, the piping should be provided with an automatic damper in passing through a fire-wall.
6. Ducts and all metal parts should be grounded to reduce the danger of dust explosions by static electricity.
7. The design of an exhaust system should afford easy access to parts for inspection and care.

[1]For more detailed requirements see Safe Practice Pamphlets Nos. 32 and 37, published by the *National Safety Council*, Chicago.

REQUIREMENTS FOR SUCTION AND VELOCITY

The removal of dust or waste by means of an exhaust hood requires a movement of air at the point of origin sufficient to carry it to a collecting system. The air velocities necessary to accomplish this depend upon the physical properties of the material to be eliminated and the

TABLE 1. SIZE OF CONNECTIONS FOR WOOD-WORKING MACHINERY

TYPE OF MACHINE	DIAMETER OF CONNECTIONS IN INCHES
Circular saws, 12-in. diam	4
Circular saws, 12-24 in. diam	5
Circular saws, 24-40 in. diam	6
Band saws, blade under 2 in. wide	4
Band saws, blade 2-3 in. wide	5
Band saws, blade 3-4 in. wide	6
Band saws, blade 4-5 in. wide	7
Band saws, blade 5-6 in. wide	8
Small mortisers	6
Single end tenoners	6
Double end tenoners	7
Double end, double head tenoners	10
Planers, matchers, moulders, stickers, jointers, etc.—	
With knives, 6-10 in	5–6
With knives, 10-20 in	6–8
With knives, 20-30 in	6–10
Shapers, light work	4–5
Shapers, heavy work	8
Belt sander, belt less than 6 in. wide	5
Belt sander, belt 6-10 in. wide	6
Belt sander, belt 10-14 in. wide	7
Drum sander, 24 in	5
Drum sander, 30 in	6
Drum sander, 36 in	7
Drum sander, 48 in.	8
Drum sander, over 48 in	10
Disc sander, 24 in. diam	5
Disc sander, 26-36 in. diam	6
Disc sander, 36-48 in. diam	7
Arm sander	4

direction and speed with which it is thrown off. If the dust to be removed is already in motion, as is the case with high-speed grinding wheels, the hood should be installed in the path of the particles so that a minimum air volume may be used effectively. It is always desirable to design and locate a hood so that the volume of air necessary to produce results is as small as possible.

The static suction at the throat of a hood is frequently used in practice as a measure of the effectiveness of control. This is of considerable value where exhaust systems adapted to particular operations have been standardized by practice. Tables 1 and 2 present the duct sizes usually employed for standard wood-working machinery and for grinding and buffing wheels. Static pressures, which in practice have been found necessary to control and convey various materials, are given in Table 3. It must be remembered, however, that the *suction* is merely a rough

TABLE 2. SIZE OF CONNECTIONS FOR GRINDING AND BUFFING WHEELS

Diameter of Wheels	Max. Grinding Surface Sq In.	Min. Diam. of Branch Pipes in Inches
Grinding—		
6 in. or less, not over 1 in. thick	19	3
7 in. to 9 in., inclusive, not over 1½ in. thick	43	3½
10 in. to 16 in., " " " 2 in. "	101	4
17 in. to 19 in., " " " 3 in. "	180	4½
20 in. to 24 in., " " " 4 in. "	302	5
25 in. to 30 in., " " " 5 in. "	472	6
Buffing—		
6 in. or less, not over 1 in. thick	19	3½
7 in. to 12 in., inclusive, not over 1½ in. thick	57	4
13 in. to 16 in., " " " 2 in. "	101	4½
17 in. to 20 in., " " " 3 in. "	189	5
21 in. to 27 in., " " " 4 in. "	338	6
27 in. to 33 in., " " " 5 in. "	518	7

measure of the air volume handled and consequently of the air velocity at the opening of the hood. The elimination of any dusty condition requires added information concerning the shape, size and location of the hood used with regard to the operation in question.

In some states grinding, polishing and buffing wheels are subject to regulation by codes. The static suction requirements, which range from 1½ to 5 in. water displacement in a *U*-tube, should be followed although in several instances they may appear to be excessive. Frequently, in these operations, a large part of the wheel must be exposed and the dust-laden air within the hood is thrown outward by the centrifugal action of the wheel, thus counteracting useful inward draft. This tendency may be diminished by locating the connecting duct so as to create an air flow of not less than 200 fpm about the lower rim of the wheel.

Exact determinations of hood control velocities are not available, but it is safe to assume that for most dusty operations they should not be less

TABLE 3. SUCTION PRESSURES REQUIRED AT HOODS

Type of Installation	Static Suction in Inches of Water
Exhausting from grinding and buffing wheels	1½–5
Exhausting from tumbling barrels	2
Exhausting from wood-working machinery—light duty	2
Exhausting from wood-working machinery—heavy duty	2–4
Shoe machinery exhaust	2–3
Exhausting from rubber manufacturing processes	2
Flint grinding exhaust	2
Exhausting from pottery processes	2
Lead dust and fume exhaust	2–4
Fur and felt machinery exhaust	2–3
Exhausting from textile machinery	2–3
Exhausting from elevating and crushing machinery	2
Conveying bulky and heavy materials	3–5

than 200 fpm at the point of origin. For granite dust generated by pneumatic devices, Hatch et al[2] give velocities from 150 to 200 fpm, depending on the type of hood used, as sufficient for safe control. Considering the character of the industry, air velocities of this order may be extended to similar dusty operations. The method for approximately determining these velocities in terms of the velocity at the hood opening is given below.

HOODS

No set rule can be given regarding the shape of a hood for a particular operation, but it is well to remember that its essential function is to create an adequate velocity distribution. The fact that the zone of greatest effectiveness does not extend laterally from the edges of the opening may frequently be utilized in estimating the size of hood required. Where complete enclosure of a dusty operation is contemplated, it is desirable to leave enough free space to equal the area of the connecting duct. Hoods for grinding, polishing and buffing should fit closely, but at the same time should provide an easy means for changing the wheels. It is advisable to design these hoods with a removable hopper at the base to capture the heavy dusts and articles dropped by the operator. Such provisions are of assistance in keeping the ducts clear. Air volumes used to control many dust discharges may often be reduced by effective baffling or partial enclosure of an operation. This procedure is strongly urged where dusts are directed beyond the zone of influence of the hood.

Axial Velocity Formula for Hoods

When the normal flow of air into a hood is unobstructed, the following formula may be used to determine the air velocity at any point along the axis[3]:

$$V = \frac{0.1\,Q}{x^2 + 0.1\,A} \tag{1}$$

where

V = velocity at point, feet per minute.
A = area of opening, square feet.
x = distance along axis, feet.
Q = volume of air handled, cubic feet per minute.

Velocity Contours

It is possible by use of a specially constructed Pitot tube[4] to map contours of equal velocity in any axial plane located in the field of influence. It has been found that the positions of these contours for any hood can be expressed as percentages of the velocity at the hood opening and are purely functions of the shape of the hood[5].

[2]Control of the Silicosis Hazard in the Hard Rock Industries. I. A Laboratory Study of the Design of Dust Control Systems for Use with Pneumatic Granite Cutting Tools, by Theodore Hatch, Philip Drinker and Sarah P. Choate. (*Journal of Industrial Hygiene*, Vol. XII, No. 3, March, 1930).

[3]The Control of Industrial Dust, by J. M. DallaValle (*Mechanical Engineering*, Vol. 55, No. 10, October 1933).

[4]Studies in the Design of Local Exhaust Hoods, by J. M. DallaValle and Theodore Hatch (*A.S.M.E. Transactions*, Vol. 54, 1932).

[5]Velocity Characteristics of Hoods under Suction, by J. M. DallaValle (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 387).

Further, the velocity contours are identical for similar hood shapes when the hoods are reduced to the same basis of comparison. These facts are applicable to all hood problems so that when the velocity contour distribution is known, the air flow required can be determined. Fig. 1 shows the contour distribution in two axial planes perpendicular to the sides of a rectangular hood with a side ratio of one-half. The distribution shown is identical for all openings with a similar side ratio provided the mapping is as shown in the figure. The contours, of course, are expressed as percentages of the velocity at the opening.

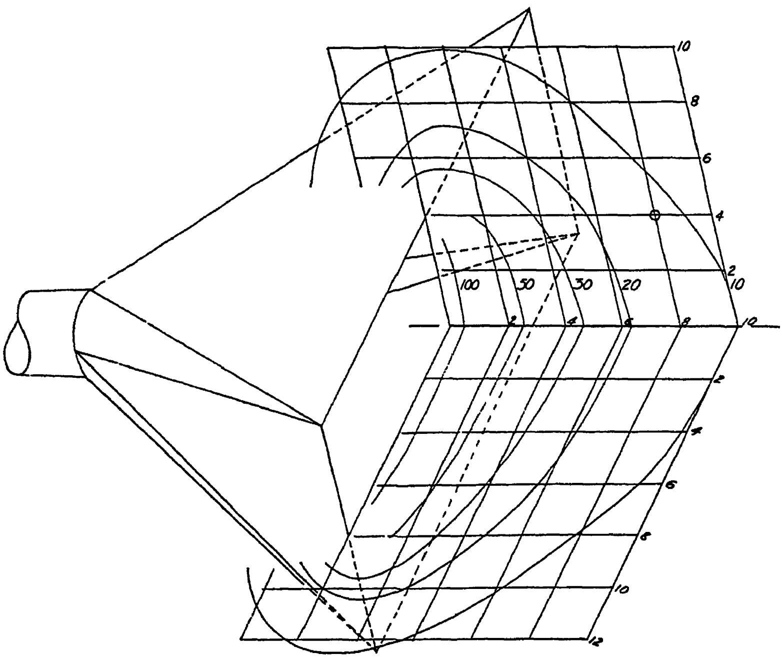

FIG. 1. VELOCITY CONTOURS FOR A RECTANGULAR OPENING WITH A SIDE RATIO OF ONE-HALF. CONTOURS ARE EXPRESSED AS PERCENTAGES OF THE VELOCITY AT THE OPENING

Air Flow from Static Readings

The volume of air flow through any hood may be determined from the following equation:

$$Q = 4005 f A \sqrt{h_t} \tag{2}$$

where

Q = volume of air flow, cubic feet per minute.

A = area of connecting duct, square feet.

h_t = static suction at throat of hood, inches of water.

= orifice or restriction coefficient which varies from 0.6 to 0.9 depending on the shape of the hood.

An average value of f is 0.71, although for a well-shaped opening a value of 0.8 may be used. The factor f is determined from the equation:

$$f = \sqrt{\frac{h_v}{h_t}} \tag{3}$$

where h_v is the velocity head in the connecting duct.

The term *static suction* is not a good measure of the effectiveness of a hood unless the area of the opening and the location of the operation with respect to the hood are known. This is clearly indicated by Equation 1 which shows that the velocity at any point along the axis varies inversely as the area of the opening and the square of the distance. However, this formula coupled with Equation 2 should serve to indicate the velocity conditions to be expected when operations are conducted external to the hood opening.

Large Open Hoods

Large hoods, such as are used for electroplating and pickling tanks, should be sub-divided so the area of the connecting duct is not less than one-fifteenth of the open area of the hood. Frequently, it will be found necessary to branch the main duct in order to obtain a uniform distribution of flow. *Canopy hoods* should extend 6 in. laterally from the tank for every 12 in. elevation, and wherever possible they should have side and rear aprons so as to prevent short circuiting of air from spaces not directly over the vats or tanks. In most cases, hoods of this type take advantage of the natural tendency of the vapors to rise, and air velocities may be kept low. Cross drafts from open doors or windows disturb the rise of the vapors and therefore provision must be made for them. The air velocities required also depend upon the character of the vapors given off, cyanide fumes, for example, requiring an air velocity of approximately 75 fpm on the surface of the tank and acid and steam vapors requiring velocities as low as 25 to 50 fpm. The total volume of air flow necessary to obtain these velocities may be approximately determined from the following simple formula:

$$Q = 1.4PDV \tag{4}$$

where

Q = total volume of air handled by hood, cubic feet per minute.
P = perimeter of the tank, feet.
D = distance between tank and hood opening, feet.
V = air velocity desired along edges and surface of tank, feet per minute.

Lateral Exhaust Systems

The lateral exhaust method, as developed for chromium plating[6], is applicable in many instances in preference to the canopy type hoods. The method makes use of drawing air and fumes laterally across the top of vats or tanks into slotted ducts at the top and extending fully along one or more sides of the tanks. The slots are 2 in. wide and for effective

[6]Health Hazards in Chromium Plating, by J. J. Bloomfield and Wm. Blum (*U. S. Public Health Report*, Vol. 43, No. 26, September 7, 1928).

ventilation a 2000 fpm exhaust air velocity at the slot face is advisable. In addition, the duct should not be required to draw the air laterally for a distance of more than 18 in. and the level of the solution should be kept 6 to 8 in. below the top of the tanks.

Flexible Exhaust Systems

The flexible exhaust tube method may be advantageously used for removing dust or fumes. Flexible tubes having one end connected to an exhaust system and a slotted hood attached to the other end may be shaped at will to fit in with industrial processes without affecting the ease of operation. Efficient dust or fume removal may be had with use of relatively small exhaust volumes. This type of system may be used on swing grinders, portable grinding wheels, soldering operations, stone cutting, rock drilling, etc.

Spray Booths

In the design of an efficient spray booth, it is essential to maintain an even distribution of air flow through the opening and about the object being sprayed. While in many instances spraying operations can be performed mechanically in wholly enclosed booths, the volatile vapors may reach injurious or explosive concentrations. At all times the concentrations of these vapors, and particularly those containing benzol, should be kept below 100 parts per million. Spray booth vapors are dangerous to the health of the worker and care should be taken to minimize exposure to them.

It is recommended in the design of spray booths that the exhaust duct be located in a horizontal position slightly below the object sprayed. Stagnant regions within the booth should be carefully avoided or should be provided with exhaust. The air volume should be sufficient to maintain a velocity of 150 to 200 fpm over the open area of the booth, and the vapors may be discharged through a suitable stack to permit dilution, but it is better practice to pass the fumes or vapors through baffle type washers or scrubbers designed for efficient spray fume removal[7].

Hoods for Chemical Laboratories

Hoods used in chemical laboratories are generally provided with sliding windows which permit positive control of the fumes and vapors evolved by the apparatus. Their design should offer easy access for the installation of chemical equipment and should be well lighted. Air velocities should exceed 50 fpm when the window is opened to its maximum height.

DUCT SYSTEM DESIGN

The duct system should be large enough to transport the fumes or material without causing serious obstruction to the air flow. It is good practice to proportion the ducts to obtain the desired velocities and suction pressures at the hoods, although in many cases only an approximation to an ideal design is possible. Many exhaust hoods, and par-

[7]For a discussion of spray booths, see Special Bulletin No. 16, Spray Painting in Pennsylvania, Department of Labor and Industry, 1926, Harrisburg, Pa.

ticularly those used in buffing and polishing, are connected by short branch pipes to the main duct which renders proportioning impractical.

Construction

The ducts leading from the hoods to the exhaust fan should be constructed of sheet metal not lighter than is shown in Table 4. The piping should be free from dents, fins and projections on which refuse might catch.

All permanent circular joints should be lap-jointed, riveted and soldered, and all longitudinal joints either grooved and locked or riveted and soldered. Circular laps should be in the direction of the flow, and piping installed out-of-doors should not have the longitudinal laps at the bottom. Every change in pipe size should be made with an eccentric taper flat on the bottom, the taper to be at least 5 in. long for each inch change in diameter. All pipes passing through roofs should be equipped with collars so arranged as to prevent water leaking into the building.

The main trunks and branch pipes should be as short and straight as possible, strongly supported, and with the dead ends capped to permit inspection and cleaning. All branch pipes should join the main at an

TABLE 4. GAGE OF SHEET METAL TO BE USED FOR VARIOUS DUCT DIAMETERS

DIAMETER OF DUCT	GAGE OF METAL
8 in. or less	24
9 to 18 in.	22
19 to 25 in.	20
26 in. or more	18

acute angle, the junction being at the side or top and never at the bottom of the main. Branch pipes should not join the main pipes at points where the material from one branch would tend to enter the branch on the opposite side of the main.

Cleanout openings having suitable covers should be placed in the main and branch pipes so that every part of the system can be easily reached in case the system clogs. Either a large cleanout door should be placed in the main suction pipe near the fan inlet, or a detachable section of pipe, held in place by lug bands, may be provided.

Elbows should be made at least two gages heavier than straight pipe of the same diameter, the better to enable them to withstand the additional wear caused by changing the direction of flow. They should preferably have a throat radius of at least one and one-half times the diameter of the pipe.

Every pipe should be kept open and unobstructed throughout its entire length, and no fixed screen should be placed in it, although the use of a trap at the junction of the hood and branch pipe is permissible, provided it is not allowed to fill up completely.

The passing of pipes through fire-walls should be avoided wherever possible, and sweep-up connections should be so arranged that foreign material cannot be easily introduced into them.

TABLE 5. AIR SPEEDS IN DUCTS NECESSARY TO CONVEY VARIOUS MATERIALS

MATERIAL	AIR VELOCITIES (FPM)
Grain dust	2000
Wood chips and shavings	3000
Sawdust	2000
Jute dust	2000
Rubber dust	2000
Lint	1500
Metal dust (grindings)	2200
Lead dusts	5000
Brass turnings (fine)	4000
Fine coal	4000

At the point of entrance of a branch pipe with the main duct, there should be an increase in the latter equal to their sum. Some state codes specify that the combined area be increased by 25 per cent. While this is not always necessary and is frequently done at the expense of a reduced air velocity, it is none the less advisable where future expansion of the exhaust system is contemplated.

Air Velocities in Ducts

When the static suction has been fixed for a given hood, the air velocity in the duct may be determined from Equation 2. Air velocities for conveying a material should be moderate. Table 5 gives the velocities generally employed for conveying various substances. Equations 5 and 5a may be used as tests to determine the conveying efficiency of a system[8]. Velocities determined from these formulae should be increased by at least 25 per cent since they represent the minimum at which a stated size and density of material can be transported.

For vertical ducts:

$$V = 13{,}300 \frac{s}{s+1} d^{0.57} \tag{5}$$

For horizontal ducts:

$$V = 6000 \frac{s}{s+1} d^{0.40} \tag{5a}$$

where

V = air velocity in duct, feet per minute.
s = specific gravity of particles.
d = average diameter of largest particles conveyed, inches.

Example 1. Granular material, the largest size of which is approximately 0.37 in. in diameter, with a specific gravity of 1.40 is to be conveyed in a vertical pipe, the velocity of the air in which is 4100 fpm; find whether the material can be transported at this velocity.

Substitute data in Equation 5a and multiply by 1.25:

$$V = 1.25 \times 13{,}300 \times \frac{1.4}{2.4} \times 0.37^{0.57}$$

Antilog ($0.57 \times \log 0.37$) = 0.568; the required velocity is, therefore, 5500 fpm.

[8]Determining Minimum Air Velocities for Exhaust Systems, by J. M. DallaValle (A S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, September, 1932, p. 639).

TABLE 6. LOSS THROUGH 90-DEG ELBOWS

ELBOW CENTER LINE RADIUS IN PER CENT OF PIPE DIAMETER	LOSS IN PER CENT OF VELOCITY HEAD
50	75
100	26
150	17
200 to 300	14

Hence, the duct velocity must be increased either by speeding up the fan or decreasing the diameter of the duct, or both.

Duct Resistance

The resistance to flow in any galvanized duct riveted and soldered at the joints may be obtained from Fig. 2, Chapter 32. The pressure drop through elbows depends upon the radius of the bend. For elbows whose centerline radii vary from 50 to 300 per cent of pipe diameter, the loss may be estimated from Table 6. It is sometimes convenient to express the resistance of an elbow in terms of an equivalent length of duct of the same diameter. Thus with a throat radius equal to the pipe diameter the resistance is equivalent to a section of straight pipe approximately 10 diameters long, while with a throat diameter radius 1½ times the diameter, the resistance is about the same as that of seven diameters of straight pipe.

COLLECTORS

The most common method of separating the dust and other materials from the air is to pass the mixture through a centrifugal or *cyclone* collector. In this type of collector the mixture of the air and material is introduced on a tangent, near the cylindrical top of the collector, and the whirling motion sets up a centrifugal action causing the comparatively heavy materials suspended in the air to be thrown against the side of the separator, from which position they spiral down to the tail piece, while the air escapes through the stack at the center of the collector.

The diameter of the cyclone should be at least 3½ times the diameter of the fan discharge duct. When two or more separate ducts enter a cyclone, gates should be provided to prevent any back draft through a system which may not be operating. Cyclones working in conjunction with two or more fans should be designed to operate efficiently at two-thirds capacity rating. The following formula is useful in computing the loss through a cyclone when the velocity of the air in the fan discharge duct is known:

$$h_c = 0.13 \left(\frac{V}{1000}\right)^2 \tag{6}$$

where

h_c = the pressure drop through the cyclone, inches of water.
V = the air velocity in the fan discharge duct, feet per minute.

If a cyclone is used to collect light dusts such as buffing wheel dusts,

feathers and lint, the exhaust vent should be large enough to permit an air velocity of 200 to 500 fpm. This will, of course, require a cyclone of larger dimensions than given for the foregoing general case.

When a high collection efficiency is desired, or the material is very fine, multicyclones may be used. These are merely small cyclones arranged in parallel which utilize the principle of high centrifugal velocity to attain separation. The capacities and characteristics of this type of separator should be obtained from the manufacturers.

Cloth Filters

Filters are used when the material collected by an exhaust system is valuable or cannot be separated efficiently from the air with an ordinary cyclone. They are also employed when it is desirable to recirculate the air drawn from a room by the exhaust system, which otherwise might entail considerable loss in heat. Bag filters which are properly housed may be operated under suction. *Bag houses* used in the manufacture of zinc oxide and other chemical products are operated on the positive side of the fan.

Wool, cotton and asbestos cloths are commonly used as filtering mediums. When woolen cloths are employed, the filtering capacities vary from ½ to 10 cfm per square foot of filtering surface, depending on the character of the material collected. The rates for cotton and asbestos cloths are lower. The type of filter cloth and the rates of filtration depend, of course, on the material to be collected and the fan capacity. The time increase of resistance varies with the amount of material permitted to build up on the surface of the filter and can be determined only by experiment. The limits of the increase may be regulated by adjustment of the shaking or cleaning mechanism. These limits may be regulated further according to the capacity of the fan and the effective performance of the hoods and the duct system.

For additional information on dust and cinders, see Chapter 29, Air Cleaning Devices.

RESISTANCE OF SYSTEM

The maintained resistance of the exhaust system is composed of three factors: (1) loss through the hoods, (2) collector drop, and (3) friction drop in the pipes.

The loss through the hoods is usually assumed to be equal to the suction maintained at the hoods. The collector drop in inches of water is given approximately by Equation 6, but where possible the resistance of the particular collector to be used should be ascertained from the manufacturer.

Friction drop in the pipes must be computed for each section where there is a change in area or in velocity. Find the velocities in each section of pipe starting with the branch most remote from the fan. The friction drop for these sections can be determined by reference to Table 6. Total friction loss in the piping system is the friction drop in the most remote branch plus the drop in the various sections of the main, plus the drop in the discharge pipe.

EFFICIENCY OF EXHAUST SYSTEMS

The efficiency of an exhaust system depends upon its effectiveness in reducing the concentration of dusts, fumes, vapors and gases below the safe or threshold limits[9].

Too much emphasis cannot be placed on the necessity of testing exhaust systems frequently by determining the concentration of atmospheric contamination at the worker's breathing level. Commonly accepted values of threshold limits for the usual gases and vapors are given in Table 7.

SELECTION OF FANS AND MOTORS

Manufacturers generally provide special fans for the collection of various industrial wastes. These are available for the collection of coal dust, wood shavings, wool, cotton and many other substances. For

TABLE 7. THRESHOLD LIMITS OF COMMON VAPORS AND GASES[a]

SUBSTANCE	SPEC. GRAV. OF GAS OR VAPOR (AIR 1)	INFLAMMABLE LIMITS (%)	PHYSIOLOGICAL ACTION	MAXIMUM ALLOWABLE CONCENTRATION (PPM)
Chlorine	2.486	non-inflamm.	irritant	0.35
Ozone	5.5	do	do	0.80
Hydrogen chloride	1.2678	do	do	10.0
Sulphur dioxide	2.2638	do	do	10.0
Carbon monoxide	0.9671	12.5–74	asphyxiant	100.0
Hydrogen sulphide	1.190	4.3–46	do	85–130
Benzene	2.73	1.4–7.0	anesthetic	100.0
Methanol	1.1	7.5–26.5	do	100.0
Carbon tetrachloride	5.3	non-inflamm.	do	100.0

[a]The Prevention of Occupational Diseases, by R. R. Sayers and J. M. DallaValle (*Mechanical Engineering*, Vol. 57, No. 4, April, 1935).

particular features concerning special fans, consult the *Catalog Data Section* of THE GUIDE and manufacturers' data. When substances having an abrasive character are conveyed, the fan blades and housing should be protected from wear. This may be accomplished by placing a collector on the negative side of the fan or by lining the housing and blades with rubber.

If no future expansion of an exhaust system is contemplated, the fan motor should be chosen to provide the calculated air volume. Should, however, the exhaust system be required to handle more air in the future, the motor should be adequate for the maximum load anticipated. Further information regarding the choice of fans and motors is given in Chapters 30 and 36.

PROTECTION AGAINST CORROSION

The removal of gases and fumes in many chemical plants requires that metals used in the construction of the exhaust system be resistant to

[9]Criteria for Industrial Exhaust Systems, by J. J. Bloomfield (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 353).

TABLE 8. MATERIALS TO BE USED FOR THE PROTECTION OF EXHAUST SYSTEMS AGAINST CORROSION[a]

Type of Fume Conveyed	Protective Material to be Used
Chlorine	Rubber lining or chrome-nickel alloys
Hydrogen sulphide	Aluminum coated iron, aluminum, high chrome-nickel alloys
Ammonia	Iron or steel
Sulphurous gases	High chrome-nickel alloys
Hydrochloric acid	Rubber lining, chrome-nickel alloys
Nitrous gases	Nickel-chrome alloys

[a]Condensed from data given by Chilton and Huey (*Industrial and Engineering Chemistry*, Vol. 24, 1932).

chemical corrosion. A list of the materials which may be used to resist the action of certain fumes is given in Table 8. Hoods and ducts when short, may frequently be constructed of wood and be quite effective. Rubberized paints are available and may be applied as protective coatings in handling such gases and fumes as chlorine and hydrochloric acid.

Chapter 41

DRYING SYSTEMS

Drying Methods, Driers, Mechanism of Drying, Moisture, General Rules for Drying, Equipment, Humidity Chart, Combustion, Design, Estimating Methods

DRYING, in its broader sense, refers to the removal of water, or other volatile liquid from either a gaseous, liquid, or solid material. In practice, the process of direct drying gaseous material is referred to generally as dehumidifying, or condensing, and in some cases chemicals are used in the adsorption or absorption of moisture. Drying a liquid is called evaporation or distillation. The common usage of the word *drying* refers to the removal of water or other liquid, such as a solvent, by evaporation from a solid material.

When the solid to be dried contains large amounts of free water, the actual drying process is frequently preceded by the removal of part of the water by some mechanical means, such as filtration, settling, pressing or centrifuging. Removal of as much water as possible by such methods is usually advisable, as the cost of these operations, per pound of water removed, is generally much less than by evaporation.

DRYING METHODS

Drying may be accomplished in any one or combination of the following methods:

1. Radiation.
2. Conduction, or direct contact.
3. Convection.

Radiation

The source of heat for radiation may be either the sun, or heated surfaces. Sun drying is practiced where danger from rain is slight, and where sufficient time can be allowed. Where a strict adherence to a schedule is necessary, or where dusty atmosphere is present, this method is not in favor. Fruits are often dried in the sun.

Radiation from hot surfaces (heated by steam, electricity, or other means) furnishes generally, from one-third to one-half the total heat required for evaporation. Convection currents set up by these hot surfaces and the cooler materials carry the balance of the heat.

TABLE 1. DRIERS FOR EVAPORATION OF WATER

Type	Kind	Materials Handled	Means of Handling	Temp. Range Deg F	Heat Supply	Uses and Remarks
Batch or Intermittent	Compartment	Paper, Leather, Yarns, Lumber, Foodstuffs	Suspended, Truck, Tray	80 to 180	Steam Coils, Air, Electricity	When production does not warrant continuous drier
	Agitated	Chemicals too sticky for Rotary Drier	Shoveled into Drum or Pan	100 to 330	Water, Steam Jacketed, may have Vacuum on top	Where dust must be saved
	Vacuum	Chemicals, Explosives, Pharmaceuticals, Food Products	Tray, Basket, Tumbling Drum	80 to 300	Water, Steam	Cost of operation high, for expensive materials
Continuous	Tunnel	Ceramics, Chemicals, Lumber, Food Products	Truck, Tray, Belt	100 to 350	Steam Coils, Air, Electricity, Products of Combustion	For high production
	Rotary	Bulk	Cascades through	80 to 500	Air, Steam, Products of Combustion	Where material will stand rough handling and is not subject to balling up
	Drum	Liquids, Slurries	Flowed on Drum, Dry Material Scraped off	to 310	Steam, may have Vacuum on Top	Hygroscopic materials dried with vacuum, and packed immediately
	Cylinder	Paper, Textiles, Chemicals	Continuous Sheets, Endless Chain Belt	to 350	Steam inside of Drum	Where material comes in sheets or rolls, and will stand direct contact with heating surface
	Festoon	Paper, Chemicals	Continuous Sheets, Suspended on Metal Screens	to 200	Air, Steam Coils	Where one side cannot come in contact with supports until dry
	Tower or Column	Grains, Sand	Falls through by Gravity	125 to 250	Air, Steam Coils	Where headroom is available
	Spray	Solutions over 30% Solids	Sprayed into Chamber	120 to 350	Air, Products of Combustion	Drying is almost instantaneous
	Induction	Metals, for removal of traces of Water	Placed in High Frequency Field	to 400	Electricity	Where heating of metal from inside out is important

Conduction or Direct Contact Drying

This method of drying is advantageous where the material can be flowed on to the drying surface and the dried material scraped off, or where the material to be dried can be handled in a sheet, and where there is no danger of subjecting the product to the full temperature of the heating medium. The source of heat for this method may be steam, electricity, hot oil or hot water.

Convection

The circulation of heated air or other gases about the material to be dried is generally termed convection drying. The convection may be either natural or forced. With forced circulation, the temperature of the

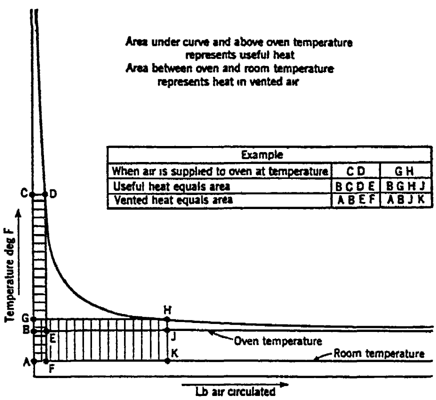

Fig. 1. Relation Between Useful and Total Heat Supplied

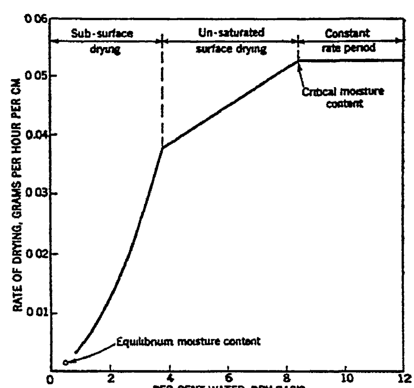

Fig. 2. Rate of Drying of Whiting Slab

drier is more uniform and the rate of drying is much higher than with natural circulation. Where humidity is used, the control is much easier, and more accurate.

The source of heat for a convection drier may be steam, electricity, hot water, oil-fired heater, gas-fired heater, or coal furnace. Where either oil, gas or coal is used, the type of heater may be direct or indirect; *i.e.*, the products of combustion may be used (direct), or the circulated air may be heated through an interchanger (indirect).

Where the direct type is used, there is naturally a higher thermal efficiency, but it can only be used where the odor, soot, or the chemical elements of the products of combustion do not affect the material being dried. When heat economy is an important consideration this method (Fig. 1) may be used, permitting a small amount of air to be circulated, if a sacrifice of accurate control of temperature and humidity can be justified.

DRIERS

The term *adiabatic drier* is applied to a drier in which all the heat is supplied by air externally heated. The temperature of the air in the

drier decreases as a transfer of heat to the material being dried takes place. Where part or all of the heat is supplied by steam coils or other means, within the drier itself, the drier is known as a *constant temperature drier.* Driers using little air for heating medium with a high temperature drop are difficult to hold at uniform temperatures; the more air used, the easier it is to secure accurate control of temperature and humidity. Driers may be classified as shown in Table 1.

MECHANISM OF DRYING

The modern theory of drying may be summed up as follows: Assuming uniform velocity and distribution of air at a constant temperature and humidity over the surface to be dried, the drying cycle will be divided into two distinct stages:

1. Constant rate period.
2. Falling rate period.

The *constant rate period* occurs while the material being dried is still very wet, and continues as long as the water in the material comes to the surface so rapidly that the surface remains thoroughly wet, and evaporation proceeds at a constant rate, precisely as from a free water surface. The material assumes a temperature corresponding to the wet-bulb temperature of the surrounding air, or slightly higher, due to radiation and conduction from dry surfaces adjoining the material. The constant rate period continues until a time when the moisture no longer comes to the surface as fast as it is evaporated. This point is called the *critical moisture content.*

As the drying proceeds, a period of *uniform falling rate* is entered. During this period, the surface of the material is gradually drying out, and the rate of drying falls as the remaining wet surface decreases in area. This period is also known as unsaturated surface drying.

As drying continues, the surface is completely dry and the water from the interior evaporates and comes through the surface as vapor. As the plane of water recedes, the diffusion of the vapor becomes more difficult and hence the period is known as *varying falling rate period*, or sub-surface drying.

As drying progresses another point called *equilibrium moisture content* is reached, where the vapor pressure of the moisture in the air and the vapor pressure of the moisture in the material are equal, and drying ceases. The drying of a slab of whiting is shown in Fig. 2 and illustrates the principles pointed out above. The factors affecting the variations of drying rates during the above periods are pointed out in Table 2.

Omissions in the Cycle

Many solids, such as lumber, are so dry at the beginning of the drying operation that the constant rate period of free surface evaporation does not occur. Frequently the surface of the material is dry enough so that no surface drying can take place, in which case only the final stage of sub-surface drying is involved. In other instances, the critical moisture content of a wet solid is sufficiently low that sub-surface drying starts almost immediately after the conclusion of the constant rate period. Thus the

intermediate state of unsaturated surface drying does not occur and the drying is of the sub-surface type during practically the whole of the falling rate period. With other kinds of material, particularly thin sheets, such as newsprint paper, sub-surface drying may occur at such a low moisture content that it is not encountered in commercial work, the

TABLE 2. FACTORS INFLUENCING DRYING

Factor	Drying Period	
	Constant Rate, Unsaturated Surface	Sub-Surface
Temperature	Increase in temperature increases drying rate	Increase in temperature increases drying rate, because with decreased viscosity, diffusion increases
Humidity	Drying rate increases as humidity is decreased	No effect until equilibrium content is reached; drying then ceases
Air Velocity	Drying rate varies approximately as the 0.6 power of the velocity	No effect
Air direction	Drying rate increases, the more nearly the air blows perpendicular to surface; for dead air film becomes thinner	No effect
Thickness of Material	Drying rate is not affected by the thickness	Drying rate varies inversely as the square of the thickness

falling rate period being confined solely in practice, to unsaturated surface drying.

MOISTURE

Moisture in the solid may be in either of two forms:

1. Capillary or free.
2. Hygroscopic or chemically combined.

Free moisture is contained in the capillary spaces between the particles or fibers of the materials. The loss of this moisture changes only the weight of the material. *Chemically combined* or hygroscopic moisture is intimately associated with the physical nature of the material and its removal changes both the physical characteristics as well as the chemical properties. The amount of hygroscopic moisture a material can contain is limited. This limit is called the *fiber saturation point.* When material is dried below this point, care must be exercised to avoid physical changes in the material, such as shrinkage, hardening, etc. All hygroscopic materials have definite equilibrium moisture contents dependent on temperature and humidity. Materials are frequently dried to a lower moisture content than those of equilibrium conditions in use, and allowed to regain the necessary moisture after leaving the drier to equalize the moisture in the material. Fig. 3[1] shows the equilibrium moisture content of wood.

[1] *U. S. Department of Agriculture Bulletin*, No. 1136.

GENERAL RULES FOR DRYING

Temperature

The highest temperature possible should be used because of faster drying and smaller requirements for ventilation. The amount of moisture that can be carried by a pound of air increases rapidly with rise in temperature as shown in the humidity chart of Fig. 4. Too high a temperature may cause spoilage of materials; many materials calcine or change their chemical properties if heated too hot; gypsum and glauber salts lose some of the chemically combined water, fall apart, and change their chemical properties. Too high or rapid rise in temperatures in drying lumber or ceramics may create a liquid vapor tension within the material so high that the cells explode, causing permanent injury to the fiber. If too high a temperature is used on some chemicals, they begin to react

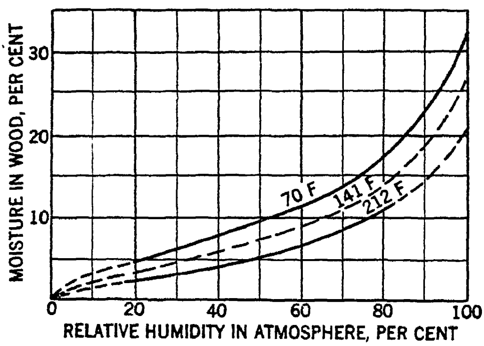

FIG. 3. RELATION OF EQUILIBRIUM MOISTURE CONTENT IN WOOD TO THE RELATIVE HUMIDITY OF SURROUNDING AIR

exothermally; a temperature rise and chemical action from within will burn the materials, *e.g.*, bakelite products, gunpowder, etc. During the constant rate period of drying, the material heats only to the wet-bulb temperature of the surrounding air, consequently high temperatures will not injure the material in this stage.

Humidity

Moisture in the drying air may be very important. Many materials tend to case-harden, dry on the outside, forming a skin which retards the moisture flow from the inside to the surface, or stops it completely, and so increases the drying time very much or causes a change of the physical properties of the material. It is often necessary to add humidity to the air in the initial stage of drying. Lumber case-hardens, cracks, and warps if the outside is dried too fast. Ceramics crack if not heated through before drying commences. Elastic materials warp while others crack if not evenly dried. Many paints case-harden if not dried under high humidity.

On the other hand, in the case of those materials whose physical or chemical properties require that they be dried at relatively low temperatures high humidity tends to retard drying in the first stage and may even stop it altogether in the final stage. Where drying temperatures

below 120 to 140 F are used the drying rate may be highly dependent on atmospheric humidity conditions. In such instances it is often desirable to dehumidify the air entering the drier during periods of high atmospheric humidity; where a high degree of uniformity is required it is often possible to secure complete independence of atmospheric conditions by recirculating the air in a closed system which includes a suitable dehumidifier. For this purpose absorptive dehumidifying systems have the advantage of accomplishing the desired reduction of humidity without appreciably elevating or lowering the dry-bulb temperature of the air; for this reason after-cooling is not required, and reheating is reduced to a minimum. Complete descriptions of such dehumidifying systems are given in Chapter 24 on Cooling and Dehumidification Methods.

Air Circulation

As noted under Mechanism of Drying, air velocity is more important in the first two stages of drying than in the last, and for this reason zone drying in continuous driers is frequently considered. It permits accurate regulation of temperature, humidity, and velocity in the different zones. High velocity results in more rapid drying, more even distribution of temperature and consequently more even drying in the first period. Too high a velocity may be detrimental because of excessive power needed for creating it, or because the material may blow away if it is light and fluffy. In the drying of paints, varnishes, and enamels, high velocity or improper distribution of the air even with the use of filters, may cause dust already in the drier to be blown against the material, ruining the finish. Table 3 presents data on drying of various materials.

EQUIPMENT FOR DRYING

Equipment for drying may be divided into the following classes:

1. Heat and humidity supply.
2. Methods of handling.
3. Ovens.

The heat and humidity supply for low temperature work up to 250 F is often steam; steam coils either in the oven or outside, heat the air used for drying. Circulation of heated oil is used to a limited extent, but the danger of leaks is serious, for if the oil is hotter than the flash point, a fire may start if the oil is released to the atmosphere. In many cases where steam is not available, direct or indirect-fired heaters are used with gas or oil as fuel. Indirect heaters should be carefully selected from a standpoint of long life and efficiency. The heat exchange surface should be adequate in area and easily accessible for cleaning and removal. For extremely high temperatures, alloy surface may be used. With direct-fired equipment care must be used in the selection of burners and sufficient combustion space allowed to insure complete combustion of fuel. Humidity can be obtained in driers by the use of steam spray, humidifiers, or recirculation.

Methods of *handling of material* have been indicated in Table 1.

For low temperature work up to 200 F ovens and driers are commonly built of two thicknesses of insulating board (fireproof preferred), with air space between. As the temperature increases materials better able to

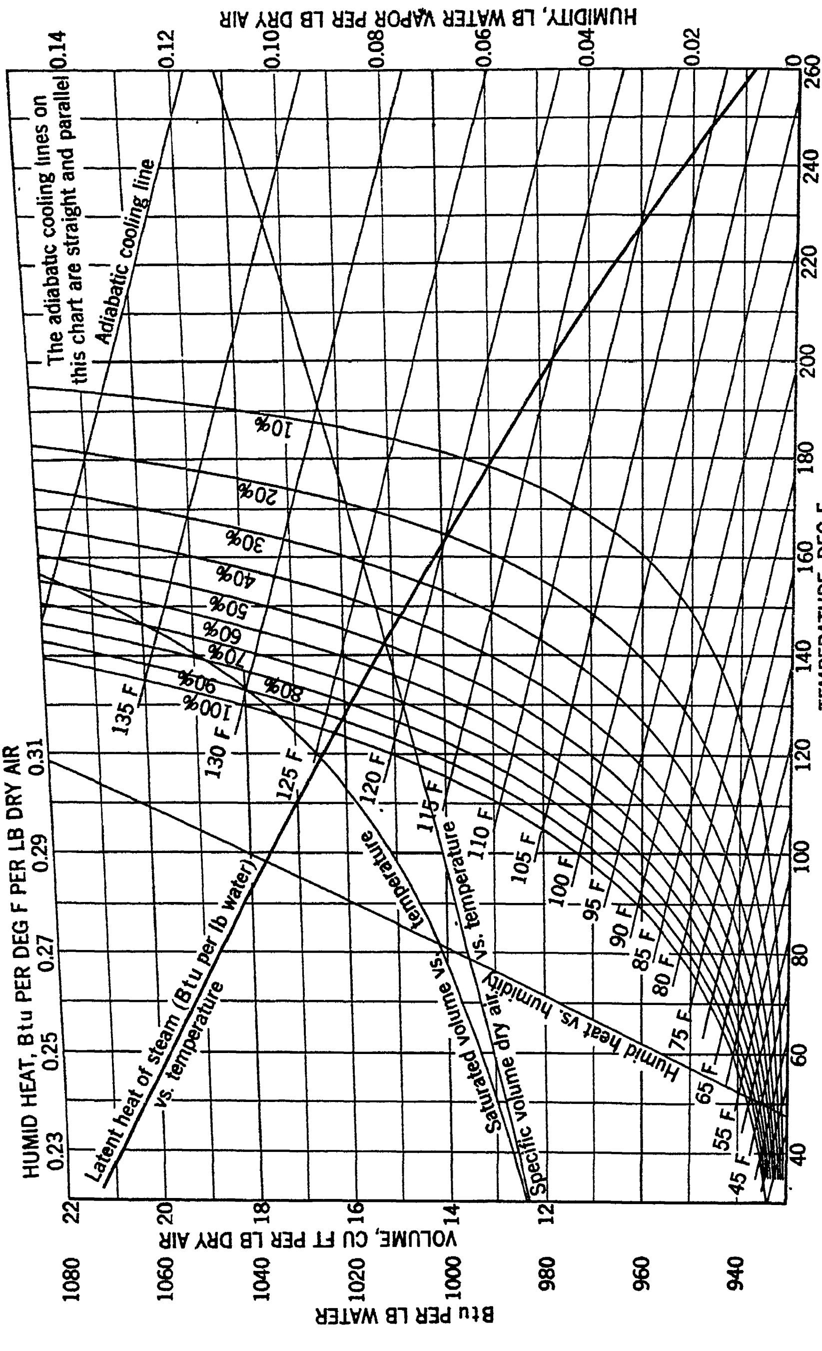

Fig. 4. Humidity Chart

withstand the heat must be used. Metal lined ovens are easy to keep clean, and many high temperature driers up to 1000 F are made of metal panels with insulation between. Care should be taken to avoid *through metal* (metal extending through the oven from inside to out). Batch type ovens are entirely closed while in use and control of air leakage is easily taken care of. In the continuous drier where the ends are open, heat and air leakage becomes important. Warm air leaking out of the ends of ovens means a heat loss, and often the temperature and humidity outside the oven becomes unbearable. For this reason, inclined or bottom entry ovens are used, as the warm air leakage can be more easily controlled. See Figs. 5 and 6.

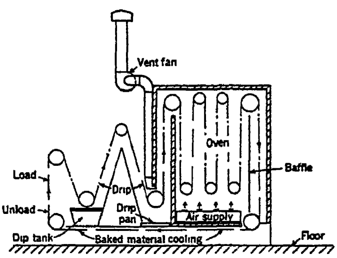

FIG. 5. SMALL PART MULTIPLE PASS OVEN

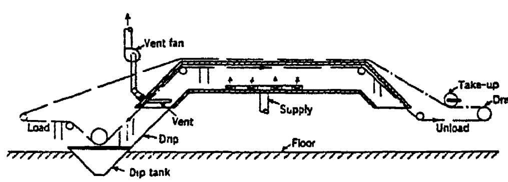

FIG. 6. INCLINED END ENAMELING OVEN

HUMIDITY CHART FOR DRYING WORK

In drying problems the chemical engineer uses different psychrometric values than those used by the heating, ventilating and air conditioning engineer. The humidity chart illustrated in Fig. 4 is based upon values determined from the following explanations:

Humidity (H) is the number of pounds of water vapor carried by one pound of dry air.

Percentage Humidity ($\% H$) is the number of pounds of water vapor carried by one pound of dry air at a definite temperature, divided by the number of pounds of vapor that one pound of dry air would carry if it were completely saturated at the same temperature.

Per Cent Relative Humidity (Φ) is the ratio of weight of water vapor contained in any given volume of air, to the weight of water vapor present in the same volume of saturated air, all values referring to the same temperature.

To convert from one relation to the other,

$$\% H = \frac{29.92 - p_s}{29.92 - p} \times \Phi \qquad (1)$$

where

p_s = vapor pressure of water, inches mercury; at dry-bulb temperature, degrees Fahrenheit.

$p = \Phi\, p_s$.

COMBUSTION

Where products of combustion are used directly in the oven, a knowledge of their formation and heat values is important. The properties of

TABLE 3. DRYING TIME AND CONDITIONS FOR REPRESENTATIVE MATERIALS[a]

Material	Temperature Deg F	Per Cent Relative Humidity	Drying Time
Apples	140–180		6 Hrs
Armatures Varnish	200		2.5 Hrs
Banana Food ¼ in. Thick	140		4–6 Hrs
Barrels	300		15 Min
Beans	140		18 Hrs
Bedding	150–190		
Blankets	120		40 Min
Brake Lining	325		12 Min
Brick continuous	350 to 90		24 Hrs
Briquets	1100		108 Min
Cabbage Raw	150		4.5 Hrs
Candied Peel	165		2 Hrs
Casein	180		5 Hrs
Cereals	110–150		
Ceramics before firing	150	70 to 20	24 Hrs
Chicle	95–100		
Coco-fiber mats	170–210		10 Hrs
Cocoanut	150–200		4–6 Hrs
Coffee	160–180		24 Hrs
Conduit (Enamel)	400 Max		2 Hrs
Cores, Oil sand for molding ½—1 in. thick	300		30 Min
Black sand with goulic binder about 0.6 of time 3 in. thick	480		2.5 Hrs
Black sand with goulic binder about 0.6 of time 8 in. thick	480		4.5 Hrs
Black sand with goulic binder about 0.6 of time 16 in. thick	700		10 Hrs
Cores, Crank case (in continuous ovens)	525–600		2–3 Hrs
Cores, Radiator (in continuous ovens)	275–450		1.5 Hrs
Cornstalk Board	150		2 Hrs
Cotton Linters	180		
Enamels synthetic			
Finish coat on autos	225		2 Hrs + Air Dry
Ice boxes all metal (white)	290–425		1 Hr
Ice boxes wood inside (white)	225		3 Hrs
Enamel not synthetic			
Fence posts green	200		1 Hr
Golf balls (white)	90–95	40–50	18–36 Hrs
Small parts (auto) black	450		1 Hr
Steel furniture	225–300		30–350 Min
License plates	250		1.5 Hrs
Feathers	150–180		
Films, Photographic	85–110		20–30 Min
Fruits and Vegetables	140		2–6 Hrs
Furs	110		
Gelatin	110		
Glue bone, thin sheets on wire trays	70–90		6–9 Days
Glue skin	70–90		2 Days
Glue size on furniture	130		4 Hrs
Gut	150		
Gypsum board ⅜ in. thick Start Wet	350		60 Min
Gypsum board ⅜ in. thick Finish	275		
Gypsum block	350–190		8–16 Hrs
Hair felt	180–200		
Hair goods	150–190		1 Hr
Hanks on poles	120		2 Hrs
Hats felt	140–180		
Hides thin leather	90		2–4 Hrs
Hides heavy	70–90		4–6 Days

[a]See references at end of chapter.

TABLE 3. DRYING TIME AND CONDITIONS FOR REPRESENTATIVE MATERIALS[a]—Con.

Material	Temperature Deg F	Per Cent Relative Humidity	Drying Time
Hops	120–180		
Ink printing	70–300		
Japan beds	300		1.5–2 Hrs
Japan cash register	300–450		1.5 Hrs
Japan metal shelving	200		30 Min
Knitted fabrics	140–180		
Leather mulling	78–95	85	
Leather thick sole	90	70	4 Days
Leather uppers	80		2–3 Days
Linoleum varnish	110–145	10–30	6–10 Hrs
Lithographing on tin color work	250–270		18–25 Min
Lithographing on tin Japan	350		
Lumber green hardwood	100–180		3–180 Days
Lumber green soft wood	160–220		2–14 Days
Macaroni	90–110		24–48 Hrs
Matches	140–180		
Matrix	350		15 Min
Milk and other liquid foods spray dried	135–300		Instantaneous
Millboard sheets	95		10 Hrs
Moulds green sand C.I. flasks (one surface only exposed) 8 in. thick	600		6 Hrs
Moulds green sand C.I. flasks (one surface only exposed) 13 in. thick	700		13 Hrs
Motors, field coils	180		6 Hrs
Motors, stators	250		6.5 Hrs
Noodles	90–95		
Nuts	75–140		24 Hrs
Oil cloth	150		
Paint, wood wheels	150	35	8–24 Hrs
Paint, on sheet metal	135–140	22–30	2.5 Hrs
Paper, machine dried	180		
Paper, air dried	90–200		
Paper wall, ground coat	140		3 Min
Paper wall, varnished	140–160	45	15 Min
Paper cardboard, spirit varnish	150		1–2 Min
Peaches	135		26 Hrs
Pears	140		24 Hrs
Peas	150		6 Hrs
Potatoes sliced	85		4 Hrs
Potatoes steamed	170		6.5 Hrs
Prunes	140		
Rags	180		
Ramie fiber	140		10 Hrs
Rice	150		
Rock wool insulation	300		8 Hrs
Rubber	85–90		6–12 Hrs
Rubber reclaimed	140–200		1–2 Hrs
Rugs	190		4–8 Hrs
Salt	350		Rotary Drier
Sand loose 1 in. deep	300		10–15 Min
Sausage casings	110		5 Hrs
Shade cloth	240		1–2 Hrs
Shirts	120		20 Min
Soap	100–125		12–72 Hrs
Starch	180–200		1–4 Hrs
Stock feed mixed	180–220		20–30 Min
Storage battery plates	100–110	90 for	24 Hrs
	250	Low for	6 Hrs
Sugar	150–200		20–30 Min

[a]See references at end of chapter.

TABLE 3. DRYING TIME AND CONDITIONS FOR REPRESENTATIVE MATERIALS[a]—Con.

MATERIAL	TEMPERATURE DEG F	PER CENT RELATIVE HUMIDITY	DRYING TIME
Tanin and other chemicals (spray dried)	250–300		Instantaneous
Terra Cotta (air drying in conditioned room)	150–200		12–96 Hrs
Tobacco leaves	85–130		12 Hrs
Tobacco stems	180–200		12 Hrs
Varnish refrigerator boxes	110	35	5–7 Hrs
Varnish steering wheels	110–140	25–35	Overnight
Veneer ¼ in. 3-ply	120–130	35–40	6–8 Hrs + 2 Hrs acclimation
13/16 in. 5-ply	120–130	35–40	16–18 Hrs + 4 Hrs acclimation
1¼ in. 5-ply	120–130	35–40	20–24 Hrs + 4 Hrs acclimation
Vitreous Enamel sheets before firing	170		
Wallboard pasted plywood	300		15–20 Min
Wallboard fiber insulating, roller type drier	300–385		2½–3 Hrs
Wallboard fiber insulating, truck type drier	300–385		24–48 Hrs
Walnuts	100		24 Hrs
Wheat, corn, oats, rice, barley	180		
Wire cloth Japan	200		20 Min
Wool	105		

[a]See references at end of chapter.

the common constituents of fuel are shown in Table 4. The heating values of oils are shown in Fig. 7. The sensible heat in Btu contained in the products of combustion of an average fuel oil and various gases is given in Fig. 8. The problem of securing complete combustion in a heater is important, in order to secure efficiency and the absence of soot formation, but unlike the ordinary power or heating boiler, excess air need not be maintained at a minimum in most cases. Excess air is generally admitted either in the heater or before the products go into the drier.

DESIGN

In all drying problems, data regarding temperatures, time, and humidity must be obtained by experiment or previous experience. Experiments are best performed at the temperatures, humidities, and velocities to be actually used in the full sized drier, and with full size samples.

The following nomenclature and explanation of terms will be used in the discussion of drying calculations:

H = humidity of air, pounds of water vapor per pound of dry air.

G = pounds of dry air supplied to the drier per unit of time.

S = pounds of stock dried per unit of time in a continuous drier.

S' = pounds of stock charged per batch to a discontinuous drier.

Θ = time.

Q = total heat supplied to the drier.

t = air temperature.
t^{I} = stock temperature.
t^{II} = average stock temperature over short time interval, in a batch drier.
t_w = wet-bulb temperature.
s^{I} = specific heat of the stock.
B = total radiation and conduction losses per unit time.
w = pounds of water per pound of dry stock.
r = heat of evaporation of water.
s = humid heat of air, *i.e.*, heat necessary to raise 1 lb of dry air + H lb of steam 1 F.

Subscript (1) designates conditions at the point where the material in question (air or stock) enters and (2) where it leaves the drier.

Air driers may be divided into two classes, those in which *all moisture* evaporated from the stock *leaves the drier as vapor* in the effluent air, and those in which *part or all* of the moisture *is condensed* from the air *in the drying equipment itself*. In any continuously operating drier of the first type the relation between moisture content of the stock and quantity of air required for the drying operation is given by the equation:

$$G\,(H_2 - H_1) = S(w_1 - w_2) \tag{2}$$

TABLE 4. GAS COMBUSTION CONSTANTS[a]

Gas	Chemical Formula	Molecular Weight	Cu Ft per Lb	Heat of Combustion		Lbs per Lb of Combustible					
				Btu per Lb		Required for Combustion			Flue Products		
				Gross	Net	O_2	N_2	Air	CO_2	H_2O	N_2
Carbon	C	12.000	......	14,140	14,140	2.667	8.873	11.540	3.667	......	8.873
Hydrogen	H_2	2.015	187.723	61,100	51,643	7 939	26.414	34.353	......	8.939	26.414
Oxygen	O_2	32.000	11.819	......	......	......	......	......	......	......	......
Nitrogen	N_2	28.016	13.443	......	......	......	......	......	......	......	......
Carbon Monoxide	CO	28.000	13.506	4,369	4,369	0.571	1.900	2.471	1.571	......	1.900
Carbon Dioxide	CO_2	44.000	8.548	......	......	......	......	......	......	......	......
Methane	CH_4	16.031	23.565	23,912	21,533	3.992	13.282	17.274	2.745	2.248	13 282
Ethane	C_2H_6	30.046	12.455	22,215	20,312	3.728	12.404	16.132	2.929	1.799	12.404
Propane	C_3H_8	44.062	8 365	21,564	19,834	3.631	12.081	15.712	2.996	1.635	12.081
Sulphur Dioxide	SO_2	64.060	5.770	......	......	......	......	......	......	......	......
Water Vapor	H_2O	18.015	21.017	......	......	......	......	......	......	......	......
Air	......	28.900	13.063	......	......	......	......	......	......	......	......

[a]All gas volumes corrected to 60 F and 30 in. mercury barometric pressure dry.

In discontinuous driers, *e.g.*, compartment driers, the drying operation is given by the equation:

$$G\,(H_2 - H_1) = S' \frac{dw}{d\Theta} \tag{2a}$$

In the continuous drier, the heat consumption per unit time is:

$$\frac{Q}{\Theta} = Gs_1(t_2 - t_1) + G(r_2 + t_2 - t'_2)\,(H_2 - H_1) + S(t'_2 - t'_1)\,(s' + w_1) + B \tag{3}$$

Equation 3 assumes continuity of operation. For charge or batch operations, the total time of the drying cycle may be broken up into a number of periods, sufficiently short so that over each period average

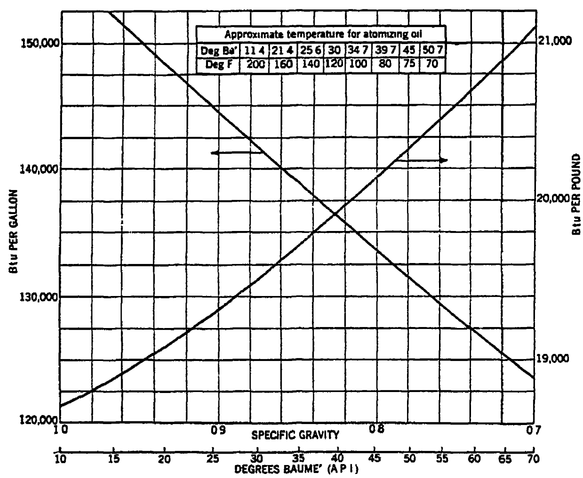

FIG. 7. HEATING VALUES OF FUEL OIL, BTU GROSS

values of t, t' and H may be employed provided the third term of the right hand member of the equation is modified to read:

$$S'\,(t''_1 - t''_1)\,(s' - w_1)$$

and in the second term t'_2 be replaced by

$$\frac{t'_1 + t''_2}{2}$$

Theoretically these periods should be very short and the equation integrated. Practically the error introduced by using a small number of long periods and employing average values of the variables over each, rarely introduces serious error. The evaluation of Equation 2a may be approximated in a similar manner.

The first term of the right hand member of Equation 3 represents heat lost as sensible heat in the effluent air. In many drying operations this becomes excessive. Each pound of air supplied should remove the maximum amount of moisture. This is best accomplished by bringing the air

into contact with the stock with sufficient intimacy so that the air leaving the drier is saturated, or nearly so. Counter-current as against parallel flow of air and stock gives rise to optimum operating conditions, resulting in a minimum quantity of air required (G), and a corresponding minimum loss, as sensible heat, in the exit air. Similarly, continuous operation is superior to intermittent operation.

Despite the fact that the sensible heat loss increases with the rise in temperature of the air, the percentage of heat lost from this source decreases, provided the increase in moisture carrying capacity of the air,

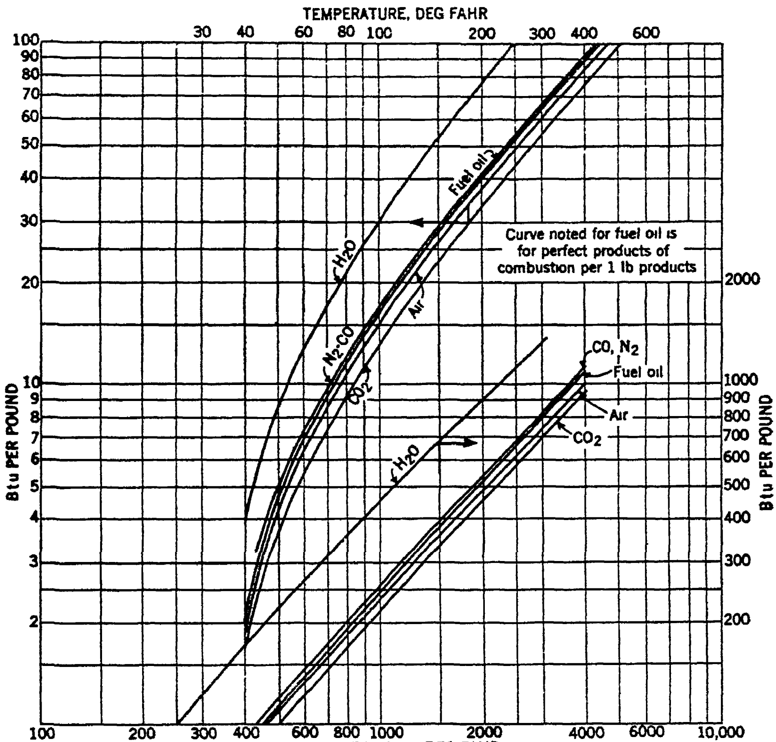

FIG. 8. HEAT CONTENT OF GASES ABOVE 32 F IN BTU PER POUND

due to high temperature, is actually utilized. To secure maximum thermal efficiency in drying, a high drying temperature and high saturation of the outlet air is imperative.

Ventilation Phase

The technique of attack of the *ventilation phase* of a drying problem is best made clear by an illustration. Assume that a material containing 40 per cent moisture is to be dried until this quantity of moisture is reduced to 5 per cent by weight. The material will stand an air temperature of 150 F and it is possible to provide sufficiently good contact between the material and the drying air so that the effluent air can be

brought up to 50 per cent humidity at 150 F. The drier is to use room air, the temperature and humidity of which may be assumed to average 70 F and 50 per cent. A counter-current drier will be employed and the air in this drier will be kept at a substantially constant temperature of 150 F by heaters thermostatically controlled. The stock enters at 70 F, rises quickly to the wet-bulb temperature of the air, with which it is in

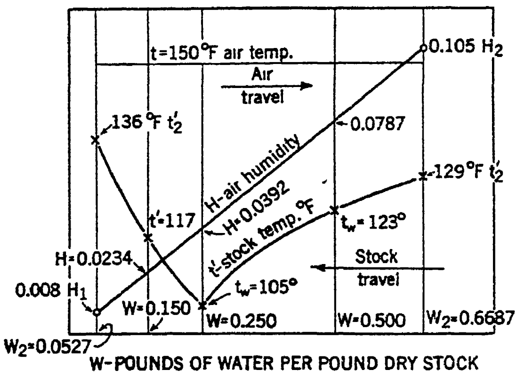

FIG. 9. TEMPERATURE HUMIDITY RELATIONS IN A DRIER

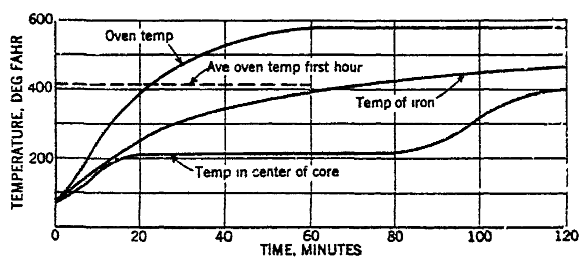

FIG. 10. CORE DRYING TIME TEMPERATURE RELATIONS

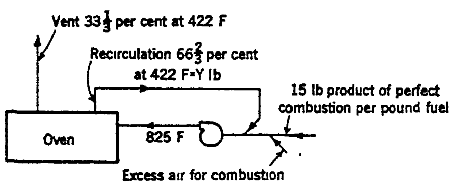

FIG. 11. CORE DRYING DIAGRAM OF COMBUSTION PRODUCTS AND AIR

contact, and is found experimentally to maintain wet-bulb temperature until the moisture content has fallen to 20 per cent. From this point its temperature rises progressively as it dries. In this range the difference in temperature between stock and air, divided by the wet-bulb depression, may be assumed proportional to the moisture content.

The moisture content of the entering stock, in the units here employed, is:

$$w_1 = \frac{40 \text{ per cent water}}{60 \text{ per cent dry stock}} = 0.6667: \; w_2 = \frac{5 \text{ per cent water}}{95 \text{ per cent dry stock}} = 0.0527$$

$w_1 - w_2 = \Delta w = 0.614$ lb water evaporated per pound of dry stock. Since the air leaving the drier is 50 per cent saturated at 150 F from Fig. 4, $H_2 = 0.105$. Similarly, $H_1 = 0.008$, corresponding to 50 per cent humidity at 70 F. Consequently $H_2 - H_1 = \Delta H = 0.097$ lb water evaporated per pound dry air.

Inspection of Equation 2 shows that (H) is linear in w. Hence, one can construct on Fig. 9, the line marked (H) being drawn connecting the initial and final points just computed.

Since the air leaving the drier has a temperature of 150 F and a humidity of 0.105, Fig. 4 shows that its wet-bulb temperature is 129 F. This is plotted at the right hand side of Fig. 9. Since the stock maintains a wet-bulb temperature down to 20 per cent moisture, where $w = 0.25$, the corresponding humidity can be computed by the use of Equation 2 or by reading directly from the diagram, the value being 0.0392. Fig. 4 shows that the corresponding wet-bulb temperature is 105 F. Any intermediate point on the wet-bulb temperature curve can be calculated similarly. The points for $w = 0.5$ are shown in Fig. 9.

Below the point, $w = 0.25$, the temperature of the stock begins to rise appreciably above the wet-bulb temperature. Its temperature at any given point in this range, for example at $w = 0.15$, may be computed as follows: At this point, $H = 0.0234$ (from Equation 2) and from Fig. 4, $t_w = 95$ F. Hence the wet-bulb depression, $t - t_w = 150 - 95 = 55$ F. The assumption made regarding the relation between stock temperature and moisture content in this range may be formulated:

$$\frac{\Delta t^1}{t - t_w} = \frac{w}{0.25}$$

At the point $w = 0.15$, $\Delta t^1 = 33$ F, $t^1 = 117$ F. The temperature of the stock leaving the drier, similarly computed, is 136 F.

Fig. 9 thus computed gives in graphical form the information as to the temperature humidity relationships in the drier. The air requirements can be computed by Equation 2. Thus, per 100 lb of dry stock, it is necessary to supply 633 lb of dry air. Furthermore, since from Fig. 4 it is seen that the volume of 50 per cent saturated air at 70 F, is 13.55 cu ft per pound; 8580 cu ft of room air must be supplied per 100 lb dry stock. Similarly, since the volume of 50 per cent saturated air at 150 F is 18.0 cu ft per pound, the volume of hot wet air discharged from the drier is 11,400 cu ft per 100 lb of dry stock. Finally, the heat necessary to supply to the drier, as a whole, or to any section of it, may be computed from Equation 3.

High Temperature Drier

In the design of a high temperature drier unit a method of approach to the necessary calculations involved is outlined as follows:

Example 1. Cores 4 and 5 in. thick are to be dried by heating to a temperature at 400 F. An intermittent type box oven is to be used, size 12 x 14 x 10 ft with 856 sq ft surface having an average heat transfer of 0.3 Btu per square foot per degree per hour. Drying time as determined by test is 2 hr (Fig. 10). Cores weighing 6 tons, and 15-ton steel plates, trucks etc. are delivered to the drier at 70 F. The oven is heated by an external heater; the products of combustion and 66⅔ per cent recirculated air will be delivered to the oven at 825 F. Fuel oil of 19,980 Btu gross and 18,830 Btu per pound net heating value, weighing 6.75 lb per gallon and having 15 lb product per pound fuel

for perfect combustion. Cores consist of 91 per cent sand, 3 per cent oil binder, and 6 per cent water.

Solution. Heat required per ton of cores:

	Lb Material × Temp. Rise × Sp. Ht.	≐	Btu
Sand	0.91 × 2,000 × (400 − 70) × 0.2	=	120,120
Binder	0.03 × 2,000 × (400 − 70) × 0.4	=	7,920
Water heating	0.06 × 2,000 × (212 − 70) × 1.0	=	17,040
Water evaporation	0.06 × 2,000 × 970 (Fig. 4)	=	116,520
Water superheating (approx. 50 per cent reaches 575 F)	= 0.5 × 0.06 × 2,000 × (575 − 212) × 0.45	=	9,800
Total Heat			271,400 Btu

Heating Load First Hour

	Heated to			Btu
Sand	212 F	$\frac{142}{330}$ × 120,120	=	51,688
Binder[a]	212 F	$\frac{142}{330}$ × 7,920	=	3,408
Water	212 F		=	17,040
Evaporation	66.7%	0.667 × 116,520	=	77,680
Superheat	66.7%	0.667 × 9,800	=	6,530
Total Per Ton				156,346
For 6 ton		6 × 156,346	=	938,076
Steel plates	390 F	320 × 30,000 × 0.12	=	1.152,000
Radiation[b]	422 F Avg	352 × 856 × 0.30	=	90,394
Total				2,180,470

Heating Load Second Hour

Sand	400 F	$\frac{188}{330}$ × 120,120	=	68,432
Binder[a]	400 F	$\frac{188}{330}$ × 7,920	=	4,512
Water				
Evaporation	33.3%	0.333 × 116,520	=	38.840
Superheat	33.3%	0.333 × 9,800	=	3.270
Total Per Ton				115,054
For 6 ton		6 × 115,054	=	690,324
Steel plates	460	70 × 30,000 × 0.12	=	252,000
Radiation[b]	575	505 × 856 × 0.30	=	129,684
Total				1,072,008

[a]Binder oxidizes and liberates heat, which is neglected in this calculation.

[b]Average value of coefficient is less than 0.3 because oven is not up to 575 F. This is neglected. 422 F is arrived at by taking area under curve as compared to area under 575 F ordinate.

Heat in 1 lb fuel oil	=	18,830 Btu
Heater Loss (10 per cent)	= 1883	
Duct Loss (5 per cent)	= 942	2,825 Btu
		16,005 Btu available to heat oven.

Heat content of gases in 1 lb fuel oil at 825 F is 205 Btu (Fig. 8)

15 lb × 205	=	3,075 Btu sensible heat in products of perfect combustion.
		12,930 Btu to heat air X and Y (Fig. 11).

$$Y\,(S_{825} - S_{422}) + X\,(S_{825} - S_{70}) = 12{,}930 \tag{4}$$

$Y = 2\,(X + 15)$ for 66.7 per cent recirculation

where

S = heat content of air at temperature noted taken from Fig. 8.

(Recirculation and exhaust contains water vapor, products of combustion, and a greater portion of air. Heat capacities of all vary so little that they have all been assumed to be air).

$S_{825} - S_{422} = 190 - 91 = 99$

$S_{825} - S_{70} = 190 - 8.6 = 181.4$

Substituting values of Y, H, etc. in Equation 4,

$(2\,X + 30)\,99 + 181.4\,X = 12{,}930$

$X = 26.3$ lb excess air.

$Y = 82.6$ lb recirculating air.

Total = 26.3 + 82.6 + 15 = 123.9 lb air and products of combustion circulated per pound fuel burned.

Heat in air exhausted from oven at 422 F per pound fuel burned = $0.333 \times 123.9 \times (S_{422} - S_{70}) = 41.3\,(91 - 8.6) = 3{,}400$ Btu.

Btu available for heating material = 16,005 − 3,400 = 12,605 Btu per pound fuel.

Fuel used in first hour = 2,180,470 ÷ 12,605 = 173 lb = 25.6 gal.

During the second hour the heater capacity will be much greater than required. If an automatic oven temperature control operates on the oil supply, the delivery temperature of the air entering the oven and the quantity of oil burned will decrease, the air supply being constant.

Heat in air exhausted = $41.3\,(S_{575} - S_{70}) = 41.3\,(127 - 8.6) = 4{,}880$ Btu per pound fuel.

Heat available for heating material = 16,005 − 4,880 = 11,125 Btu.

Fuel used in second hour = 1,072,008 ÷ 11,125 = 96.5 lb oil = 14.3 gal.

Total oil used per load = 25.6 + 14.3 = 39.9 gal.

ESTIMATING METHODS

Values based on practical experience are available for rough estimating of drying problems. The temperature will drop approximately 8.5 F per grain of water evaporated per cubic foot of air (measured at 70 F) or approximately 0.62 F per pound of air at any temperature. Air will drop 55 F per cubic foot for each Btu extracted. Generally air will absorb from 2 grains to 5 grains per cubic foot of air in one passage through an air drier, depending on the temperature and the degree of contact with the material. The amount of steam required to evaporate a pound of water will vary from 1.5 lb to a more usual figure of from 2.5 to 3 lb of steam per pound of water evaporated.

REFERENCES

Commercial Drying Apparatus, by L. P. Dwyer (A.S.H.V.E. TRANSACTIONS, Vol. 22, 1916, p. 479).

Artificial Drying with Special Reference to the Use of Gas, by G. C. Shadwell (A.S.H. V.E. TRANSACTIONS, Vol. 23, 1917, p. 231).

Drying by Evaporation, by F. R. Still (A.S.H.V.E. TRANSACTIONS, Vol. 23, 1917, p. 255).

High Temperature Drying, by Burt S. Harrison (A.S.H.V.E. TRANSACTIONS, Vol. 24, 1918, p. 7).

The Temperature of Evaporation, by W. H. Carrier (A.S.H.V.E. TRANSACTIONS, Vol. 24, 1918, p. 25).

Commercial Dehydration, by J. E. Whitley (A.S.H.V.E. TRANSACTIONS, Vol. 26, 1920, p. 551).

Drying as an Air Conditioning Problem, by A. W. Lissauer (A.S.H.V.E. TRANSACTIONS, Vol. 27, 1921, p. 251).

A Chronological Survey of Drying and Driers, by J. E. Bolling (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1921, p. 715).

Calculations for Drying Design, by Grosvenor (*Transactions, American Institute Chemical Engineering*, 1908, p. 184).

The Rate of Drying Solid Materials, by J. Lewis (*Industrial Engineering Chemistry*, 1921, p. 427).

Adiabatic Drying of Hygroscopic Solids, by A. M. McCready, and W. L. McCale (*Transactions, American Institute Chemical Engineers*, 1933).

Lignite Drier, by Lavine & Sutherland (*Chemical and Metallurgical Engineering*, July, 1929).

Enameling Oven Economy, by B. S. Harrison (*Fuels & Furnaces*, February, 1931).

Powdered Yeast Prepared by Spray Drying, by A. W. Farrell (*Food Industry*, December, 1931).

Factors that Influence Drier Performance, by A. Weisselberg (*Chemical and Metallurgical Engineering*, August, 1932).

Principles of Drying Lumber and Humidity Diagram, by H. D. Tiemann (*Forest Service Bulletin* 104, 1912).

Symposium on Drying. Articles by W. K. Lewis, W. H. Carrier, A. E. Stacey, Jr. and R. S. Fleming, R. G. Metz, G. B. Ridley, C. O. Lavett, D. J. Van Marle (*Journal Industrial Engineering Chemistry*, May, 1921).

The Drying of Solids, by T. K. Sherwood (*Bulletin Massachusetts Institute Technology* Nos. 237, 247 and 258).

Air Conditioning and Engineering, American Blower Co.

Combustion (*American Gas Association*, 3rd edition, 1932).

Die Trockentechnik, by M. Hirsch (Julius Springer, Berlin, 1932).

Drying (*Kent's, Mechanical Engineers Handbook*, 10th edition, 1923; 11th edition, 1936).

Drying, by W. H. Carrier (*Marks', Mechanical Engineers Handbook*, 3rd edition, 1930).

Drying, by Perry (*Chemical Engineers Handbook*, 1934).

Drying by Means of Air and Steam, by E. Hausbrand (D. Van Nostrand & Co., 1901).

Drying in Industrial Plants, by J. O. Ross.

Elements of Chemical Engineering, by Badger and McCabe (McGraw Hill Co., 1931).

Fan Engineering, Buffalo Forge Co.

Fuels and Their Combustion, by Haslam and Russell (McGraw Hill Co., 1926).

Heat Transmission, by W. H. McAdams (McGraw Hill Co., 1933).

Modern Drying Machinery, by H. B. Grenshaw, London, 1926.

Principles of Chemical Engineering, by Walker, Lewis, McAdams (Chapters on Evaporation, Humidity and Drying, 2nd edition, McGraw Hill Co.).

The Kiln Drying of Lumber, by A. Koehler and R. Thelen, New York, 1926.

The Kiln Drying of Lumber, by H. D. Tiemann (Lippincott, 1920).

Chapter 42

NATURAL VENTILATION

Wind Forces, Stack Effect, Openings, Windows, Doors, Skylights, Roof Ventilators, Stacks, Principles of Control, General Rules, Measurements, Dairy Barn Ventilation, Garage Ventilation

VENTILATION by natural forces, finds application in industrial plants, public buildings, schools, dwellings, garages, and in farm buildings.

The natural forces available for moving air into, through and out of buildings are: (*a*) wind forces, and (*b*) the difference in temperature between the air inside and outside a building. The air movement may be caused by either of these forces acting alone or by a combination of the two, depending upon atmospheric conditions, building design and location. The ventilating results obtained will vary, from time to time, due to variation in the velocity and direction of the wind and the heat generated in the building. The arrangement, location, and control of the ventilating openings should be such that the two forces act cooperatively rather than in opposition.

WIND FORCES

In considering the use of natural wind forces for producing ventilation, account must be taken of: (1) average wind velocity, (2) prevailing wind direction, (3) seasonal and daily variations in velocity and direction, and (4) local wind interference by nearby buildings, hills or other obstructions of similar nature.

Values are given in Table 1, Chapter 7 for the average summer wind velocities and the prevailing wind directions in various localities throughout the United States, while Table 2, Chapter 6, lists similar values for the winter. In almost all localities the summer wind velocities are lower than those in the winter, and in about two-thirds of the localities the prevailing direction is different during the summer and winter. While the tables give no average velocities below 5 mph, there will be times when the velocity is lower, even in localities where the seasonal average is considerably above 5 mph. There are relatively few places where the velocity falls below one half of the average for many hours per month. Consequently, if the natural ventilating system is designed for wind velocities of one-half of the average seasonal velocity, it should prove satisfactory in almost every case.

Equation 1 may be used for calculating the quantity of air forced through ventilation openings by the wind, or for determining the proper size of such openings to produce given results:

$$Q = EAV \tag{1}$$

where

Q = air flow, cubic feet per minute.
A = free area of inlet openings, square feet.
V = wind velocity, feet per minute, = miles per hour $\times$ 88.
E = effectiveness of openings. (E should be taken at 0.50 to 0.60 for perpendicular winds and 0.25 to 0.35 for diagonal winds[1].)

The accuracy of the results obtained by the use of Equation 1 depends upon the placing of the openings, as the formula assumes that ventilating openings have a flow coefficient slightly greater than that of a square-edged orifice. If the openings are not advantageously placed with respect to the wind, the flow per unit area of the openings will be less and, if unusually well placed, the flow will be slightly more than that given by the formula. Inlets should be placed to face directly into the prevailing wind, while outlets should be placed in one of the following five places:

1. On the side of the building directly opposite the direction of the prevailing wind.
2. On the roof in the low pressure area caused by the jump of the wind (see Fig. 1).
3. On the sides adjacent to the windward face where low pressure areas occur.
4. In a monitor on the side opposite from the wind.
5. In roof ventilators or stacks.

TEMPERATURE DIFFERENCE FORCES[2]

The stack effect produced within a building when the outdoor temperature is lower is due to the difference in weight of the warm column of air within the building and the cooler air outside. The flow due to stack effect is proportional to the square root of the draft head, or approximately:

$$Q = 9.4\,A\sqrt{H\,(t - t_o)} \tag{2}$$

where

Q = air flow, cubic feet per minute.
A = free area of inlets or outlets (assumed equal), square feet.
H = height from inlets to outlets, feet.
t = *average* temperature of indoor air in height H, degrees Fahrenheit.
t_o = temperature of outdoor air, degrees Fahrenheit.
9.4 = constant of proportionality, including a value of 65 per cent for effectiveness of openings. This should be reduced to 50 per cent (constant = 7.2) if conditions are not favorable.

HEAT REMOVAL

In problems of heat removal, knowing the amount of heat to be removed and having selected a desirable temperature difference, the amount

[1]Predetermining Airation of Industrial Buildings, by W. C. Randall and E. W. Conover (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 605).

[2]Neutral Zone in Ventilation, by J. E. Emswiler (A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926, p. 59).

of air to be passed through the building per minute to maintain this temperature difference can be determined by means of Equation 3.

$$H = 0.0175\ Q\ (t - t_o) \tag{3}$$

where

H = heat removed, Btu per minute.

Q = air flow, cubic feet per minute.

$t - t_o$ = inside-outside temperature difference, degrees Fahrenheit.

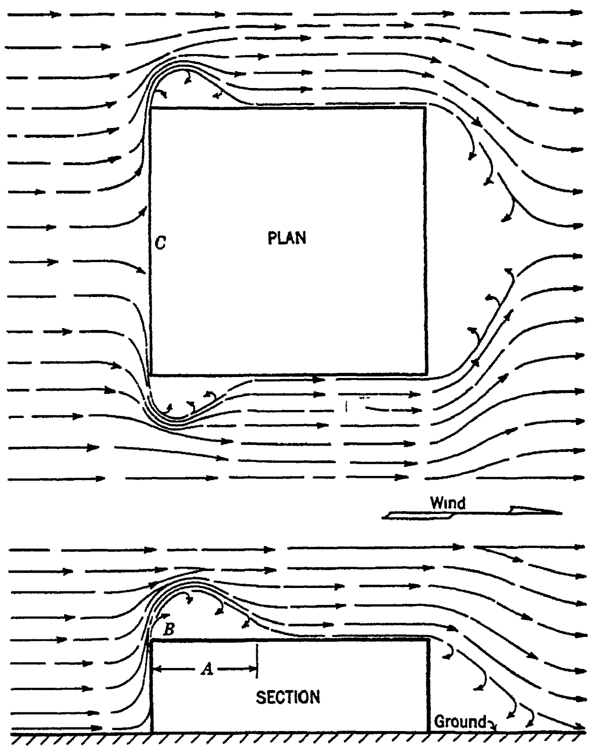

FIG. 1. THE JUMP OF WIND FROM WINDWARD FACE OF BUILDING. (A—LENGTH OF SUCTION AREA; B—POINT OF MAXIMUM INTENSITY OF SUCTION; C—POINT OF MAXIMUM PRESSURE)

EFFECT OF UNEQUAL OPENINGS

The largest flow per unit area of openings is obtained when inlets and outlets are equal, and the equations given previously are based on this condition. Increasing outlets over inlets, or vice-versa, will increase the air flow, but not in proportion to the added area. When solving problems having an unequal distribution of openings, use the smaller area, either inlet or outlet, in the equations and add the increase as determined from Fig. 2.

COMBINED FORCES OF WIND AND TEMPERATURE

Equations for determining the air flow due to temperature difference and wind have already been given. It must be remembered that when both forces are acting together, even without interference, the resulting air flow is not equal to the sum of the two estimated quantities. The flow through any opening is proportional to the square root of the sum of the forces acting on that opening.

When the two forces are about equal in intensity and the ventilating openings are operated so as to coordinate them, the total air flow through the building is about 10 per cent greater than that produced by either force acting independently under conditions ideal to that force. This

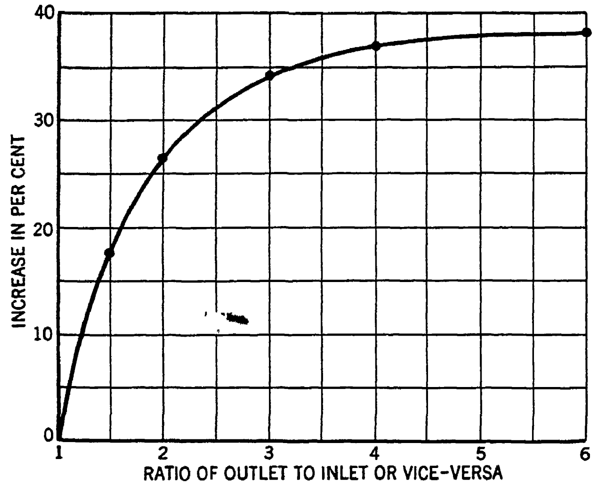

FIG. 2. INCREASE IN FLOW CAUSED BY EXCESS OF ONE OPENING OVER ANOTHER

percentage decreases rapidly as one force increases over the other and the larger force will predominate.

The wind velocity and direction, the outdoor temperature, or the indoor distribution, cannot be predicted with certainty, and refinement in calculations is not justified; consequently, a simplified method can be used. This may be done by using the equations and calculating the flows produced by each force separately under conditions of openings best suited for coordination of the forces. Then by determining as a percentage, the ratio of the flow produced by temperature difference to the sum of the two flows, the actual flow due to the combined forces can be approximated from Fig. 3.

Example 1. Assume a drop forge shop, 200 ft long, 100 ft wide, and 30 ft high. The cubical content is 600,000 cu ft, and the height of the air outlet over that of the inlet is 30 ft. Oil fuel of 18,000 Btu per pound is used in this shop at the rate of 15 gal per hour (7.75 lb per gal). Desired summer temperature difference is 10 F and the prevailing wind is 8 mph perpendicular to the long dimension. What is the necessary area for the inlets and outlets, and what is the rate of air flow through the building?

Solution for Temperature Difference Only. The heat $H = \dfrac{15 \times 7.75 \times 18{,}000}{60} =$ 34, 875 Btu per minute.

By Equation 3, the air flow required to remove this heat with an average temperature difference of 10 F is:

$$Q = \frac{H}{0.0175\ (t - t_o)} = \frac{34{,}875}{0.0175 \times 10} = 199{,}286 \text{ cfm.}$$

This is equal to about 20 air changes per hour. From Equation 2 the inlet (or outlet) opening area should be:

$$A = \frac{Q}{9.4 \sqrt{H\ (t - t_o)}} = \frac{199{,}286}{9.4 \sqrt{30 \times 10}} = 1224 \text{ sq ft.}$$

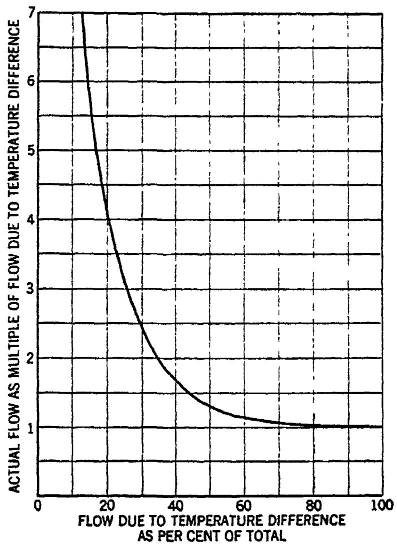

Fig. 3. Determination of Flow Caused by Combined Forces of Wind and Temperature Difference

The flow per square foot of inlet or outlet would be 199,286 ÷ 1224 = 163 cfm with all windows open.

Solution for Wind Only. With 1,224 sq ft of inlet openings distributed around the sidewalls, there would be about 410 sq ft in each long side and 202 sq ft in each end. The outlet area will be equally distributed on the two sides of the monitor, or 612 sq ft on each side. With the wind perpendicular to the long side, there will be 410 sq ft of opening in its path for inflow and 612 in the lee side of the monitor for outflow with the windward side closed. The air flow, as calculated by Equation 1, will be:

$$Q = 0.60 \times 410 \times 704 = 173{,}200 \text{ cfm.}$$

This gives 17.3 air changes per hour, which should be more than ample when there is no heat to be removed.

Solution for Combined Forces. Since the windward side of the monitor is closed when the wind is blowing, the flow due to temperature difference must be calculated for this condition, using Fig. 2. This chart shows that when inlets are twice the size of the outlets, in this case 1,224 sq ft in the sidewalls and 612 sq ft in the monitor, the flow will be increased 26.5 per cent over that produced by equal openings. Using the smaller

opening and the flow per square foot obtained previously, the calculated amount for this condition will be:

$$612 \times 163 \times 1.265 = 126{,}200 \text{ cfm.}$$

Adding the two computed flows:

Temperature Difference	= 126,200 =	42 per cent.
Wind	= 173,200 =	58 per cent.
Total	299,400 =	100 per cent.

From Fig. 3, it is determined that when the flow, due to temperature difference, is 42 per cent of the total, the actual flow, due to the combined forces, will be about 1.6 times that calculated for temperature difference alone, or 201,920 cfm.

The original flow, due to temperature difference alone, was 199,286 cfm with all openings in use. The effect of the wind is to increase this to 201,920 cfm even though half of the outlets are closed.

A factor of judgment is necessary in the location of the openings in a building, especially those in the roof, where heat, smoke and fumes are to be removed. Usually windward monitor openings should be closed, but if the wind is low enough for the temperature head to overcome it, all windows may be opened.

TYPES OF OPENINGS

Types of openings may be classified as: (1) windows, doors, monitor openings and skylights, (2) roof ventilators, (3) stacks connecting to registers, and (4) specially designed inlet or outlet openings.

Windows, Doors and Skylights

Windows have the advantage of transmitting light, as well as providing ventilating area when open. Their movable parts are arranged to open in various ways; they may open by sliding either vertically or horizontally, by tilting on horizontal pivots at or near the center, or by swinging on pivots at the top, bottom or side. Regardless of their design, the air flow per square foot of opening will be the same under the same conditions. The type of pivoting should receive consideration from the standpoint of weather protection, and certain types may be advantageous in controlling the distribution of incoming air. Deflectors are sometimes used for the same purpose, and these devices should be considered a part of the ventilation system.

Roof Ventilators

The function of a roof ventilator is to provide a storm and weather proof air outlet. These are actuated by the same forces of wind and temperature head, which create flow through other types of openings. The capacity of a ventilator depends upon four things: (1) its location on the roof, (2) the resistance it and the duct work offers to air flow, (3) the height of draft, and (4) the efficiency of the ventilator in utilizing the kinetic energy of the wind for inducing flow by centrifugal or ejector action.

For maximum flow induction, a ventilator should be located on that part of the roof where it will receive the full wind without interference. If ventilators are installed within the suction region created by the wind

passing over the building, or in a light court, or on a low building between two high buildings, their performance will be the same there as for any other type of opening of the same area. Their normal ejector action, if any, will be of no value in such a location.

The base of the ventilator should be of a taper-cone design to produce the effect of a bell-mouth nozzle whose coefficient of flow is considerably higher than that of a square-entrance orifice. If a grille is provided at the base, additional resistance is introduced, and it should be increased in size accordingly.

Air inlet openings located at lower levels in the building should be at least equal to, and preferably larger than the combined throat areas of all roof ventilators. The air discharged by a roof ventilator depends on wind velocity and temperature difference, and, in general, their performance will be the same as any monitor opening located in the same place, but due to the four capacity factors already mentioned, no simple formula can be devised for expressing ventilator capacity.

Roof ventilators may be classified as stationary, pivoting or oscillating, and rotating. Generally, these have a round throat, but the continuous-ridge ventilator, or so-called heat valve, would fall in the stationary classification. When selecting roof ventilators, some attention should be given to ruggedness of construction, storm proofing features, dampers and damper operating mechanisms, possibility of noise, original cost and maintenance.

Natural ventilation units may be used to supplement power-driven supply fans, and under favorable weather conditions it may be possible to stop the power-driven units.

Controls

Gravity ventilators may have dampers controlled by (1) hand, (2) thermostat, and (3) wind velocity, in combination with a fan. The thermostat station may be located anywhere in the building, or it may be located within the ventilator itself. The purpose of wind velocity control is to obtain a definite volume of exhaust regardless of the natural forces, the fan motor being energized when the natural exhaust capacity falls below a certain minimum, and again shut off when the wind velocity rises to the point where this minimum volume can be supplied by natural forces.

Stacks

Stacks or vertical flues are really chimneys and utilize both the inductive effect of the wind and the force of temperature difference. Like the roof ventilator, the stack outlet should be located so that the wind may act upon it from any direction. With little or no wind, chimney effect depends on temperature difference to produce a removal of air from the rooms where the inlet openings are located.

GENERAL RULES

A few of the important requirements in addition to those already outlined are:

1. Inlet openings in the building should be well distributed, and should be located on

the windward side near the bottom, while outlet openings are located on the leeward side near the top. Outside air will then be supplied to the zone to be ventilated.

2. Inlet openings should not be obstructed by buildings, trees, sign boards, etc., outside nor by partitions inside.

3. Greatest flow per square foot of total opening is obtained by using inlet and outlet openings of nearly equal areas.

4. In the design of window ventilated buildings, where the direction of the wind is quite constant and dependable, the orientation of the building together with amount and grouping of ventilation openings can be readily arranged to take full advantage of the force of the wind. Where the wind's direction is quite variable, the openings should be arranged in sidewalls and monitors so that, as far as possible, there will be approximately equal areas on all sides. Thus, no matter what the wind's direction, there will always be some openings directly exposed to the pressure force and others to a suction force, and effective movement through the building will be assured.

5. Direct short circuits between openings on two sides at a high level may clear the air at that level without producing any appreciable ventilation at the level of occupancy.

6. In order that temperature difference may produce a motive force, there must be vertical distance between openings. That is, if there are a number of openings available in a building, but all are at the same level, there will be no motive head produced by temperature difference, no matter how great that difference might be.

7. In order that the force of temperature difference may operate to maximum advantage, the vertical distance between inlet and outlet openings should be as great as possible. Openings in the vicinity of the neutral zone are less effective for ventilation.

8. In the use of monitors, windows on the windward side should usually be kept closed, since, if they are open, the inflow tendency of the wind counteracts the outflow tendency of temperature difference. Openings on the leeward side of the monitor result in cooperation of wind and temperature difference.

9. In an industrial building where furnaces that give off heat and fumes are to be installed, it is better to locate them in the end of the building exposed to the prevailing wind. The strong suction effect of the wind at the roof near the windward end will then cooperate with temperature difference, to provide for the most active and satisfactory removal of the heat and gas laden air.

10. In case it is impossible to locate furnaces in the windward end, that part of the building in which they are to be located should be built higher than the rest, so that the wind, in splashing therefrom will create a suction. The additional height also increases the effect of temperature difference to cooperate with the wind.

11. The intensity of suction or the vacuum produced by the jump of the wind is greatest just back of the building face. The area of suction does not vary with the wind velocity, but the flow due to suction is directly proportional to wind velocity.

12. Openings much larger than the calculated areas are sometimes desirable, especially when changes in occupancy are possible, or to provide for extremely hot days. In the former case, free openings should be located at the level of occupancy for psychological reasons.

13. In single story industrial buildings, particularly those covering large areas, natural ventilation must be accomplished by taking air in and out of the roof openings. Openings in the pressure zones can be used for inflow and openings in the suction zone, or openings in zones of less pressure, can be used for outflow. The ventilation is accomplished by the manipulation of openings to get air flow through the zones to be ventilated.

DAIRY BARN VENTILATION [3]

A successful barn ventilating system is one which continuously supplies the proper amount of air required by the stock, with proper distribution and without drafts, and one which removes the excessive heat, moisture,

[3]Dairy Barn Ventilation, by F. L. Fairbanks (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 181). Cow Barn Ventilation, by Alfred J Offner (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 149). For additional information on this subject refer to *Technical Bulletin, U. S. Department of Agriculture* (1930), by M. A. R. Kelley.

and odors, and maintains the air at a proper temperature, relative humidity, and degree of cleanliness.

Barn temperatures below freezing and above 80 F affect milk production. Milk producing stock should be kept in a barn temperature between 45 and 50 F. Dry stock, at reduced feeding, may be kept in a barn 5 to 10 deg higher. Calf barns are generally kept at 60 F, while hospital and maternity barns usually have a temperature of 60 F or somewhat higher.

The heat produced by a cow of an average weight of 1000 lb may be taken as 3000 Btu per hour. The average rate of moisture production by a cow giving 20 lb of milk per day is 15 lb of water per day, or 4375 grains per hour. To set a standard of permissible relative humidity for cow barns is difficult. For 45 F an average relative humidity of 80 per cent is satisfactory, with 85 per cent as a limit.

Where the barn volume is within the limit that can be heated by the stabled animals, the air supply need not be heated. The air should be supplied through or near the ceiling. It is better to have the exhaust openings near the floor as larger volumes of warm air are then held in the barn and there is better temperature control with less likelihood of sudden change in barn temperature.

If a cow weighs 1000 lb and produces 3000 Btu of heat per hour, and if a barn for the cow has 600 cu ft of air space with 130 sq ft of building exposure, one cow will require 2600 to 3550 cu ft per hour of ventilation, depending on the temperature zone in which the barn is located. The permissible heat losses through the structure, based on one cow and depending on the temperature zone, vary between 0.043 and 0.066 Btu per hour per cubic foot of barn space, and 0.197 to 0.305 Btu per hour per square foot of barn exposure.

GARAGE VENTILATION

On account of the hazards resulting from carbon monoxide and other physiologically harmful or combustible gases or vapors in garages, the importance of proper ventilation of these buildings cannot be overemphasized. During the warm months of the year, garages are usually ventilated adequately because the doors and windows are kept open. As cold weather sets in, more and more of the ventilation openings are closed and consequently on extremely cold days the carbon monoxide concentration runs high.

Many garages can be satisfactorily ventilated by natural means particularly during the mild weather when doors and windows can be kept open. However, the A.S.H.V.E. Code for Heating and Ventilating Garages, adopted in 1929 and revised in 1935, states that natural ventilation may be employed for the ventilation of storage sections where it is practical to maintain open windows or other openings at all times. The code specifies that such openings shall be distributed as uniformly as possible in at least two outside walls, and that the total area of such openings shall be equivalent to at least 5 per cent of the floor area. The code further states that where it is impractical to operate such a system of natural ventilation, a mechanical system shall be used which shall provide for either the supply of 1 cu ft of air per minute from out-of-doors

for each square foot of floor area, or for removing the same amount and discharging it to the outside as a means of flushing the garage.[4]

Research

Research on garage ventilation undertaken by the A.S.H.V.E. Committee on Research at Washington University, St. Louis, Mo., and at the University of Kansas, Lawrence, Kans., in cooperation with the A.S.H. V.E. Research Laboratory, and at the A.S.H.V.E. Research Laboratory has resulted in authoritative papers on the subject.

Some of the conclusions from work at the Laboratory are listed in the following statements:

1. Upward ventilation results in a lower concentration of carbon monoxide at the breathing line and a lower temperature above the breathing line than does downward ventilation, for the same rate of carbon monoxide production, air change and the same temperature at the 30-in. level.
2. A lower rate of air change and a smaller heating load are required with upward than with downward ventilation.
3. In the average case upward ventilation results in a lower concentration of carbon monoxide in the occupied portion of a garage than is had with complete mixing of the exhaust gases and the air supplied. However, the variations in concentration from point to point, together with the possible failure of the advantages of upward ventilation to accrue, suggest the basing of garage ventilation on complete mixing and an air change sufficient to dilute the exhaust gases to the allowable concentration of carbon monoxide.
4. The rate of carbon monoxide production by an idling car is shown to vary from 25 to 50 cu ft per hour, with an average rate of 35 cu ft per hour.
5. An air change of 350,000 cu ft per hour per idling car is required to keep the carbon monoxide concentration down to one part in 10,000 parts of air.

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[4]Code for Heating and Ventilating Garages (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 355), (A.S. H.V.E. Reprint, January, 1935).

Airation Study of Garages by W. C. Randall and L. W. Leonhard (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 233).

A.S.H.V.E. RESEARCH REPORT No. 874—Carbon Monoxide Concentration in Garages, by A.S. Langsdorf and R. R. Tucker (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 511).

A S.H.V.E. RESEARCH REPORT No. 935—Carbon Monoxide Distribution in Relation to the Ventilation of an Underground Ramp Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 439).

A.S.H.V.E. RESEARCH REPORT No. 934—Carbon Monoxide Distribution in Relation to the Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 424).

A.S.H.V.E. RESEARCH REPORT No. 967—Carbon Monoxide Distribution in Relation to the Heating and Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 395).

Carbon Monoxide Surveys of Two Garages, by A. H. Sluss, E. K. Campbell and Louis M. Farber (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 263).

Chapter 43

PIPE AND DUCT HEAT LOSSES

Heat Losses from Bare and Insulated Pipes, Low Temperature Pipe Insulation, Insulation of Pipes to Prevent Freezing, Economical Thickness of Pipe Insulation, Underground Pipe Insulation, Heat Losses from Ducts

THE heat transfer through uninsulated pipes and ducts may be of considerable magnitude if the temperature of the surrounding medium differs appreciably from that of the fluid conveyed. Careful consideration must, therefore, be given to this factor in a properly designed system and adequate insulation provided, if necessary.

HEAT LOSSES FROM BARE PIPES

Heat losses from horizontal bare steel pipes, based on tests at *Mellon Institute* and calculated from the fundamental radiation and convection equations (Chapter 3), are given in Table 1. Heat losses from horizontal copper tubes and pipes with bright, lacquered and tarnished surfaces, are given in Tables 2, 3 and 4[1].

The monetary values of the heat losses given in Tables 1, 2, 3 and 4 may be obtained by means of Fig. 1 for various heating system efficiencies, temperature differences, and calorific values, and costs of coal. This chart, however, is intended for heat losses greater than 2 Btu per linear foot per hour per degree Fahrenheit temperature difference. To solve a problem, select the proper heat loss coefficient from Tables 1, 2, 3 or 4 and locate this value on the upper left-hand margin of the chart. Then draw lines in the order indicated by the dotted lines, the dollar value of the heat loss per 100 linear feet of pipe per 1000 hours being given on the upper right-hand scale. In using the chart, the cost of coal should also include the labor for handling it, boiler room expense, etc.

The area in square feet per linear foot of pipe is given in Table 5 for various standard pipe sizes, and Table 6 for copper tubing, while Table 7 gives the area in square feet of flanges and fittings for various standard pipe sizes. These tables can be used to advantage in estimating the amount of insulating cement required for various equipment.

Very often, when pipes are insulated, flanges and fittings are left bare so as to allow for easy access to the fittings in case of repairs. The fact that a pair of 8-in. standard flanges having an area of 2.41 sq ft would

[1]Heat loss from Copper Piping, by R. H. Heilman (*Heating, Piping and Air Conditioning*, September, 1933, p. 458).

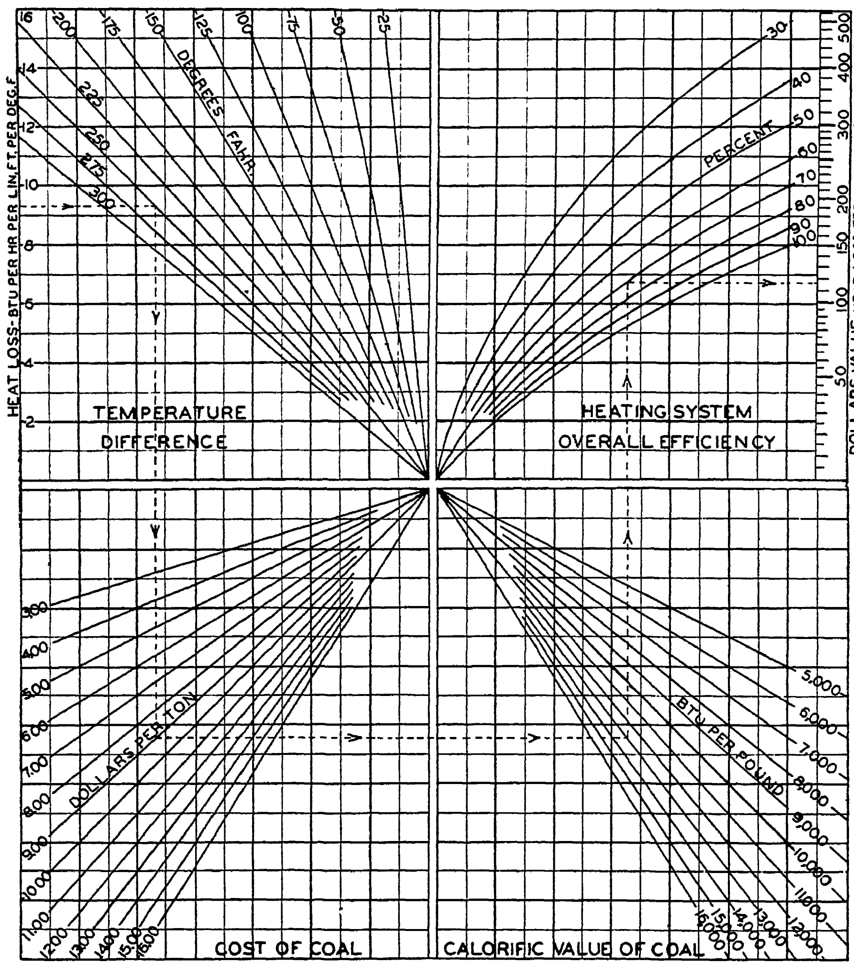


FIG. 1. CHART FOR ESTIMATING DOLLAR VALUE OF HEAT LOSS FROM BARE PIPES. (SEE TABLES 1, 2, 3 AND 4)[a]

[a]This chart is based on 100 linear feet per 1000 hours. For fractions or multiples of these actors, multiply by proper percentage.

lose, at 100 lb steam pressure, an amount of heat equivalent to more than a ton of coal per year shows the necessity for insulating such surfaces.

Example 1. Compute the total annual heat loss from 165 ft of 2 in. bare pipe in service 4000 hours per year. The pipe is carrying steam at 10 lb pressure and is exposed to an average air temperature of 70 F.

Solution. The pipe temperature is taken as the steam temperature, which is 239.4 F, obtained by interpolation from Table 8 Chapter 1. The temperature difference between the pipe and air = 239.4 − 70 = 169.4 F. By interpolation of Table 1 between temperature differences of 157.1 and 227.7 F, the heat loss from a 2 in. pipe at a temperature difference of 169.4 F is found to be 1.624 Btu per hour per linear foot per degree temperature difference. The total annual heat loss from the entire line = 1.624 × 169.4 × 165 (linear feet) × 4000 (hours) = 181,600 *Mb.*

Example 2. Coal costing $11.50 per ton and having a calorific value of 13,000 Btu per pound is being burned in the furnace supplying steam to the pipe line given in the previous example. If the system is operating at an overall efficiency of 55 per cent, determine the monetary value of the annual heat loss from the line.

Solution. The cost of heat per 1000 *Mb* supplied to the system = 1,000,000 × 11.5

TABLE 1. HEAT LOSSES FROM HORIZONTAL BARE STEEL PIPES

Expressed in Btu per hour per linear foot per degree Fahrenheit difference in temperature between the pipe and surrounding still air at 70 F

Nominal Pipe Size (Inches)	Hot Water				Steam		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	299.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.455	0.495	0.546	0.584	0.612	0.706	0.760
¾	0.555	0.605	0.666	0.715	0.748	0.866	0.933
1	0.684	0.743	0.819	0.877	0.919	1.065	1.147
1¼	0.847	0.919	1.014	1.086	1.138	1.324	1.425
1½	0.958	1.041	1.148	1.230	1.288	1.492	1.633
2	1.180	1.281	1.412	1.512	1.578	1.840	1.987
2½	1.400	1.532	1 683	1.796	1.883	2.190	2.363
3	1.680	1.825	2.010	2.153	2.260	2.630	2.840
3½	1.900	2.064	2.221	2.433	2.552	2.974	3.215
4	2.118	2.302	2.534	2.717	2.850	3.320	3.590
5	2.580	2.804	3.084	3.303	3.470	4.050	4.385
6	3.036	3.294	3.626	3.886	4.074	4.765	5.160
8	3.880	4.215	4.638	4.960	5.210	6.100	6.610
10	4.760	5.180	5.680	6.090	6.410	7.490	8.115
12	5.590	6.070	6.670	7.145	7.500	8.800	9.530

TABLE 2. HEAT LOSS FROM HORIZONTAL BARE BRIGHT COPPER PIPE

Expressed in Btu per hour per linear foot per degree Fahrenheit between the pipe and surrounding still air at 70 F

Nominal Pipe Size (Inches)	Hot Water (Type K Copper Tube)				Steam (Standard Pipe Size Pipe)		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.180	0.210	0.218	0.229	0.299	0.338	0.355
¾	0 236	0.275	0.291	0.307	0.357	0.408	0.418
1	0 290	0 338	0 354	0 373	0.440	0.492	0.523
1¼	0.340	0 400	0.418	0.443	0.510	0.571	0.598
1½	0.390	0.463	0.473	0 507	0 598	0.671	0.710
2	0 490	0.525	0.600	0 628	0.719	0.813	0.851
2½	0 580	0.675	0 709	0 750	0.840	0.953	1.008
3	0.680	0.788	0 848	0 871	0.987	1.107	1.165
3½	0.760	0 888	0 946	1 000	1.114	1.235	1.307
4	0.940	1.000	1.045	1.107	1.210	1.361	1.456
4½	.....	...	...	....	1.335	1.495	1.488
5	1.020	1.200	1.255	1.320	1.465	1.670	1.755
6	1.160	1.375	1.410	1.500	1.685	1 890	1.942
8	1.460	1.725	1.820	1.890	2.100	2.373	2.510

(dollars) ÷ 13,000 (Btu) × 2000 (lb) × 0.55 (efficiency) = $0.804. The total cost of heat lost per year = 0.804 × 181.6 (thousand Btu) = $146.00. (A closely approximate solution of such a problem may be made quickly by the use of the estimating chart given in Fig. 1.)

Table 3. Heat Loss from Bright Copper Pipe Given One Thin Coat of Clear Lacquer

Expressed in Btu per hour per linear foot per degree Fahrenheit between the pipe and surrounding still air at 70 F

Nominal Pipe Size (Inches)	Hot Water (Type K Copper Tube)				Steam (Standard Pipe Size Pipe)		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.240	0.265	0.282	0.307	0 401	0.461	0.478
¾	0.320	0.356	0.373	0.414	0.477	0.571	0.578
1	0.390	0.437	0.463	0.507	0.598	0 681	0.710
1¼	0.470	0.537	0 554	0.614	0.700	0.812	0 840
1½	0.540	0.612	0.645	0.714	0.830	0.966	0.990
2	0.690	0.762	0 818	0.892	1.005	1.164	1.201
2½	0.840	0.937	0.991	1.085	1.178	1.361	1.420
3	0.960	1.025	1.135	1.270	1.400	1 625	1.700
3½	1.100	1.250	1.318	1.442	1.580	1 845	1.905
4	1.241	1.400	1.480	1.556	1.750	2.040	2.130
4½	....	.....	.....	.....	1.910	2.240	2.350
5	1.480	1.685	1.790	1.965	2.130	2.415	2.610
6	1.700	1.936	2.052	2.272	2.450	2.810	2.990
8	2.200	2.500	2.630	2.854	3.120	3.425	3.730

Table 4. Heat Loss from Horizontal Tarnished Copper Pipe

Expressed in Btu per hour per linear foot per degree Fahrenheit between the pipe and surrounding still air at 70 F

Nominal Pipe Size (Inches)	Hot Water (Type K Copper Tube)				Steam (Standard Pipe Size Pipe)		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.250	0.287	0.300	0.321	0.433	0.500	0.530
¾	0.340	0.381	0.409	0.429	0.533	0.543	0.654
1	0.440	0.475	0.509	0.536	0.636	0.746	0.803
1¼	0.500	0.559	0.618	0.622	0.764	0.878	0.934
1½	0.580	0.656	0.710	0.750	0.904	1.053	1.120
2	0.730	0.825	0.890	0.957	1.101	1.273	1.364
2½	0.880	1.000	1.091	1.143	1.305	1.490	1.605
3	1.040	1.175	1.272	1.343	1.560	1.800	1.940
3½	1.180	1.350	1.454	1.535	1.750	2.020	2.170
4	1.460	1.500	1.635	1.715	1.941	2.240	2.430
4½	.....	.....	.....	.....	2.131	2 465	2.650
5	1.600	1.812	1.980	2.071	2.387	2.770	2.990
6	1.840	2.125	2.270	2.430	2.740	3.210	3.440
8	2.400	2.685	2.910	3.110	3.310	4.050	4.370

HEAT LOSSES FROM INSULATED PIPES

The conductivities of various materials used for insulating steam and hot water systems are given in Table 8. They are given as functions of the mean temperatures or the mean of the inner and outer surface tem-

TABLE 5. RADIATING SURFACE PER LINEAR FOOT OF PIPE

Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)
½	0.22	2	0.622	5	1.456
¾	0.275	2½	0.753	6	1.734
1	0.344	3	0.917	8	2.257
1¼	0.435	3½	1.047	10	2.817
1½	0.498	4	1.178	12	3.338

TABLE 6. RADIATING SURFACE PER LINEAR FOOT OF COPPER TUBING

Outside diameter ⅛ in. greater than nominal size

Tube Size (Inches)	Surface Area (Sq Ft)	Tube Size (Inches)	Surface Area (Sq Ft)	Tube Size (Inches)	Surface Area (Sq Ft)
½	0.164	2	0.556	5	1.342
¾	0.229	2½	0.687	6	1.604
1	0.295	3	0.818	8	2.128
1¼	0.360	3½	0.949	..	.....
1½	0.426	4	1.080	..	.....

TABLE 7. AREAS OF FLANGED FITTINGS, SQUARE FEET[a]

Nominal Pipe Size (Inches)	Flanged Coupling		90 Deg Ell		Long Radius Ell		Tee		Cross	
	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy
1	0.320	0.438	0.795	1.015	0.892	1.083	1.235	1.575	1.622	2.07
1¼	0.383	0.510	0.957	1.098	1.084	1.340	1.481	1.925	1.943	2.53
1½	0.477	0.727	1.174	1.332	1.337	1.874	1.815	2.68	2.38	3.54
2	0.672	0.848	1.65	2.01	1.84	2.16	2.54	3.09	3.32	4.06
2½	0.841	1.107	2.09	2.57	2.32	2.76	3.21	4.05	4.19	5.17
3	0.945	1.484	2.38	3.49	2.68	3.74	3.66	5.33	4.77	6.95
3½	1.122	1.644	2.98	3.96	3.28	4.28	4.48	6.04	5.83	7.89
4	1.344	1.914	3.53	4.64	3.96	4.99	5.41	7.07	7.03	9.24
4½	1.474	2.04	3.95	5.02	4.43	5.46	6.07	7.72	7.87	10.07
5	1.622	2.18	4.44	5.47	5.00	6.02	6.81	8.52	8.82	10.97
6	1.82	2.78	5.13	6.99	5.99	7.76	7.84	10.64	10.08	13.75
8	2.41	3.77	6.98	9.76	8.56	11.09	10.55	14.74	13.44	18.97
10	3.43	5.20	10.18	13.58	12.35	15.60	15.41	20.41	19.58	26.26
12	4.41	6.71	13.08	17.73	16.35	18.76	19.67	26.65	24.87	34.11

[a]Including areas of accompanying flanges bolted to the fitting.

peratures of the insulations. It should be emphasized that they are the average values obtained from a number of tests made on each type of material, also that all variables due to differences in thickness, pipe sizes, and air conditions are eliminated. Individual manufacturer's materials will, of course, vary in conductivity to some extent from these values.

resistance to heat flow inherent in the insulation itself. The maximum increase in heat loss due to air velocity ranges from about 30 per cent in the case of 1-in. thick insulation, to about 10 per cent in the case of 3-in. thick insulation, provided that the insulation is thoroughly sealed so that air can flow only over the surface.

If the conditions are such that the air may circulate through cracks and crevices in the insulation, the increases may be far greater than those given. Therefore, it is essential that insulation be sealed as tightly as

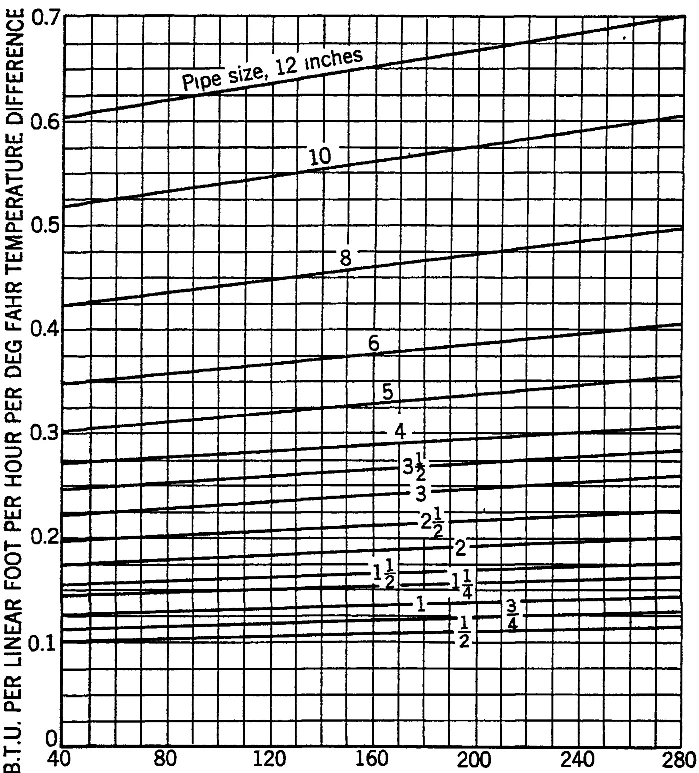


Fig. 4. Heat Loss Through 2 In. Thick 85 per cent Magnesia Type Covering

possible. Pipe insulation exposed to the elements should be thoroughly waterproofed.

Example 3. If the steam line given in Examples 1 and 2 is covered with 1 in. thick 85 per cent magnesia, determine the resulting total annual loss through the insulation. Also compute the monetary value of the annual saving and the percentage of saving over the heat loss from the bare pipe.

Solution. By referring to Fig. 2, the coefficient for 1 in. magnesia on a 2 in. pipe is found to be 0.285 Btu per hour per linear foot of pipe per degree temperature difference at a temperature difference of 169.4 F. The total hourly loss per linear foot of pipe will then be $0.285 \times 169.4 = 48.3$ Btu. The total annual loss through the insulation = 48.3×165 (linear feet) $\times 4000$ (hours) = 31,900 *Mb*. The annual bare pipe loss as

determined in the solution of Example 1 was found to be 181,600 *Mb*. The saving due to insulation is then 181,600 − 31,900 = 149,700 *Mb* per year.

From the solution of Example 2, it was found that the heat supplied to the system cost $0.804 per thousand *Mb*. Therefore, the monetary value of the saving = 0.804 (dollars) × 149.7 (thousand *Mb*) = $120.36, or 82.4 per cent of the cost when using uninsulated pipe.

TABLE 8. CONDUCTIVITY (k) OF VARIOUS TYPES OF INSULATING MATERIALS FOR MEDIUM AND HIGH TEMPERATURE PIPES[a]

Types of Insulating Materials	Mean Temperature, Deg F				
	100	200	300	400	500
85 per cent Magnesia Type	0.359	0.403	0.448	0.493	0.539
Corrugated Asbestos Type (4 Plies per 1 in. thick)	0.495	0.618	0.741	0.864	-------
Corrugated Asbestos Type (8 Plies per 1 in. thick)	0.505	0.598	0.692	0.786	-------
Laminated Asbestos Type (30-40 Laminations per 1 in. thick)	0.326	0.380	0.434	0.488	0.543
Laminated Asbestos Type (14-20 Laminations per 1 in. thick)	0.374	0.445	0.518	0.589	0.662
Mineral Wool Type	0.350	0.410	0.470	0.530	0.590
High Temperature Type (Diatomaceous Earth and Asbestos)	0.576	0.614	0.652	0.689	0.726
Brown Asbestos Type (Felted Fiber)	0.338	0.396	0.453	0.510	0.568

[a]From tests conducted at *Mellon Institute*.

TABLE 9. PIPE COVERING FACTORS

Types of Insulating Materials	Temperature Difference, Pipe to Air, Deg F					
	100	200	300	400	500	600
85 per cent Magnesia Type	1.050	1.024	0.997	0.971	0.944	0.918
Corrugated Asbestos Type (4 Plies per 1 in. thick)	1.425	1.465	1.505	1.545	-------	-------
Corrugated Asbestos Type (8 Plies per 1 in. thick)	1.435	1.437	1.438	1.440	-------	-------
Laminated Asbestos Type (30-40 Laminations per 1 in. thick)	0.969	0.960	0.951	0.942	0.933	0.924
Laminated Asbestos Type (14-20 Laminations per 1 in. thick)	1.103	1.104	1.105	1.106	1.107	1.108
Mineral Wool Type	1.023	1.028	1.033	1.038	1.043	1.048
High Temperature Type (Diatomaceous Earth and Asbestos)	1.560	1.489	1.418	1.347	1.276	1.205
Brown Asbestos Type (Felted Fiber)	1.003	0.997	0.990	0.984	0.977	0.971

LOW TEMPERATURE PIPE INSULATION

Surfaces maintained at temperatures lower than the surrounding air are insulated to reduce the flow of heat and to prevent condensation and frost. The insulating material should absorb a minimum amount of moisture, because the absorption of moisture substantially increases the conductivity of the material. This property is particularly important in the insulation of surfaces that are below the dew-point of the surrounding

air. In such cases, due to vapor pressure difference, it is necessary to seal the surface of the insulating material against the penetration of water vapor which would condense within the material, causing a serious increase in heat flow, possible breakdown of the material, and corrosion of metal surfaces. An insulating material with a high degree of moisture absorption might pick up moisture before application and then, when

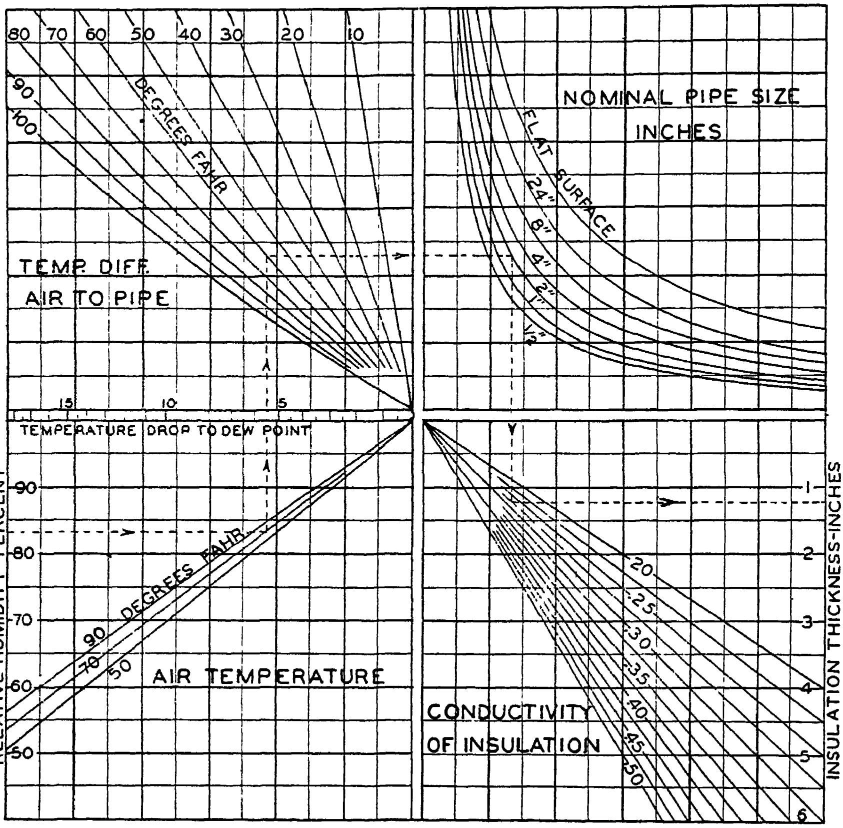


FIG. 5. THICKNESS OF PIPE INSULATION TO PREVENT SWEATING[a]

[a]Solve problems by drawing lines as indicated by dotted line, entering chart at lower left hand scale.

the seal is in place and the temperature of the insulated surface reduced, release that moisture to the cold surface.

The thickness of insulation required to prevent sweating is that thickness which will raise the temperature of the outer surface of the insulation to a point slightly higher than the dew-point for the corresponding air temperature and relative humidity. The difference in temperature between the air and the dew-point for various humidities can be readily ascertained from a psychrometric chart.

The approximate required thickness of insulation to prevent conden-

sation on pipes and flat metallic surfaces may be obtained from Fig. 5. The maximum permissible temperature drop is indicated at the point where the guide line passes through the horizontal scale at the left center of the chart. This temperature drop represents the difference between the dry-bulb temperature and the dew-point temperature for the conditions involved. (See discussion of Condensation in Chapter 4.) The surface resistances used for calculating the family of curves in Fig. 5 are based on tests made on canvas covered pipe insulation surfaces at *Mellon Institute.* However, it has been found that the resistance for asphaltic and roofing surfaces is practically the same as for canvas surfaces, so that the curves may be followed with no alteration for surfaces commonly used.

Heat gains for pipes insulated with a material having a conductivity of

TABLE 10. HEAT GAINS FOR INSULATED COLD PIPES

Rates of heat transmission given in Btu per hour per degree Fahrenheit temperature difference between fluid in pipe and surrounding still air

Based on materials having conductivity, k = 0.30

Nominal Pipe Size (Inches)	Ice Water Thickness			Brine Thickness			Heavy Brine Thickness		
	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface
½	1.5	0.110	0.502	2.0	0.098	0.446	2.8	0.087	0.394
¾	1.6	0.119	0.431	2.0	0.111	0.405	2.9	0.094	0.340
1	1.6	0.139	0.403	2.0	0.124	0.352	3.0	0.104	0.294
1¼	1.6	0.155	0.357	2.4	0.131	0.300	3.1	0.113	0.260
1½	1.5	0.174	0.351	2.5	0.134	0 270	3.2	0.118	0.238
2	1.5	0.200	0.322	2.5	0.151	0.244	3.3	0.134	0.214
2½	1.5	0.228	0.303	2.6	0.170	0.226	3.3	0.147	0.197
3	1.5	0.269	0.293	2.7	0.186	0.202	3.4	0.162	0.176
3½	1.5	0.295	0.282	2.9	0.191	0.183	3.5	0.176	0.167
4	1.7	0.294	0.248	2.9	0.209	0.176	3.7	0.182	0.154
5	1.7	0.349	0.239	3.0	0.241	0.165	3.9	0.202	0.138
6	1.7	0.404	0.233	3.0	0 259	0.150	4.0	0.228	0.130
8	1.9	0.455	0.201	3.0	0 318	0.140	4.0	0.263	0.116
10	1.9	0.559	0.198	3.0	0 383	0.135	4.0	0.309	0.110
12	1.9	0 648	0.194	3.0	0.438	0.131	4.0	0.364	0.108

0.30 Btu per square foot per hour per degree Fahrenheit difference per inch thickness are given in Table 10.

INSULATION OF PIPES TO PREVENT FREEZING

If the surrounding air temperature remains sufficiently low for an ample period of time, insulation cannot prevent the freezing of still water, or of water flowing at such a velocity that the quantity of heat carried in the water is not sufficient to take care of the heat losses which will result and cause the temperature of the water to be lowered to the freezing point. Insulation can materially prolong the time required for the water to give up its heat, and if the velocity of the water flowing in the pipe is maintained at a sufficiently high rate, freezing may be prevented.

Table 11 may be used for making estimates of the thickness of insulation necessary to take care of still water in pipes at various water and

surrounding air temperature conditions. Because of the damage and service interruptions which may result from frozen water in pipes, it is essential that an efficient insulation be utilized. This table is based on the use of a material having a conductivity of 0.30. The initial water temperature is assumed to be 10 F above, and the surrounding air temperature 50 F below the freezing point of water (temperature difference, 60 F).

The last column of Table 11 gives the minimum quantity of water at initial temperature of 42 F which should be supplied every hour for each linear foot of pipe, in order to prevent the temperature of the water from being lowered to the freezing point. The weights given in this column should be multiplied by the total length of the exposed pipe line expressed in feet. As an additional factor of safety, and in order to provide against

Table 11. Data for Estimating Requirements to Prevent Freezing of Water in Pipes with Surrounding Air at −18 F

Nominal Pipe Size (Inches)	Number of Hours to Cool 42 F Water to Freezing Point			Water Flow Required at 42 F to Prevent Freezing, Pounds per Linear Foot of Pipe per Hour		
	Thickness of Insulation in Inches (Conductivity, $k = 0.30$)					
	2	3	4	2	3	4
½	0.42	0.50	0.57	0.54	0.45	0.40
1	0.83	1.02	1.16	0.68	0.55	0.48
1½	1.40	1.74	2.02	0.84	0.68	0.58
2	1.94	2.48	2.90	0.95	0.75	0.64
3	3.25	4.27	5.08	1.24	0.94	0.79
4	4.55	6.02	7.20	1.47	1.11	0.93
5	5.92	7.96	9.69	1.73	1.29	1.06
6	7.35	9.88	12.20	1.98	1.46	1.19
8	10.05	13.90	17.25	2.46	1.78	1.43
10	13.00	18.10	22.70	2.96	2.12	1.70
12	15.80	22.20	28.10	3.43	2.45	1.93

temporary reductions in flow occasioned by reduced pressure, it is advisable to double the rates of flow listed in the table. It must be emphasized that the flow rates and periods of time designated apply only for the conditions stated. To estimate for other service conditions the following method of procedure may be used.

If water enters the pipe at 52 F instead of 42 F, the time required to cool it to the freezing point will be prolonged to twice that given in the table, or the rate of flow of water may be reduced so that the quantity required will be one-half that shown in the last column of Table 11. However, if the water enters the pipe at 34 F it will be cooled to 32 F in one-fifth of the time given in the table. It will then be necessary to increase the rate of flow so that five times the specified quantity of water will have to be supplied in order to prevent freezing.

If the minimum air temperature is − 38 F (temperature difference 80 F) instead of −18 F, the time required to cool the water to the freezing point will be 60/80 of the time given in the table, or the necessary quantity of water to be supplied will be 80/60 of that given.

In making calculations to arrive at the values given in Table 11, the loss of heat stored in the insulation, the effect of a varying temperature

difference due to the cooling of pipe and water, and the resistance of the outer surface of the insulation to the transfer of heat to the air have all been neglected. When these factors enter into the computations it is necessary to enlarge the factor of safety. Also as stated, the time shown in the table is that required to lower the water to the freezing point. A longer period would be required to freeze the water but the danger point is reached when freezing starts. The flow of water will stop and the entire line will be in danger as soon as the water freezes across the section of the pipe at any point.

When water must remain stationary longer than the times designated in Table 11, the only safe way to insure against freezing is to install a steam or hot water line or to place an electric resistance heater along the side of the exposed water line. The heating system and the water line are then insulated so that the heat losses from the heating system are not

TABLE 12. THICKNESSES OF PIPE INSULATION ORDINARILY USED INDOORS[a]

Steam Pressures (Lb Gage) or Conditions	Steam Temperatures Degrees Fahrenheit	Thickness of Insulation		
		Pipes Larger Than 4 In.	Pipes 2 In. to 4 In.	Pipes ½ In. to 1½ In.
0 to 25	212 to 267	1 in.	1 in.	1 in.
25 to 100	267 to 338	1½ in.	1 in.	1 in.
100 to 200	338 to 388	2 in.	1½ in.	1 in.
Low Superheat	388 to 500	2½ in.	2 in.	1½ in.
Medium Superheat	500 to 600	3 in.	2½ in.	2 in.
High Superheat	600 to 700	3½ in.	3 in.	2 in.

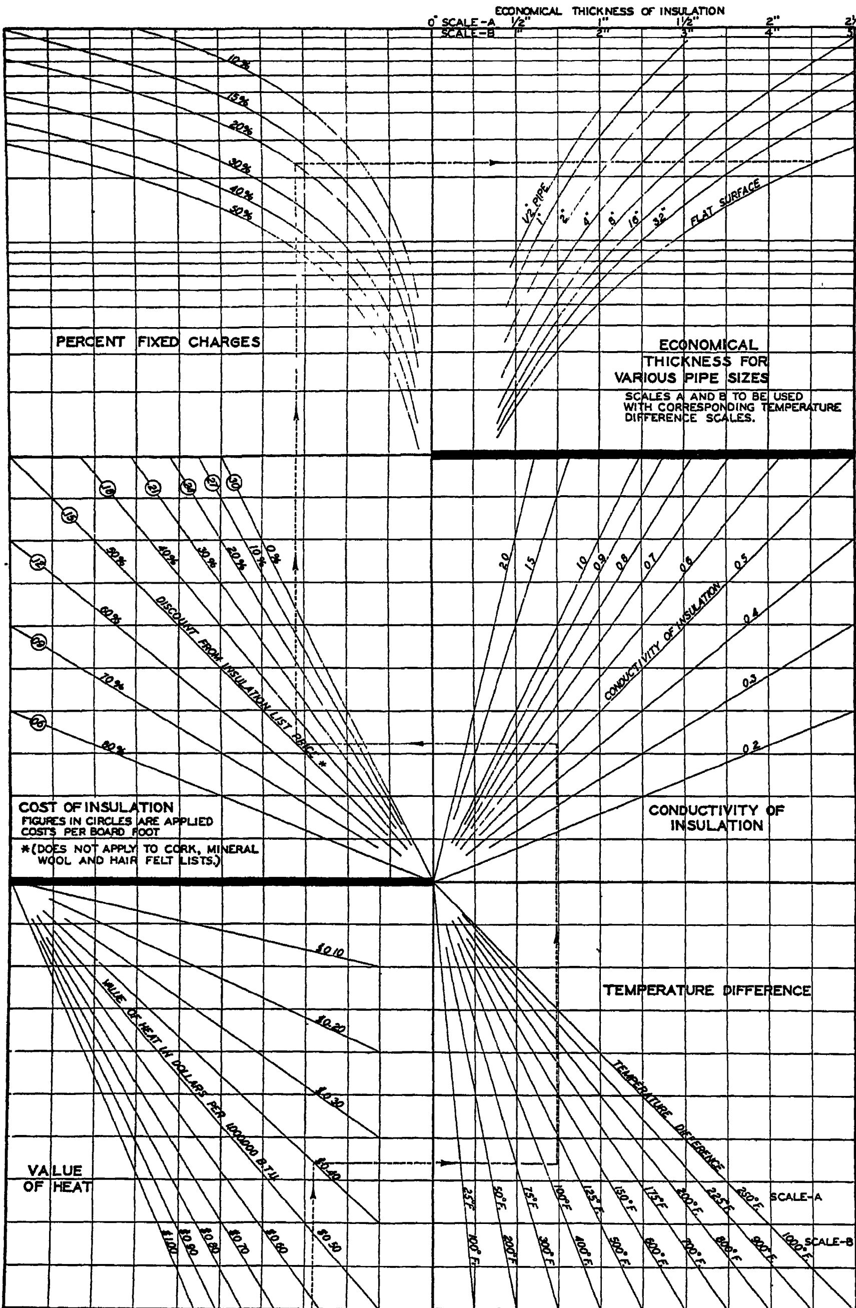
[a]All piping located outdoors or exposed to weather is ordinarily insulated to a thickness ½ in. greater than shown in this table, and covered with a waterproof jacket.

excessive, and the heating effect is concentrated against the water pipe where it is needed. For this form of protection 2 in. of an efficient insulation may be applied.

ECONOMICAL THICKNESS OF PIPE INSULATION

The thicknesses of insulation which ordinarily are used for various temperature conditions are given in Table 12. Where a thorough analysis of economic thickness is desired this may be accomplished through the use of the chart, Fig. 6.

The dotted line on the chart illustrates its use in solving a typical example. In using the chart, start with the scale at the left bottom margin representing the given number of hours of operation per year; then proceed vertically to the line representing the given value of heat; thence horizontally to the right, to the line representing the given temperature difference; thence vertically to the line representing the conductivity of the given material; thence horizontally, to the left, to the line representing the given discount on that material; thence vertically to the curve representing the required per cent return on the investment; thence horizontally to the right, to the curve representing the given pipe size; thence vertically to the scale at the top right margin where the economical thickness may be read off directly.



(L. B. McMillan, *Proc. National Dist. Heating Assn.*, Vol. 18, p. 138).

FIG. 6. CHART FOR DETERMINING ECONOMICAL THICKNESS OF PIPE INSULATION

UNDERGROUND PIPE INSULATION

Underground steam distribution lines are carried in protective structures of various types, sizes and shapes. (See Chapter 17.) Detailed data on commonly used forms of tunnels and conduit systems have been published by the *National District Heating Association*[2].

Pipes in tunnels are covered with sectional insulation to provide maximum thermal efficiency and are also finished with good mechanical protection in the form of metal or waterproofing membrane outer jackets. Conduit systems are in more general use than tunnels. Pipes carried in conduits may be insulated with sectional insulation; however, the more usual practice is to fill the entire section of the conduit around the pipes with high quality, loose insulating material. The insulation must be kept dry at all times, and for this purpose effective waterproofing membranes enclose the insulation. A drainage system is also provided to divert water which may tend to enter the conduit.

The economical thickness of insulation for underground work is difficult to determine accurately due to the many variables which have to be considered. As a result of theories[3] previously developed, together with other experimental data which have been presented, the usual endeavor is to secure not less than 90 per cent efficiency for underground piping. Table 13 can be used as a guide in arriving at the minimum thickness of loose insulation fills to use for laying out conduit systems. Other factors such as the number of pipes and their combination of sizes, as well as the standard conduit sizes, are primary controlling factors in the amount and thickness of insulation for use.

Table 13. Thickness of Loose Insulation for Use as Fill in Underground Conduit Systems

Steam Pressures (Lb Gage) or Conditions	Steam Temperatures Degrees Fahrenheit	Minimum Thickness of Insulation in Inches					Minimum Distance Between Steam and Return
		Steam Lines			Return Lines		
		Pipes Less than 4 In.	Pipes 4 In. to 10 In.	Pipes Larger than 12 In.	Pipes Less than 4 In.	Pipes 4 In. and Larger	
Hot Water, or 0 to 25	212 to 267	1½	2	2½	1¼	1½	1
25 to 125	267 to 352	2	2½	3	1¼	1½	1¼
Above 125, or superheat	352 to 500	2½	3	3½	1¼	1½	1½

When sectional insulation is applied to lines in tunnels or conduits, usual practice is to apply the most efficient materials ½ in. less in thickness than that determined by the use of Fig. 6. The data in Fig. 6 are based on conditions of insulation exposed to the air, whereas normal ground temperature is substituted for air temperature in determining the temperature difference for use with the chart when applying it for underground pipe line estimates.

[2]Handbook of the *National District Heating Association*, Second Edition, 1932.

[3]Theory of Heat Losses from Pipes Buried in the Ground, by J. R. Allen (A.S.H.V.E. Transactions, Vol. 26, 1920, p. 335).

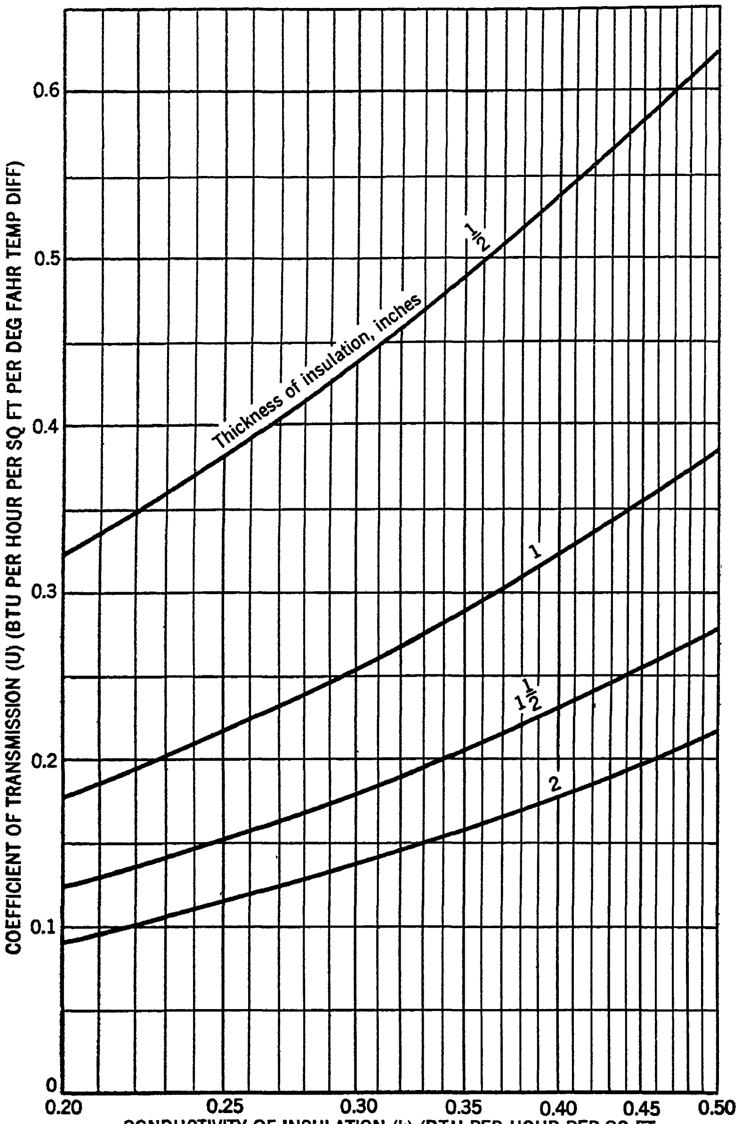


FIG. 7. HEAT LOSS COEFFICIENTS FOR INSULATED DUCTS.[a]

[a]For round ducts less than 30 in. diameter, increase heat transmission values by the following percentages:

THICKNESS OF INSULATION (Inches)	½	1	1½	2
21 to 30 in. Duct Diameter	1%	2%	3%	4%
12 to 21 in. Duct Diameter	3%	5%	7%	9%

HEAT LOSSES FROM DUCTS

The thermal transmission coefficient U for an uninsulated metal duct can be obtained from the equation:

$$U = \frac{1}{\frac{1}{f_i} + \frac{1}{f_o}} \tag{1}$$

where

U = thermal transmittance, Btu per square foot per hour per degree Fahrenheit difference in temperature between the average temperature inside the duct and the air outside the duct.

f_i = film conductance inside the duct, Btu per hour per square foot per degree Fahrenheit.

f_o = film conductance outside the duct, Btu per hour per square foot per degree Fahrenheit.

Film conductance f_i for air flowing in ducts apparently depends only on the velocity of the air and the diameter of the duct. A fairly reliable inside coefficient can be calculated from Schultz's modified equation:

$$f_i = \frac{0.32\, V_o^{0.8}}{D^{0.25}} \tag{2}$$

where

V_o = velocity of air in duct, feet per second.
D = diameter of duct, feet.

Film conductance f_o depends on a number of variables including temperature, diameter, and emissivity of the outer surface and can readily be calculated from data in Chapter 3. From this explanation, it is seen that it is unwise to recommend a given value of U for all uninsulated metal ducts.

The heat loss from a given length of duct can be expressed by:

$$Q = UPL \left[\left(\frac{t_1 + t_2}{2} \right) - t_3 \right] \tag{3}$$

The heat given up by the air in the duct is:

$$Q = 0.24\, M\, (t_1 - t_2) = 14.4\, AVd\, (t_1 - t_2) \tag{4}$$

Equating 3 and 4 enables the determination of the temperature drop in the duct:

$$\frac{t_1 + t_2 - 2t_3}{t_1 - t_2} = \frac{28.8\, AVd}{UPL}$$

Let $x = \frac{28.8\, AVd}{UPL}$ for rectangular ducts, $= \frac{7.2\, DVd}{UL}$ for round ducts, solving for t_1 and t_2:

$$t_1 = \frac{t_2\,(x + 1) - 2t_3}{(x - 1)} \tag{5}$$

$$t_2 = \frac{t_1\,(x - 1) + 2t_3}{(x + 1)} \tag{6}$$

For low velocities and long ducts of small cross-section, a somewhat more accurate formula may be used as follows:

$$t_2 = \frac{t_1 - t_3}{e^{\left(\frac{UPL}{14.4\ AdV}\right)}} + t_3 \tag{7}$$

In these equations

Q = heat loss through duct walls, Btu per hour.
U = thermal transmission coefficient, Btu per square foot per hour per degree Fahrenheit.
P = perimeter of duct, feet.
L = length of duct, feet.
t_1 = temperature of air entering duct, degree Fahrenheit.
t_2 = temperature of air leaving duct, degree Fahrenheit.
t_3 = temperature of air surrounding duct, degree Fahrenheit.
M = weight of air per hour, through the duct, pounds.
A = cross-sectional area of duct, feet.
D = diameter of round ducts, feet.
V = velocity of air in the duct, feet per minute, at specified temperature.
d = density of air, pounds per cubic foot, at the specified temperature at which V is measured.
e = naperian base of logarithms = 2.718.

In using Equations 5, 6 and 7, one of the duct air temperatures will be unknown and will be solved for by substitution of the other known or assumed values.

Heat loss coefficients for insulated ducts with various conductivities are given in Fig. 7. The conductivities of various materials, which are based on mean temperatures, ranging from about 70 to 90 F, will be found in Table 2 of Chapter 4. For cases where the mean temperature is other than that on which the test was conducted, a correction should be made. However, in most cases the effect of this factor will be small and may be neglected.

Example 4. Determine the entering air temperature and heat loss for a duct 24 × 36 in. cross-section and 70 ft in length, insulated with ½ in. of a material having a conductivity of 0.35 Btu at 86 F mean temperature, carrying air at a velocity of 1200 fpm, measured at 70 F, to deliver air at 120 F with air surrounding the duct at 40 F.

Solution. Referring to Fig. 7, the overall heat transmission coefficient is found to be 0.49 Btu. From Table 6, Chapter 1 the density of air at 70 F and 29.92 in. Hg. is found to be 0.0749 lb per cubic foot. Substituting these and the other given values in Equation 5:

$$x = \frac{28.8 \times 6 \times 0.0749 \times 1200}{0.49 \times 10 \times 70} = 44.4$$

$$t_1 = \frac{120\ (44.4 + 1) - 80}{44.4 - 1} = 123.7$$

Substituting in Equation 3:

$$Q = 0.49 \times 10 \times 70 \left[\left(\frac{123.7 + 120}{2}\right) - 40 \right]$$

$$Q = 28.010 \text{ Btu per hour.}$$

Chapter 44

ELECTRIC HEATING

Resistors, Heating Elements, Electric Heaters, Unit Heaters, Central Fan Heating, Electric Boilers, Electric Hot Water Heating, Heating Domestic Water Supply, Radiant Drying, Reversed Cycle Refrigeration, Auxiliary Electric Heating, Control, Calculating Capacities, Power Problems

ELECTRIC heating is steadily assuming a more important place in heating, ventilating and air conditioning installations, encouraged in many localities by reduced electric rates. Electric heating is flexible, clean, safe, convenient and easy to control. It has many basic principles in common with fuel heating, but there are also important differences. When heat is delivered by wire, no combustion process is necessary, either at a central plant or at the individual room units. The output of an electric heater is a fixed constant, unaffected by the temperature of the surrounding air and it follows that the total load on an electric heating system is the total wattage of connected electric heaters, regardless of weather conditions. The main obstacle to the more general adoption of electric heating for buildings is the cost of the electricity itself.

All heat is a form of energy. Fuels hold stored chemical energy which is released into heat by combustion. Electrical power is a form of energy which can be released into heat by passing it through a resisting material. Both fuel and electric heating have two divisions: *first*, the conversion of energy into heat; *second*, the distribution and practical use of the heat after it is produced.

In converting the chemical energy of fuels into heat by combustion, there is necessarily a considerable variation in thermal efficiency. This is not true, however, when converting electric power into heat, as 100 per cent of the energy applied to the resistor is always transformed into heat. In electric heating practice no concern need be given to efficiencies of heat production, but rather to efficiencies of heat utilization. The problem is to distribute the electrically produced heat units in such manner as to obtain conditions of maximum comfort with the minimum consumption of electricity.

DEFINITIONS

Definitions of general terms used in fuel heating are given in Chapter 47. Terms which apply particularly to electric heating are:

Electric Resistor: A material used to produce heat by passing an electric current through it.

Electric Heating Element: A unit assembly consisting of a resistor, insulated supports, and terminals for connecting the resistor to electric power.

Electric Heater: A complete assembly of heating elements with their enclosure, ready for installation in service.

RESISTORS AND HEATING ELEMENTS

Solids, liquids, and gases may be used as resistors, but most commercial electric heating elements have solid resistors, such as metal alloys, and non-metallic compounds containing carbon. In some types of electric boilers, water forms the resistor which is heated by an alternating current of electricity passing through it. One of the more common resistors is nickel-chromium wire or ribbon which, in order to avoid oxidation, contains practically no iron.

Commercial electric heating elements are made in many types. Some have resistors exposed to the air being heated. The resistors may be coils of wire or metal ribbon, supported by refractory insulation, or they may be non-metallic rods, mounted on insulators. This type of element is used extensively for operation at high temperatures when radiant heat is desired, also at low temperatures for convection and fan circulation heating, especially in large installations.

Some elements have metallic resistors embedded in a refractory insulating material, encased in a protective sheath of metal. Fins or extended surfaces may be used to add heat-dissipating area. Elements are made in many forms, such as strips, rings, plates and tubes. Strip elements are used for clamping to surfaces requiring heat by conduction, and in some types of convection air heaters. Ring and plate elements are used in electric ranges, waffle irons, and in many small air heaters. Tubular elements may be immersed in liquids, cast into metal, and, when formed into coils, used in electric ranges and air heaters. Cloth fabrics woven from flexible resistor wires and asbestos thread, are used for many low temperature purposes such as heating pads and aviators' clothing.

Special incandescent lamps are used as heating elements in certain applications where radiant heat is desired. These use carbon or tungsten filaments as resistors, and are designed to produce maximum energy in the infra-red portion of the spectrum.

ELECTRIC HEATERS

Electric heaters may be divided into three groups: conduction, radiant and convection.

Conduction electric heaters, which deliver most of their heat by actual contact with the object to be heated, are used in such applications as aviators' clothing, hot pads, foot warmers, soil heaters, ice melters, and water heaters. Conduction heaters are useful in conserving and localizing heat delivery at definite points. They are not suitable for general air heating.

Radiant electric heaters, which deliver most of their heat by radiation, have high temperature heating elements and reflectors to concentrate the heat rays in the desired directions. The immediate and pleasant

sensation of warmth which is caused by radiant heat makes this type desirable for temporary use where the heat rays can fall directly upon the body. They are not satisfactory for general air heating, as radiant heat rays do not warm the air through which they pass. They must first be absorbed by walls, furniture, or other solid objects which then give up the heat to the air. For a discussion of electrically heated panels as applied to radiant heating, see Chapter 45.

Gravity convection electric heaters, designed to induce thermal air circulation, deliver heat largely by convection, and should be located and used in much the same manner as steam and hot water radiators or convectors. They generally have heating elements of large area, with moderate surface temperature, enclosed to give proper stack effect to draw cold air from the floor line. The flexibility possible with electric heating elements should discourage the use of secondary mediums for heat transfer. Water

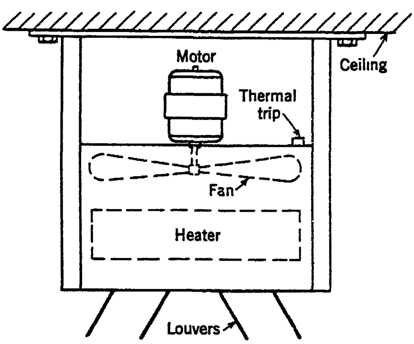


FIG. 1. CEILING MOUNTED UNIT HEATER

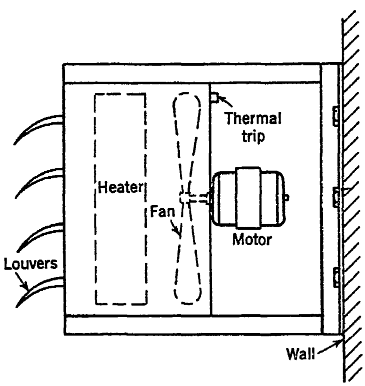


FIG. 2. WALL MOUNTED UNIT HEATER

and steam add nothing to the efficiency of an electric heater and entail expensive construction and maintenance.

UNIT HEATERS

Electric unit heaters include a built-in fan unit which circulates room air over the heating elements. Heaters of this type are manufactured in many designs and sizes, and can be located in the same manner as steam unit heaters.

Electric unit heaters are used in industrial plants, sub-stations, power houses, pumping stations, etc., where the power rate for electric heating is found to be favorable. In many large plants, such as flour mills, grain elevators, etc., in which there are a number of small offices, locker rooms, etc., scattered over wide areas, electric unit heaters are frequently economical in such locations. In small unattended stations, where freezing temperatures cannot be permitted, thermostatically-controlled electric unit heaters are frequently used to maintain a temperature above freezing. The best location for the heaters depends upon local circumstances as they can be mounted either on the ceiling to direct the air

downward, on the side wall about 7 ft from the floor, or near the floor line. Variations in design are necessary for different locations, but typical arrangements are indicated in Figs. 1 and 2.

The arrangement of the wiring circuits is very important for electric unit heaters. In principle they are all the same and include as essential elements an automatic control panel, a thermostat, and a master hand switch. All heaters should be designed with a safety thermal trip wired in series with the magnet coil of the control panel and with the hand switch and thermostat. A typical wiring diagram is shown in Fig. 3. This applies to a single phase power supply, but for 3 phase the only difference is to have a 3-pole panel and a heater arrangement for 3-phase connection.

Portable unit heaters are useful for temporary work, such as drying out damp rooms, or for warming rooms during construction.

CENTRAL FAN HEATING

Electric heating elements can be used for the prime source of heat in a central fan electric heating system or in the heating phase of a complete

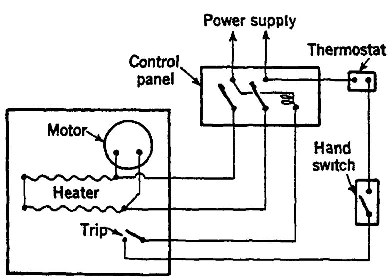


Fig. 3. Wiring Diagram for Unit Heater

air conditioning system. They can be used in the same manner as steam heating units for tempering, preheating or reheating the air at the main supply fan location and as booster heaters at the delivery terminals of the duct system. In the humidification phase of air conditioning electric heating elements can be used to provide moisture by the evaporation of water, or for controlling air washer dew-point temperatures when mounted as preheating units on the intake side of the air washer. (See Chapter 21.)

In coordinating the input of heat energy and the volume of air circulation, a basic difference between electric heating and steam heating enters into the problem. Steam is approximately a constant-temperature source of heat for any given pressure and a change in air volume flowing over steam coils does not greatly affect the temperatures of the delivered air. The amount of steam condensed (heat input) varies in proportion to the air volume, but the surface temperature of the steam coils remains about the same. Electric heat is quite different, having a constant input of energy. If the volume of air flow over electric heating elements is changed, and no change is made in the electrical power connections, there will be a corresponding change in the temperature of the air delivered.

This occurs because the electrical energy input remains constant and the surface temperature of the heating elements will vary as is necessary to force the air to accept all the heat. With electric heat the total heat is constant unless some compensating action is performed by control. Automatic variation of the electrical heat input synchronized properly with the air flow can be successfully accomplished by various special methods of control.

Electric heaters are useful in balancing the heat distribution in central fan heating systems. Even in those instances where steam is the principal heat source, the temperature of individual rooms can be controlled locally by separate electric booster heaters. These heaters can be installed in branch ducts or behind the air outlet grilles in each room. With this arrangement, the central heating unit distributes air at an average temperature, controlled from a thermostat centrally located, such as in the main return duct. The electric booster heaters may be controlled by thermo-

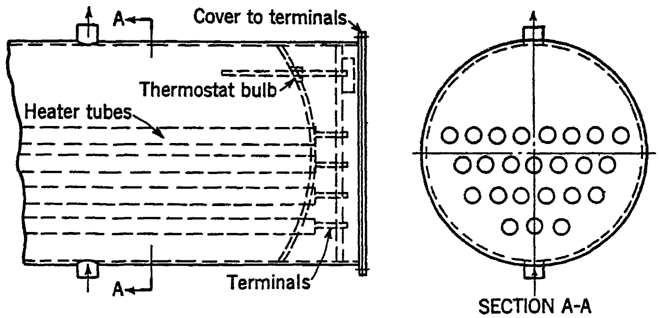


FIG. 4. RESISTANCE TYPE BOILER FOR STEAM OR HOT WATER

stats mounted in each individual room which permit the occupant to maintain any desired temperature independent of the rest of the building.

ELECTRIC BOILERS

Steam or hot water generating boilers using electric energy are entirely automatic and are well adapted to intermittent operation. Small electric boilers usually have heating elements of the enclosed metal resistor type immersed in the water. Boilers of this construction may be used either with direct or alternating current since the heat is delivered to the water by contact with the hot surfaces. To lessen the likelihood of the heating elements burning out, they should be of substantial construction, with a low heat density per unit of surface area and provision should be made for cleaning off desposits of scale which restrict the heat flow. A typical resistance type of steam or hot water boiler is shown in Fig. 4.

Large electric boilers are usually of the type employing water as the resistor, using immersed electrodes. With this type only alternating current can be used, as direct current would cause electrolytic deterioration. Such a type of electrode boiler is shown in Fig. 5.

Electric steam boilers are useful in industrial plants which require limited amounts of steam for local processes, and also for sterilizers, jacketed vessels and pressing machines which need a ready supply of steam. It sometimes is economical to shut down the main plant fuel burning boilers when the heating season ends, and to supply steam for summer needs with small electric steam boilers located close to the operation. In general, electric steam heating is confined to auxiliary or other limited applications. If the heating system is designed to use electricity exclusively, steam generating or distributing equipment is superfluous.

ELECTRIC HOT WATER HEATING

Electric water heating, using an electric boiler in place of a fuel burning boiler, like electric steam heating, is generally confined to auxiliary or

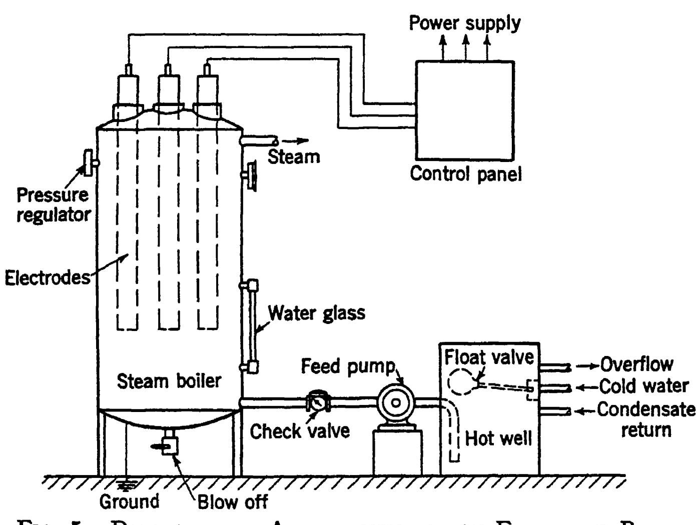


FIG. 5. DIAGRAMMATIC ARRANGEMENT OF AN ELECTRODE BOILER

other limited applications. The use of insulated water storage tanks, in which to store heat generated by electricity during off-peak hours at extremely low rates, is a development which has some special applications.

In this system of heating, the primary storage tank is simply a large, well-insulated, pressure type steel tank, equipped with electric heating elements and automatic time switches, which also have automatic limit controls for temperature and pressure. The heating system installed in the building may be of any standard individual radiator or fan-served indirect type or with provisions for the heating and humidification phases of an air conditioning system. A system of this kind requires very careful design to avoid excessive overall radiation losses during periods of low heat demand. It is also important to provide for sudden changes in heat demand. A typical hot water heating boiler is illustrated in Fig. 4.

HEATING DOMESTIC WATER BY ELECTRICITY[1]

Electric water heaters of the automatic storage type for domestic hot water supply are simple and reliable. In many sections of the country low electric rates have been established by the electric utilities to secure this load. In some localities, electric rate schedules divide the current used for water heating into two classifications, regular and off-peak. A time switch automatically limits use of the off-peak heating element to the hours of off-peak load, while the regular heating element is a stand-by at all times. Storage of this two-element type of water heater is larger than average to carry over the periods when the off-peak element is timed out, without too frequent demands on the regular heating element which takes the higher domestic lighting service rate. Some utilities now offer

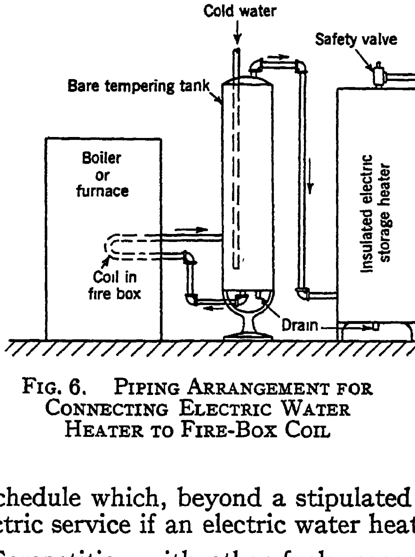


FIG. 6. PIPING ARRANGEMENT FOR CONNECTING ELECTRIC WATER HEATER TO FIRE-BOX COIL

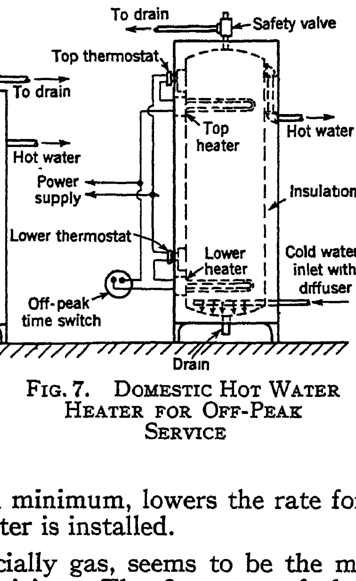


FIG. 7. DOMESTIC HOT WATER HEATER FOR OFF-PEAK SERVICE

a schedule which, beyond a stipulated minimum, lowers the rate for all electric service if an electric water heater is installed.

Competition with other fuels, especially gas, seems to be the major controlling factor in the use of electricity. The first cost of electric storage heaters is also greater than for gas, owing to the need for larger tank storage due to off-peak service and slower recuperating capacity.

In residential work, to effect a saving in the cost of operation, it is sometimes desirable to use a furnace coil or indirect heater in connection with an electric water heater. In this case it is important to make the proper connections in order to benefit by any heat obtained from the furnace and at the same time to prevent dangerous overheating. The proper piping connections are shown in Fig. 6, and in this case the electric heater will only furnish heat when insufficient heat is supplied from the furnace. This arrangement has a further advantage in the summertime in that the bare tank through which the cold water passes on its way to

[1]Test Results of Electric Water Heaters, by C. G. Hillier (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, November, 1936, p. 632). Fourteenth Range and Water Heater Survey (*Electric Light and Power*, August, 1940).

the electric heater serves as a tempering tank, absorbing heat from the basement air and requiring the use of less energy in the electric heater.

A typical domestic hot water heater as shown in Fig. 7 is arranged with upper and lower heating elements for the usual type of off-peak heating service. The lower heating element is under the control of the off-peak time switch. However, the upper heating element is usually connected to the line so that in case the supply of hot water in the tank becomes exhausted the top thermostat can turn on the top heater and heat a small supply of water. The top heater will not heat the water in the tank below its location, but when the off-peak period arrives the lower heater is turned on and the entire tank becomes heated.

RADIANT DRYING

Lacquers and similar surface films can be very effectively dried by radiation. Special electric lamp bulbs have been developed which give off a high percentage of infra-red and similar heat rays[2]. These are mounted in very efficient reflectors. For continuous manufacturing processes these reflectors are mounted in tunnels through which conveyors pass. For local applications, as for example paint drying in automobile repair shops, they may be mounted on portable racks.

In the application of this type of drying the composition of the paint or lacquer is important. In general, lacquers and those enamels using synthetic resins react most favorably. Other applications include the drying of ink, glue, and water, the softening of celluloid and bakelite for punching or shearing, and a wide variety of other uses.

REVERSED CYCLE REFRIGERATION

Reversed refrigeration is frequently referred to as a *heat pump* since the electric motor driving the refrigerating compressor furnishes the motive power to transfer heat from one temperature to a higher temperature level. The compressor acts as a reversible refrigerating unit to extract heat from the outdoor air in winter and deliver it indoors for heating purposes, and, by a reversal, to extract heat from the indoor air in summer and discharge it outdoors.

In normal use a refrigerating machine is arranged to remove heat and the heat removed is dissipated to the condenser cooling water. The driving energy is converted into heat, most of which is added to the heat removed and extracted. In so-called reversed refrigeration the heat removed together with the heat converted from the driving energy is utilized to heat the building. This conservation of the heat converted from the driving energy enables the reversed refrigeration to show a better performance in heating service than straight refrigeration can show in cooling service. In order to overcome the drop in capacity and in efficiency with lower outside temperatures, it is often desirable to use well-water instead of air as the source of heat. For a detailed description of this cycle see Chapter 25.

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[2]Infra-Red Lamps Speed Up Drying Operations (*Automotive Industries* 82:376-7; April 15, 1940). Invisible Rays Build Visible Profits, by H. M. Archer (*Electric Light and Power*, May, 1940). Radiant Energy Drying and Baking for Organic Finishing (*Metal Industry* 38:294-6; May, 1940).

AUXILIARY ELECTRIC HEATING

In conjunction with heating systems of other types, an auxiliary electric heating arrangement is a convenient means of caring for mild days in the spring and fall which require little heat to make a building comfortable. Likewise, such electric heating might be used on abnormally cold days to help out the main heating system and by this means reduce the necessary size of the system.

A few installations have been made using electric heating cable buried in the floors of bathrooms, etc., to provide auxiliary electric heating. At least one airplane hangar is heated in this manner.

Because of the feeling of comfort that a radiant type heater gives, bathrooms may be heated electrically with this type of heater while the rest of the house is cared for by some other system. Offices and rooms which require heat at periods when the main heating plant is shut down can be conveniently heated electrically.

CONTROL

Because the efficiency of electric heat production is the same for small and large units, it is possible to reduce heat waste to a minimum by applying local heating, locally controlled. Heaters are often controlled manually but thermostatic control is essential for economical operation. For duct systems having a variable volume of air flow the electric heater control must automatically vary the heat input in coordination with the changes in air volume and demand for heat.

CALCULATING CAPACITIES

The electric heating capacity required can be calculated from the heat requirement in Btu per hour by using the equation:

$$\frac{\text{Btu per hour}}{3413} = \text{kw rating of required electric heating} \tag{1}$$

For comparison with steam radiation:

$$1 \text{ kw} = \frac{3413 \text{ Btu}}{240} = 14.2 \text{ sq ft of steam radiation} \tag{2}$$

POWER PROBLEMS

The cost of electric energy varies because of several factors. Distribution costs differ for large and small users. The fact that electricity cannot be economically stored, but must be used as fast as it is generated, makes it impossible to operate electric plants at uniform loads; hence, even the time of use may affect the cost of electricity. Special low rates are sometimes available during certain prescribed hours of use.

Since the cost of production and distribution depends not only upon the quantity of energy used but also upon the maximum rate at which it is used, electric energy is often sold on a demand rate basis. In some

cases, the demand charge is based upon the rated connected load, in other cases, upon the maximum demand as indicated by a demand meter.

Homes are almost universally supplied with lighting current of 115 volts, which can only be used economically for small heaters. Usually the service lines will not permit more than plug-in devices. The Underwriters permit approved heaters of 1320 watts or less to be plugged into approved baseboard receptacles, but such heaters cannot be served on a circuit supplying much other load without overloading the fuses. There is an increasing trend toward supplying homes with three wire 115/230 volt service. Where homes have such service, larger heaters can be installed. For industrial purposes, heaters should be designed to use polyphase power, which is usually supplied at 220, 440 or 550 volts. All polyphase heaters should be balanced between phases.

REFERENCES

Electric Elements Well Adapted to the Air Conditioning Heating Cycle, by L. P. Hynes (*Heating, Piping and Air Conditioning*, January, 1940, p. 29).

Electric Air Heating Market (*Electrical West* 82:113; June, 1939).

Seven Jobs for Electric Heat (*Factory Management* 97:80-1; September, 1939).

Electric Heat for the Bathroom Floor (*Heating and Ventilating* 36:41; December, 1939).

Electric Heat Spreads in Sunny Climes (*Barron's National Financial Weekly* 20:5; January 8, 1940).

Electric Heating for Los Angeles Building (*Heating and Ventilating* 37:50-1; June, 1940).

Dracker Apartments, West Los Angeles, Calif. (*Heating and Ventilating* 37:32; July, 1940).

Electric Radiant Heat vs. Steam Convection (*Electrical World* 114:395-6; August 10, 1940).

Chapter 45

RADIANT HEATING

Physical and Physiological Factors, Control of Heat Losses, Rate of Heat Production, British Equivalent Temperature, Application Methods, Calculation Principles, Mean Radiant Temperature, Measurement and Control of Radiant Heating

FOR health and comfort, it is necessary for the rate of heat loss from the human body to be controlled by the aggregate effect of the conditions surrounding the body, so that the physiological reactions result in a feeling of comfort. No heating system serves the purpose of adding heat to the individual, but only reduces the net rate at which the body loses heat in cold weather by radiation, convection and evaporation. In convection methods of heating, the medium serves to maintain such an air temperature as will give comfort under existing conditions of humidity and of surrounding surface temperatures. The object of radiant heating, on the other hand, is to maintain an average temperature of the surrounding surfaces which will prevent too much heat loss from the human body by radiation, and thereby give comfort without needlessly heating the air. The difference between convection heating and radiant heating is therefore partly physical and partly physiological.

On a cool spring day while standing in the sunshine, one may feel perfectly comfortable but, when a cloud passes over the sun, one will instantly feel much cooler. A shielded thermometer will show no immediate reduction in air temperature, so that one actually feels a cooling effect which an ordinary thermometer cannot register. This is because light and heat waves travel at the same speed and are both interrupted by the cloud, or other shield. This proves that heat rays affect the comfort of the body more quickly and more definitely than does air temperature.

It also proves that an ordinary thermometer registering the temperature of the air is not a criterion of comfort conditions. Healthful comfort requires that heat shall escape from the body at the same rate as it is generated by the oxidation of food in the body, and in a manner suitable to physiological requirements.

Furthermore, the ambient conditions will often cause changes both in the rate of heat generation in the body, and in the operation of the several methods by which the body loses heat. The feeling of heat or cold results not only from the rate at which the body loses heat, but also from the manner in which the heat is abstracted from the body, and the ease with which the body's heat regulating mechanisms can operate. If the con-

ditions of the environment and the state of the body are not perfectly correlated, a person is vaguely conscious of a strain in the thermostatic body mechanism.

CONTROL OF HEAT LOSSES

Heat is transferred from any warm dry surface to cooler surroundings principally by convection and by radiation; the total loss is substantially the sum of these two. Where the surface is moist, as with the human body, heat is also lost through evaporation from both the body surface and the respiratory tract.

The rate of heat loss by convection depends upon the average temperature difference between the surface of the body and the surrounding air, the shape and size of the body, and the rate of air motion over the body.

The rate of heat loss by radiation depends upon the exposed surface area of the body, and upon the difference between the mean surface temperature of the body and the mean surface temperature of the surrounding walls or other objects. This latter temperature is called the mean radiant temperature (MRT).

Because these two types of heat loss supplement each other, a required rate of total heat loss can result either from a relatively low air temperature and a relatively high MRT, or vice versa. It must be clearly understood, however, that while some conditions stimulate the production of heat in the body, others merely dissipate the heat without controlling the generation of heat.

A heating installation should provide comfort for those individuals doing the least physical work, without causing undesirable changes either in the rate of heat generation, or in the body's heat regulating mechanism.

Rate of Heat Production

The normal rate of heat production in an average sized sedentary individual is about 400 Btu per hour. The heat production for persons subjected to various rates of activity is given in Chapter 2. When considering radiant heating, one must study separately the evaporation, radiation and convection losses. The human body is of complicated shape, and radiation takes place freely only from the exposed outer surfaces; there are considerable portions of the body such as the legs, arms, lower part of head, etc., which radiate most of their heat to other portions. It is necessary to determine the equivalent surface of the body from which heat is radiated and a similar value for convection. The total surface may be assumed as approximately 19.5 sq ft for convection and 15.5 sq ft for radiation, in an average sized individual.

The loss by evaporation and respiration depends on the temperature and area of the moist surfaces (outside and respiratory) of the body, the air temperature, air movement and humidity. In air at a temperature of 70 F, this loss for a sedentary individual of average size will be approximately 90 Btu per hour; and at 60 F, about 70 Btu per hour. These values are relative, because the total will vary materially with change of position, bodily activity, age, sex, race, etc.

The balance of the heat generated in the average human body (approxi-

mately 300 to 320 Btu per hour at about 70 F room temperature) is the approximate amount of heat given off by radiation and convection. It is difficult to determine the exact proportions of these two; but it appears that if the body losses are about 190 Btu per hour by radiation (or 12.25 Btu per hour per square foot of radiating body surface), the greatest comfort will result. This leaves about 120 Btu per hour to be lost by convection (or 6.01 Btu per hour per square foot of convecting body surface).

The mean surface temperature of the human body, including the whole area not only of exposed skin but also of clothing and hair, has been estimated variously at from 75 F (particularly in England) up to as high as 83 F (in America). It is, however, conceded that further research and experience will be needed to finally derive the most suitable value for the American climate. The final figures will vary with sex, age, clothing, etc., but will probably come between these extremes. From installations already in use in America an average surface temperature of 80 F appears to be more nearly correct.

The mean surface temperature of an inert body, which will cause given rates of heat loss by radiation and by convection in a uniform environment, having a given air temperature and a given mean wall temperature, may be calculated from fundamental equations[1] for radiation and natural convection, with substitution of comparable cylinders for the irregular human body.

$$q_r = 0.1730\, e \left[\left(\frac{T_s}{100}\right)^4 - \left(\frac{T_w}{100}\right)^4 \right] \tag{1}$$

$$q_c = 1.235 \left(\frac{1}{D}\right)^{0.2} \times \left(\frac{1}{T_m}\right)^{0.181} \times \left(T_s - T_a\right)^{1.266} \tag{2}$$

where

q_r = heat loss by radiation, Btu per square foot per hour.
q_c = heat loss by convection, Btu per square foot per hour.
T_s = absolute temperature of the body surface, degrees Fahrenheit.
T_w = absolute temperature of the walls, degrees Fahrenheit.
T_a = absolute temperature of the air, degrees Fahrenheit.
$T_m = \dfrac{T_s + T_a}{2}$
D = diameter of cylinder, inches.
e = the ratio of actual emission to black body emission.

If it is assumed that an average adult has a height of 5 ft 8 in. a body surface of 19.5 sq ft for convection, and 15.5 sq ft for radiation, an equivalent effect can be worked out for two cylinders, 5 ft 8 in. high by 13.15 in. diameter and 10.45 in. diameter, respectively. However, while the effects on a cylinder (of a particular size and shape) may be used to estimate average similar effects on the human body, it should be remembered that the heat loss from the body varies greatly. Every movement alters not only its shape, but also the velocity of the air passing over it and the surface exposed to radiation. This fact renders the results of any such computation only approximate.

[1]Surface Heat Transmission, by R. H. Heilman (*A.S.M.E. Transations, Fuels and Steam Power Section*, Vol. 51, No. 22, September-December, 1929).

BRITISH EQUIVALENT TEMPERATURE

The British Equivalent Temperature (BET) is the mean temperature of the entire environment which is effective in controlling the rate of sensible heat loss from a *black body* in still air when this body has a surface temperature equal to that of the human body, and a size comparable to the human body. The BET is, therefore, a function of both the air temperature and the mean radiant temperature of the surrounding objects. Its numercial value in a uniform environment (walls and air at the same temperature) is equal to the temperature of the walls and air. In a non-uniform environment (walls and air at different temperature), the BET for America is at present considered to be equivalent to that of a uniform environment in which a body with an 80 F surface temperature will lose sensible heat at the same rate as in the given non-uniform environment. As originally defined (in England) the BET was based on an average body surface temperature of 75 F, while 80 F seems to be more nearly conforming with American conditions. The most suitable temperature to assume will depend in part on the clothes worn by the individual. This explains why ladies in evening dress require a higher BET for comfort, than a man having only hands and head uncovered. The higher the BET, the less the heat loss from the body, as the rate of heat loss in still air is approximately proportional to the difference between the BET and the mean body surface temperature.

If the BET were 80 F, there could be no sensible heat loss from a surface at that temperature; so the temperature of a normal body surface would have to rise to a point where the heat generated in the tissues could be dissipated. Broadly speaking, it may be stated that with a BET of about 65 F to 70 F, the sensible heat losses from the assumed average individual will approximate those stated on page 800.

APPLICATION METHODS

There are several methods of applying radiant heating, as follows:

1. *By warming the interior surface of the building.* Pipe coils are embedded in the concrete or plaster of the walls or ceilings, the heating medium being hot water circulating through the pipe coils. These coils are generally constructed of small pipe ½ or ¾ in. I.D. and spaced about 6 in. to 9 in. apart (Fig. 1). This has the effect of warming the entire concrete or plaster surface in which the pipes are embedded. Since the temperature of the heating medium should never exceed about 130 F (due to the possibility of cracking the plaster) the area of the warmed surface must be sufficient to supply the requisite quantity of heat at this low temperature. When carefully designed, this method produces very comfortable results and great operating economy, but offers some slight obstacles when alterations or additions to the building are desirable. Normally the hot water circulation is maintained by means of a circulating pump and facilities have to be provided to eliminate all air at the top of the system. All coils and circulating pipes are welded together and tested after erection to a hydraulic pressure of 300 lb per square inch.

2. *By placing hot water or steam pipes under the floor.* With this arrangement the whole floor surface of a room is raised to a temperature sufficient to give comfortable conditions. This method is recommended for schools and hospitals where large quantities of outside air are desirable (Fig. 2). In some cases special floors are constructed in sections so that a whole floor can be lifted to examine the pipes installed under the floor. The floor surface may be of concrete, wood blocks, marble or any other material unaffected by heat. Pipes under the floor may be larger than those embedded in the floor or in the plaster walls and ceilings.

3. *By circulating warm air through shallow ducts under the floor.* In this design the entire floor surface of a room is heated as in Fig. 3. This method was used 2000 years ago in many parts of the Roman Empire. While this method is more expensive in construction, it is effective and quite suitable for cathedrals and large public buildings. To provide a uniform floor temperature, one should give special consideration to the design of the air ducts so that equal heat distribution is obtained.

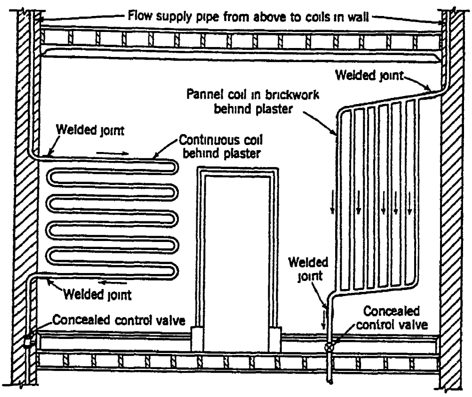


FIG. 1. PIPE COILS LOCATED IN INTERIOR WALL SURFACES

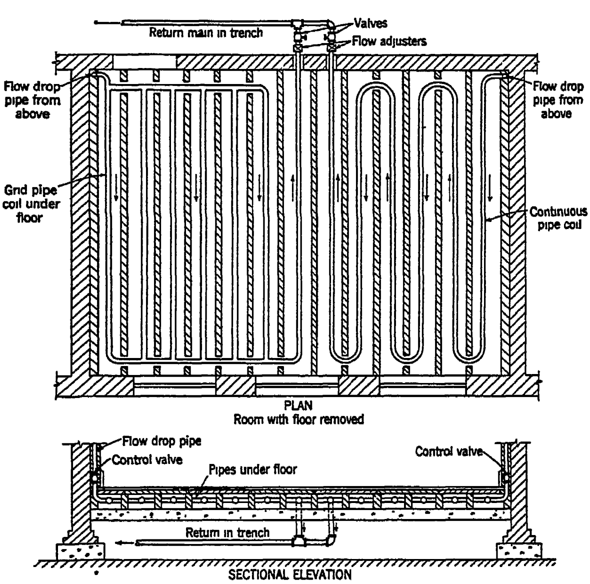


FIG. 2. ARRANGEMENT OF CONTINUOUS PIPE COIL IN FLOOR CONSTRUCTION

4. *By attaching separate heated metal plates or panels to the interior surfaces.* These plates or panels are placed either in an insulated recess so that the surface of the panel is flush with the surface of the walls or ceilings, or they may be secured to the face of the wall. They may be covered with wood veneers and decorated to harmonize with other parts of the room, or they may be cast into panels to imitate oak or other wood designs.

With flat plate panels it is common practice to use a frame of plaster, wood, metal or composition to allow for expansion. These plates may be heated with either hot water or steam and connected as an ordinary radiator system (Figs. 4 and 5).

5. *By electric heated metal plates or panels.* These plates or panels are either placed in insulated recesses of walls or ceilings or fastened to the construction, as found desirable.

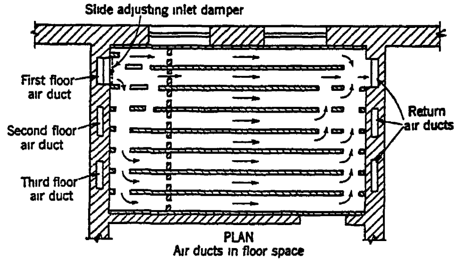


FIG. 3. DIAGRAM OF AIR DUCTS FOR FLOOR HEATING

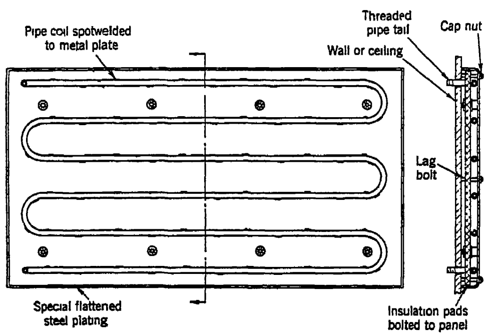


FIG. 4. PILLAR TYPE RADIANT HEAT PANEL

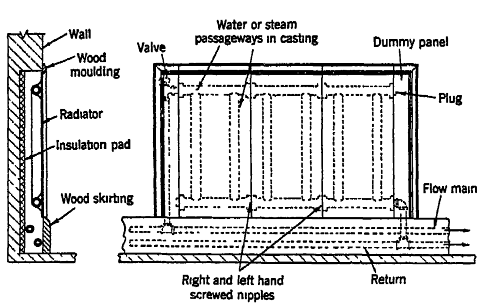


FIG. 5. FLAT TYPE PANEL INSTALLED IN WALL RECESS

They should not have a surface temperature much above 200 F; some have a much higher surface temperature but a lower temperature gives a more comfortable condition and is more efficient.

6. *By electrically heated tapestry mounted on screens and on the wall.* For this purpose the screen is woven with an electric continuous conductor. Such screens are useful to

plug in at any position for emergency local heating without taking care of a large room or office.

Note. If all of a heating panel is installed at one end of a large room there may be a marked difference between the BET on the two sides of the body. It is usually desirable, therefore, that the heat be distributed at different parts of the walls and ceilings so that no uncomfortable effect will be felt from unequal heating.

CALCULATION PRINCIPLES

The calculations for radiant heating are entirely different from those for convective heating. The purpose of the latter is to determine and compensate for the rate of heat loss from the room, when the air temperature is maintained at the desired conditions. Radiant heating, however,

TABLE 1. TOTAL RADIATION TO SURROUNDINGS AT ABSOLUTE ZERO[a]

Body or Mean Radiant Temperature Deg Fahr	Radiation in Btu per square foot per hour emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor e				Body or Mean Radiant Temperature Deg Fahr	Radiation in Btu per square foot per hour emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor e			
	e 1.00	e 0.95	e 0.90	e 0.80		e 1.00	e 0.95	e 0.90	e 0.80
30	99.3	94.3	89.4	79.4	71	136.5	129.6	122.9	109.3
35	103.5	98.3	93.2	82.8	72	137.4	130.5	123.6	109.9
40	107.6	102.4	96.8	86.1	73	138.4	131.5	124.5	110.6
45	112.1	106.5	100.9	89.7	74	139.6	132.6	125.6	111.7
46	112.9	107.3	101.6	90.4	75	141.0	133.9	126.9	112.8
47	113.9	108.2	102.5	91.1	80	146.6	139.4	132.0	117.4
48	114.8	109.1	103.4	91.9	85	152.3	144.6	137.1	121.9
49	115.6	109.9	104.1	92.4	90	157.9	149.9	142.1	126.4
50	116.5	110.6	104.9	93.2	100	169.6	161.1	152.6	135.7
51	117.5	111.6	105.8	94.0	110	181.6	172.5	163.5	145.4
52	118.4	112.5	106.5	94.7	120	194.8	185.0	175.4	155.9
53	119.4	113.4	107.4	95.5	130	210.1	199.6	189.1	168.1
54	120.2	114.2	108.2	96.2	140	223.2	212.1	201.0	178.5
55	121.1	115.1	109.0	96.9	150	237.1	225.2	213.5	189.7
56	122.1	116.0	109.9	97.7	160	251.1	238.8	226.0	201.0
57	123.1	117.0	110.9	98.5	170	270.5	257.0	243.5	216.4
58	124.0	117.8	111.6	99.2	180	288.0	273.8	259.1	230.4
59	124.9	118.6	112.4	99.9	190	306.5	291.0	275.8	245.1
60	125.8	119.5	113.4	100.7	200	325.2	309.0	292.8	260.3
61	126 6	120.3	114.0	101.4	210	348.0	330.6	313.1	278.4
62	127.7	121.4	114.9	102.2	220	371.5	353.0	334.4	297.1
63	128.6	122.2	115.8	102.9	250	437.8	415.9	394.0	350.2
64	129.6	123.1	116.7	103.7	300	575.0	546.1	517.5	460.0
65	130.5	124.0	117.5	104.4	350	740.0	703.0	666.0	592.0
66	131.6	125.0	118.4	105.4	400	942.1	895.0	847.5	753.5
67	132.5	125.9	119.3	106.0	450	1176.0	1117.0	1059.0	941.0
68	133.5	126.8	120.1	106.8	500	1464.0	1390.0	1318.0	1171.0
69	134.5	127.8	121.1	107.6	550	1791.0	1701.0	1613.0	1434.0
70	135.5	128.8	121.9	108.4	600	2405.0	2284.0	2165.0	1925.0

[a]These factors are calculated from the formula

$$q = e\left(\frac{0.1730 \times T^4}{100,000,000}\right)$$

where

q = total radiation, Btu per square foot per hour.
e = emissivity
T = absolute temperature, degrees Fahrenheit.

involves the regulation of the rate of heat loss from the human body in its several forms.

The first step in the calculations for radiant heating of a given room is to determine the desired MRT; the second, to decide on the location of the heated surfaces; the third, to establish the temperature at which the heating surface shall operate; the fourth, to compute the size of the heating surfaces required to produce this MRT; the fifth, to calculate the actual heat loss from the room and to provide, if necessary, any additional convected heat beyond that given off by the radiant surfaces for the required number of air changes. If humidification is required, this must be considered similarly to a conventional air conditioning system, except that the air temperature of the room will be much lower and will therefore require less moisture.

Mean Radiant Temperature

If the entire interior surface of a room were at the same temperature, this would be the MRT. Such a condition seldom exists, because in different parts of a room, some surfaces are exposed to the outer air while others are adjacent to heated rooms. The actual surface temperature varies with the construction and exposure of different sides of the enclosures. It is therefore necessary to calculate the thermal mean of these interior surface temperatures.

This is not the same as the arithmetic average of the various actual surface temperatures, but the radiant temperature which corresponds to the average of the several rates of heat emission (Btu per square foot) from the several surfaces. The emission at any given surface temperature, for any stated emissivity factor can be obtained directly from Table 1, while the emissivity factors for many materials may be found in Table 9 of Chapter 3. For example, from Table 1 it can be determined that if the emissivity of the surface is 0.90 then 1 sq ft of surface at 50 F will emit 104.9 Btu per square foot per hour to surroundings at absolute zero.

Such a determination of the amount of radiant heating surface needed in a room (to maintain a desired MRT) requires knowledge of the type of heating, and of the temperatures of the unheated surfaces. The latter can be estimated from Fig. 6 which is based on an inside air temperature of 70 F. For other air temperatures the inside surface temperature will be proportional to the overall temperature difference. For example if the surface temperature is 60 F based on an inside air temperature of 70 F and an outside temperature of zero, the surface temperature for 65 F air will be 65 ÷ 70 times 60 or 55.7 F. There will be some variation in surface temperature with emissivity, but, except in the case of reflective materials this may be neglected, as the variation due to ordinary building surfaces will be small.

Detailed Computation Method

Assuming the mean surface temperature of the exposed part of the human body and clothing to be 80 F and the emissivity factor to be 0.95, from Table 1 it can be determined that the body surface will give off 139.4 Btu per square foot per hour to absolute zero surroundings. Since the

average human body releases approximately 12.25 Btu per square foot per hour by radiation, the mean radiant emission from the surroundings must be 127.15 Btu per hour with an average emissivity factor of 0.93 which requires an MRT of approximately 71 F. If the body is covered less so that the mean surface temperature of the body is 85 F with an emissivity factor of 0.95, the correct MRT for the room should be 74 F. Consequently, for baths and similar rooms the MRT should be slightly higher than for offices, etc. The mean radiant emission from walls, etc., to give this desired rate can be determined from Table 1. Multiplying by the total surrounding area will give the desired total radiant heat effect. Therefore the MRT for an ordinary living room, office, or similar room to give comfort conditions is 71 F.

The location of the surfaces is generally decided according to the type of building and its use. For high ceilings it is advisable to select floor

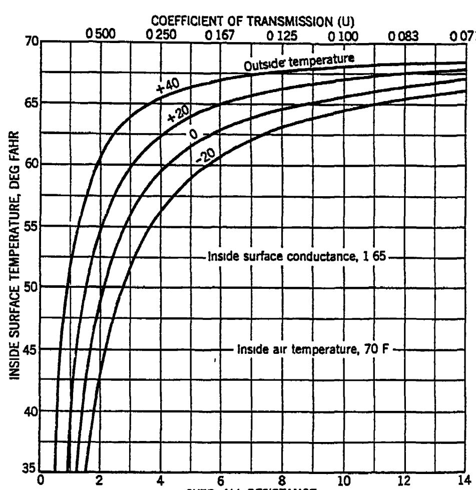


FIG. 6. CURVES FOR ESTIMATING INSIDE SURFACE TEMPERATURES OF OUTSIDE WALLS

heating or install heated panels in the walls at low level. For exposed rooms it may be necessary to have some wall or ceiling panels in addition to floor heating.

The temperature of the surface is controlled somewhat by the location. If the floor is chosen, then hot water pipes should be used as the medium; and the surface temperature should never be more than 85 F unless border heating is used. The latter comprises strips of heated surfaces where occupants will not usually rest their feet, such as portions of the floor adjacent to walls or windows, aisles of churches, halls, etc. If iron panels are used on side walls, etc., a surface temperature up to 160 F may be used with hot water as the heating medium. Vapor or low pressure steam may also be used with a maximum surface temperature of 180 F. For ceiling or other plaster heating, hot water pipes should be used with a maximum water temperature of 130 F giving a surface temperature of about 115 F.

TABLE 2. HIGHEST SAFE SURFACE TEMPERATURES FOR HEATING PANEL

TYPE OF PANEL	SURFACE TEMPERATURE DEG F
Plastered Ceiling (Pipes Embedded)	115
Plastered Walls (Pipes Embedded)	120
Floor, Any Method	85
Floor, Border and Aisles	120
Iron, Hot Water Medium[a]	160
Iron, Steam Vapor[a]	180
Electrically Heated Panels[a]	200

[a]Low surface temperature radiation is recommended regardless of the heating medium employed.

The area in square feet of each type or different surface temperature, horizontal or vertical, is multiplied by the emission value corresponding to its actual surface temperature. These products are added together to give the total radiant heat effect inside the room from all surfaces.

The difference between the desired and the actual total radiant emission represents the additional heating effect which must be supplied by the hot surfaces to be installed. The temperature of the proposed hot surface must then be selected from Table 2, and its emission per square foot at that temperature determined from Table 1. The difference between this emission and that of the unheated surface replaced by the panel is divided into the total amount of additional heat needed, and the quotient will be the area of the required heating surfaces.

It is evident that this method of calculation depends for its accuracy on a correct estimate of the ultimate surface temperatures naturally attained by the actual wall, window, ceiling and floor surfaces.

Example 1. The surface areas, assumed temperatures, and emissions for a room having a volume of 5760 cu ft are given in Table 3. The figures for temperatures are based on a room air temperature of 65 F and an outside temperature of zero. The heat emissions in Btu per square foot per hour are taken from Table 1.

The values for glass will depend on whether or not shades and curtains are provided. In offices and other similar rooms the whole glass surface will invariably be fully exposed to the occupants; whereas in a residence, curtains may cover part or the whole window, thus increasing the natural MRT and reducing the heat loss from the human body.

The mean radiant emission of the room in Example 1 is 223,912 $\div$ 2016 = 111.0 Btu per square foot per hour which from Table 1 corresponds to an MRT of 54 F for an average emissivity of 0.92.

TABLE 3. SURFACE AREA, TEMPERATURES AND EMISSIONS FOR A ROOM OF 5760 CU FT

	AREA SQ FT	ASSUMED SURFACE TEMPERATURE DEG F	EMISSIVITY	HEAT EMISSION BTU PER SQ FT PER HOUR	TOTAL HEAT EMISSION FROM AREA BTU PER HOUR
External Wall	297	50	0.95	110.6	32,850
Glass	279	40	0.80	86.1	24,022
Inner Wall	480	60	0.95	119.5	57,360
Ceiling	480	60	0.95	119.5	57,360
Floor	480	55	0.90	109.0	52,320
Total	2016				223,912

In order to determine the amount of radiating surface necessary to maintain the MRT at 71 F, assume ceiling panels have a surface temperature of 115 F, which is approximately the mean temperature for pipes embedded in the plaster with circulating water at 130 F as given in Table 2. With a ceiling surface temperature of 115 F and an emissivity of 0.95, from Table 1 the emission is 179 Btu per square foot per hour.

The difference between 179 and 119.5 used in Table 3 is the extra Btu emitted by warmed ceiling, per square foot, and therefore 480 sq ft multiplied by (179 − 119.5) = 28,560 Btu per hour.

Therefore the total radiant heat emission with ceiling heated will be 223,912 + 28,560 = 252,472 Btu per hour or an average emission of 252,472 ÷ 2016 = 125.21 Btu per square foot per hour. With an average emissivity of 0.92 it is found from Table 1 that

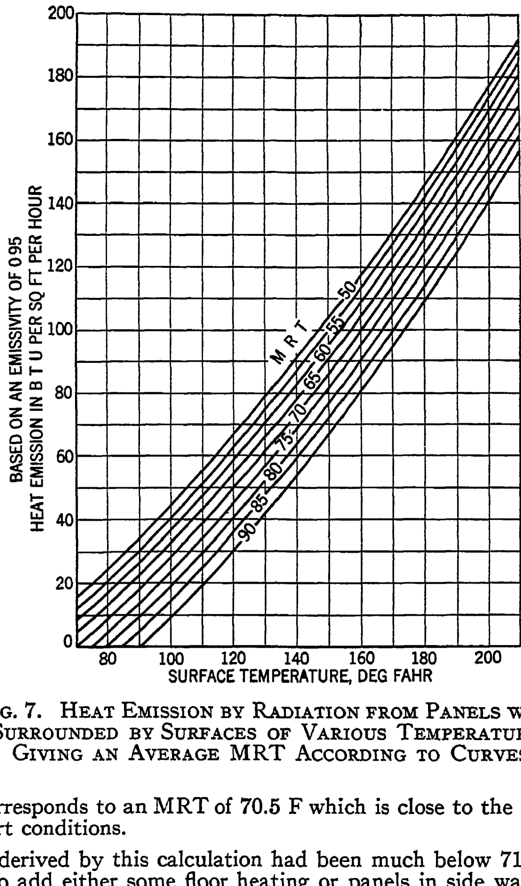


FIG. 7. HEAT EMISSION BY RADIATION FROM PANELS WHEN SURROUNDED BY SURFACES OF VARIOUS TEMPERATURES GIVING AN AVERAGE MRT ACCORDING TO CURVES

this emission corresponds to an MRT of 70.5 F which is close to the 71 F necessary for optimum comfort conditions.

If the MRT derived by this calculation had been much below 71 F, it would have been desirable to add either some floor heating or panels in side walls. On the other hand, metal panels might have been installed in the ceiling and operated at a higher temperature.

Similar calculations may be used for any kind and surface temperature of panel selected by using the difference between the actual emission from the heated surface as obtained from Table 1 and the emission for the unheated surface as used in Table 3.

Such calculations may be simplified, by preparing tables showing at the usual temperatures the area of hot surface required to bring each square foot of actual wall or other surface up to one or more desired standard MRT's.

Assuming heat losses from the room in Table 3:

Area Square Foot		Btu per Hour
297 External Wall	(U = 0.25) 297 × 0.25 × 65 F =	4,820
279 Glass	(U = 1.13) 279 × 1.13 × 65 F =	20,476
480 Inner Wall	(next to heated room)	
480 Ceiling	(heated surface)	
480 Floor	(U = 0.10) 480 × 0.10 × 65 F =	2,490
5760 cu ft Infiltration	1½ Changes per hour 0 F to 65 F =	10,200
Total		37,986 Btu per hour

Therefore the total number of Btu per hour to be supplied to the room will be 37,986 of which 10,200 will be for ventilation.

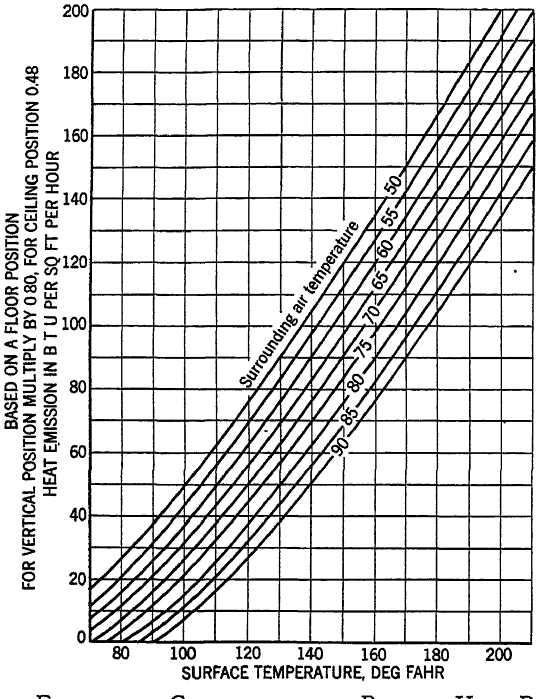


Fig. 8. Heat Emission by Convection from Radiant Heat Panels with Still Air at Various Temperatures

The ceiling with a surface temperature of 115 F and a surrounding MRT of the unheated surfaces of 54 F will emit 56 Btu per hour by radiation (see Fig. 7), and with an air temperature of 65 F will emit (50 × 0.48) = 24 Btu per hour by convection (see Fig. 8).

480 sq ft × 56 = Total by radiation = 26,880 Btu per hour
480 sq ft × 24 = Total by convection = 11,520 Btu per hour

Total = 38,400 Btu per hour

The difference between 38,400 and 37,986 Btu per hour gives a margin of 414 Btu per hour.

Should more heat have been required for ventilation, this could have been supplied by adding wall panels or by introducing direct ventilation with air introduced to the room at the correct temperature.

MEASUREMENT OF RADIANT HEATING

Convection heating, intended to maintain a given air temperature, is best measured by thermometric methods, which indicate the air temperature, and not the rate of heat loss from the human body. Radiant heating, on the other hand, aims to control this rate of heat loss and can be measured only by calorimetric methods.

The apparatus for this purpose consists essentially of a cylinder, maintained at the accepted mean surface temperature of the human body, together with an accurate (usually electrical) measuring of the varying rate of heat supply required to maintain this exact temperature. This instrument, the *eupatheoscope*, is readily adapted to function like a thermostat so as to turn heat on or off, when the desired temperature of 80 F, or any other predetermined surface temperature of the cylinder, decreases or increases as a result of changes in the BET.

For testing work, the *globe thermometer* is a useful instrument. It consists of an ordinary mercury thermometer, with its bulb placed in the center of a sphere from 6 to 9 in. in diameter, usually made of thin copper and painted black and sometimes covered with cloth. The temperature recorded by therometer with its bulb in the center of the sphere is termed the *radiation-convection temperature.*

CONTROL OF RADIANT HEATING

The effectiveness of any control depends on the time lag of the system. With warm air passing through floor ducts the time lag is usually too long for a room thermostat, and with massive brickwork and masonry it is too long for any control except a time control operated by outside conditions. Hot water pipes embedded in the plaster of ceiling and walls or in the concrete of a floor may be effectively controlled by a suitable instrument which will control the temperature of the water circulating in the system by the outside conditions. Metal panels installed on the ceiling or side walls may be controlled by the outside weather conditions or by a room thermostat.

One type of room instrument consists of a blackened copper sphere of 6 or 8 in. in diameter, in which a cylindrical sump contains a volatile liquid. A small electric heating coil creates in the sphere a vapor pressure which remains constant as long as the total heat loss from the sphere is at the desired rate. If the BET becomes too high for comfort, a greater vapor pressure results from the smaller heat loss from the sphere. This acts on a diaphragm and turns down the supply of heat to the room. With too low a BET the reverse action occurs. A similar instrument which has an electric heating element for warming the air inside the sphere and thermostat operated switch is also used for controlling room conditions.

FUEL CONSUMPTION

Because of the lower air temperature necessary with radiant heating, there is a possible saving in fuel consumption. This saving depends very largely on the method employed to heat the surfaces and the provision made to prevent heat passing from the coils to the earth or outside air.

REFERENCES

Panel Warming, by L. J. Fowler (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 287).

Room Warming by Radiation, by A. H. Barker (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 331).

What Will be the Future Development of Heating and Air Conditioning, by W. H. Carrier (*Heating, Piping and Air Conditioning*, January, 1933, p. 16).

American Practice in Panel Heating, by L. L. Munier (*Heating, Piping and Air Conditioning*, June, 1937, p. 424).

Radiant Heating and Cooling, by F. E. Giesecke (*Heating, Piping and Air Conditioning*, June, July, August, September and October, 1940).

Panel Heating in the British Embassy, by Alfred L. Jaros, Jr. and Richard A. Wolff (*Heating and Ventilating*, May and June, 1930).

Calculations for Radiant Heating, by T. Napier Adlam (*Heating and Ventilating*, October, 1931).

Notes on Electric Warming with Special Reference to Low Temperature Panel Systems, by R. Grierson (*Proceedings of The Institution of Heating and Ventilating Engineers*, London, Vol. 28, 1929).

Notes on the Theory of Radiant Heating, by C. G. Heys Hallett (*Proceedings of The Institution of Heating and Ventilating Engineers*, London, Vol. 29, 1930).

Radiant Heat, by A. F. Dufton (*Proceedings of The Institution of Heating and Ventilating Engineers*, London, Vol. 30, 1931).

Panel Heating, by C. M. Oates (*Proceedings of The Institution of Heating and Ventilating Engineers*, London, Vol. 30, 1931).

Principles of Calculation of Low Temperature Radiant Heating, by A. H. Barker (*Proceedings of The Institution of Heating and Ventilating Engineers*, London, Vol. 30, 1931).

Radiant Heat, by A. F. Dufton (*Proceedings of The Institution of Heating and Ventilating Engineers*, London, Vol. 31, 1932).

Chapter 46

WATER SUPPLY PIPING AND WATER HEATING

Maximum Flow, Factor of Usage, Water Pressure, Pipe Material, Allowance for Fittings, Sizing Up-Feed Systems, Sizing Down-Feed Systems, Hot Water Supply, Storage Capacity and Heating Load, Methods of Heating Water, Computing Grate and Coil Surface Area, Controls

THIS chapter deals only with problems of providing adequate facilities for delivering cold and heated water for domestic purposes in buildings.

The intensity of occupation and the amount of water used or wasted per occupant in buildings are subject to wide variation and present a complicated problem despite the rather exact knowledge available concerning hydraulic behavior as to friction, incrustation, heat transfer, etc. The designer's most serious problem is to provide the piping and the water heating and storage facilities of sufficient capacity to meet the peak demand without wasteful excess in investment or radiation costs.

For example, in two office buildings of similar type the metered water consumption has shown as much as 300 per cent difference per outlet. The rate of use in such buildings fluctuates tremendously with the hour of day, and therefore the design must be the result of adequate consideration of this and of the number of occupants, the number of water outlets, and the type of building occupancy.

Residences present a comparatively easy problem since long established custom has evolved reliable factors for water consumption per installed fixture. Tests have been made repeatedly of the amount of water required by standard fixtures in normal use with water at ordinary pressures so that this information gives a fairly correct basis of design.

For the purpose of this chapter the following terms will be used and should be clearly distinguished from one another:

Maximum Flow: The flow which would occur if the outlets on all fixtures were opened simultaneously. This condition is seldom obtained in actual practice except in cases of gang showers controlled from one common valve.

Probable Flow: The maximum flow which any pipe is likely to carry under the peak conditions. This is the most important amount to be considered in pipe sizing.

Average Flow: The flow likely to be required through the line under normal conditions.

It is evident that any pipe size adequate to take care of the *probable flow* will also be more than ample to take care of the *average flow*, and hence the latter has no bearing on the pipe size.

MAXIMUM FLOW

An estimate of maximum flow for various fixtures regardless of type of building with the water at about 35 lb pressure is given in Table 1.

To obtain the probable flow from Table 1, it is necessary to multiply the maximum flow by a factor of usage, and this factor varies with the

Table 1. Approximate Maximum Flow from Fixtures under Normal Water Pressures

Fixtures	Cold Water (Gallons per Minute)	Hot Water (Gallons per Minute)
Water-closets, flush valve	45[a]	0
Water-closets, flush tank	10	0
Urinals, flush valve	30[a]	0
Urinals, flush tank	10	0
Urinals, automatic tank	1	0
Urinals, perforated pipe per foot	10	0
Lavatories	3	3
Showers, 4 in. heads, ½ in. inlets	3	3
Showers, 6 in. heads or larger	6	6
Needle bath	30	30
Shampoo spray	1	1
Liver spray	2	2
Manicure table	1½	1½
Baths, tub	5	5
Kitchen sink	4	4
Pantry sink, ordinary	2	2
Pantry sink, large bibb	6	6
Slop sinks	6	6
Wash trays	3	3
Laundry tray	6	6
Garden hose bibb	10	0

[a]Actual tests on water-closet flush valves indicate 40 gpm as the maximum rate of flow with 30 lb pressure at the valve; this would increase to 60 gpm (about 50 per cent) at 90 lb pressure. The 45 gpm has been taken as an average flow; possibly, with very low pressures just sufficient to operate the flush valve, 30 gpm could be allowed with safety. Urinal flush valves would vary proportionately in the same manner.

type of occupancy and with the number of fixtures in the installation. With only two fixtures it is possible that both will at some time be in operation simultaneously. With 200 fixtures, however, it is unlikely that the entire 200 would ever operate at the same time. Consequently, the factor of usage becomes smaller as the number of fixtures becomes greater, all other things being equal.

The maximum flow per fixture for cold water should be totaled independently of that for hot water, and the sum of the two may be used in computing the probable flow through the incoming cold water supply main.

FACTOR OF USAGE

The principal fixture subject to wide variation in water demand is a flush valve closet, though special consideration should also be given to

the demand of shower baths, especially in athletic clubs and in manufacturing plants where the outgoing shifts create a heavy peak.

The curves of Fig. 1 suggest a method of selecting a factor of usage. The curve at the left should be followed for hot water piping and for cold water if the system has gravity tank closets, while the curve to the right allows amply for the influence of flush valve closets.

If the product of the number of fixtures in a building multiplied by the proper values in Table 1 totals 620 gal of water as the maximum flow, using flush tank closets, the factor of usage from Fig. 1 will be about 23 per cent, and the probable flow will be $620 \times 0.23 = 143$ gpm. This is the first item to be determined in the design of a water supply system. In a building using 143 gpm no serious difference in the size of the main supply pipe would be occasioned by use of flush valves, since the factor of usage with the latter would be increased only to about 25 per cent or 155 gpm.

The curves of Fig. 1 are believed conservative for toilet rooms in large office buildings which have early business hour peaks, especially in the men's toilets, but may not be conservative enough for plants such as were mentioned previously where heavy peak demands occur during certain bathing hours. The proper usage percentage for such cases must be a matter of judgment and might properly approach 100 per cent. The average flow will usually be considerably smaller.

Example 1. Assume that in a normal building, such as a residential hotel or an apartment house, there are 50 flush valve water-closets, 50 lavatories, 50 sinks and 50 baths, and that it is desired to determine the probable flow in a line supplying all of these fixtures with both hot and cold water.

Cold Water

50 W. C. x 45 gpm	2250 gpm
50 Lavs. x 3 gpm	150 gpm
50 Sinks x 4 gpm	200 gpm
50 Baths x 5 gpm	250 gpm
Maximum flow	2850 gpm

Fig. 1 shows a factor of usage of 9 per cent.

Probable flow of cold water is 2850×0.09 .... 257 gpm

Hot Water

50 Lavs. x 3 gpm	150 gpm
50 Sinks x 4 gpm	200 gpm
50 Baths x 5 gpm	250 gpm
Maximum flow	600 gpm

Fig. 1 shows a factor of usage of 23 per cent.

Probable flow of hot water is 600×0.23 .... 138 gpm

Total for main supplying cold and hot water $(2850 + 600) \times 0.08$ .... 276 gpm

It should be noted that this is a *rate of flow* or an *instantaneous demand.*

WATER PRESSURE

The usual practice in buildings of moderate height is to place the water supply mains near the basement ceiling, with up-feed risers feeding the various fixtures on the superimposed stories. In tall buildings the pressure due to the weight of the water becomes so great as to limit the service to vertical sections not exceeding about 20 stories in height. Beyond this approximate limit the valves on the lower stories will be noisy and the piping should be extra strong.

For these reasons, the considerations of this chapter are limited to horizontal mains and to risers which serve not more than 20 stories. In taller buildings it is usual to install separate horizontal mains for each superimposed zone, served by extra strong main risers.

The minimum practicable size of piping for any water system is governed by the amount of pressure which can be spared in overcoming resistance to flow of a given volume of water per unit of time. After the approximate amount of water required has been computed, a minimum delivery pressure at the highest fixture may be determined. This will be approximately 15 lb per square inch for flush valve closets or 5 lb per square inch for gravity tank closets. It should be remembered that for every foot in height there will be a hydrostatic loss of approximately 0.433 lb.

The pressure loss through a water meter may be significant as may be seen from Table 2. The pressure losses through filters or other water-conditioning apparatus must also be considered. After evaluating the previously mentioned factors, the total allowable friction loss for the

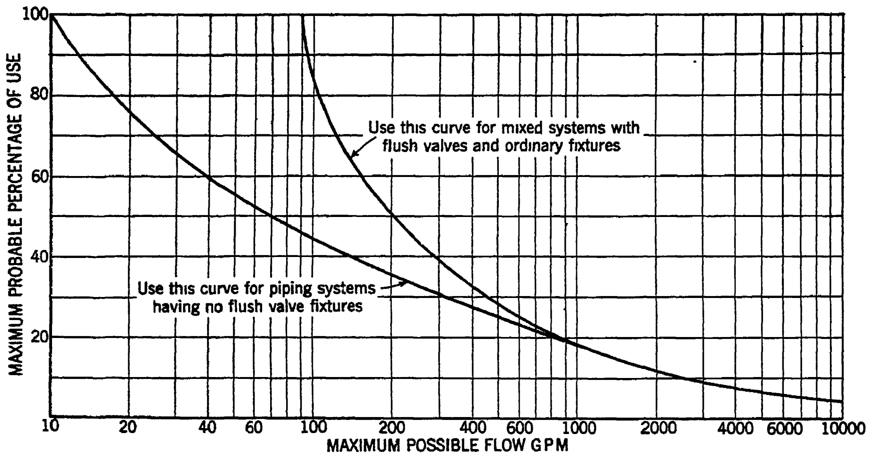


Fig. 1. Chart Showing Relation Between Maximum Flow and Probable Usage

system may be determined by subtracting from the street main pressure, the sum of the following four items:

1. Minimum allowable pressure at top fixture.
2. Meter loss.
3. 0.433 × height in feet from main to top fixture.
4. Loss for filters, softener, etc.

PIPE MATERIAL

The material used in the water piping affects its carrying capacity. For example, copper or brass pipe is not as likely to retain interior incrustation as is ferrous pipe, and galvanized pipe will not rust as quickly as uncoated pipe. Some waters tend to deposit salts, rust, and the like on the interior surfaces of pipes, greatly reducing their capacity. In some cities it has been found necessary to allow for as much as 50 per cent reduction in carrying capacity after 15 years of service.

The data given in this chapter are based on use of galvanized steel piping and on water which is not notoriously inclined to leave deposit.

If the building is in a zone having untreated water known to carry precipitable solids, the pipe sizes should be increased at least one size and no water pipe should be smaller than ¾ in.

TABLE 2. PRESSURE LOSS THROUGH WATER DISC METERS[a]
A. W. W. A. Standards

Rate of Flow Gpm	Approx. Pressure Loss Through Meters, Lb per Sq In Pipe Size (In.)							
	⅝	¾	1	1½	2	3	4	6
5	1.5	0.5	0.2					
10	6.0	2.0	1.0	0.2				
15	14.0	5.0	2.0	0.6	0.2			
20	25.0	9.0	3.5	1.0	0.4			
25		13.5	5.5	1.5	0.6			
30		19.5	8.0	2.0	0.9			
35			11.0	3.0	1.0			
40			14.0	4.0	1.5			
45			18.0	5.0	2.0			
50			22.0	6.0	2.5	0.7		
75				14.0	5.5	1.5		
100				25.0	10.0	2.8	1.0	
125					15.0	4.0	1.5	
150					22.0	6.0	2.2	
175						8.0	3.0	
200						10.4	4.0	1.0
250						16.0	6.0	1.5
300						23.0	9 0	2.2
350							12.0	3.0
400							16.0	4.0
500							25.0	6.5
600								9.0
800								16.0
1000								25.0

MINIMUM SIZE OF SERVICE RECOMMENDED

Rate of Flow Gpm	Approx. Minimum Pipe Size of Service, Main to Meter (In.) Maximum Length (Ft)				
	30	75	100	150	200
1–20	¾	¾	1	1	1
20–30	¾	1	1	1½	1½
30–50	1	1½	1½	1½	1½
50–100	1½	1½	2	2	2
100–150	1½	2	2	2½	2½

SAFE MAXIMUM DELIVERY OF METERS

Meter Size In.	Capacity, Gpm Based on 25 Lb Loss Through Meter
⅝	20
¾	34
1	53
1½	100
2	160
3	315
4	500
6	1000

[a]Pressure loss through compound and current meters is less than shown in table. For exact information consult manufacturers.

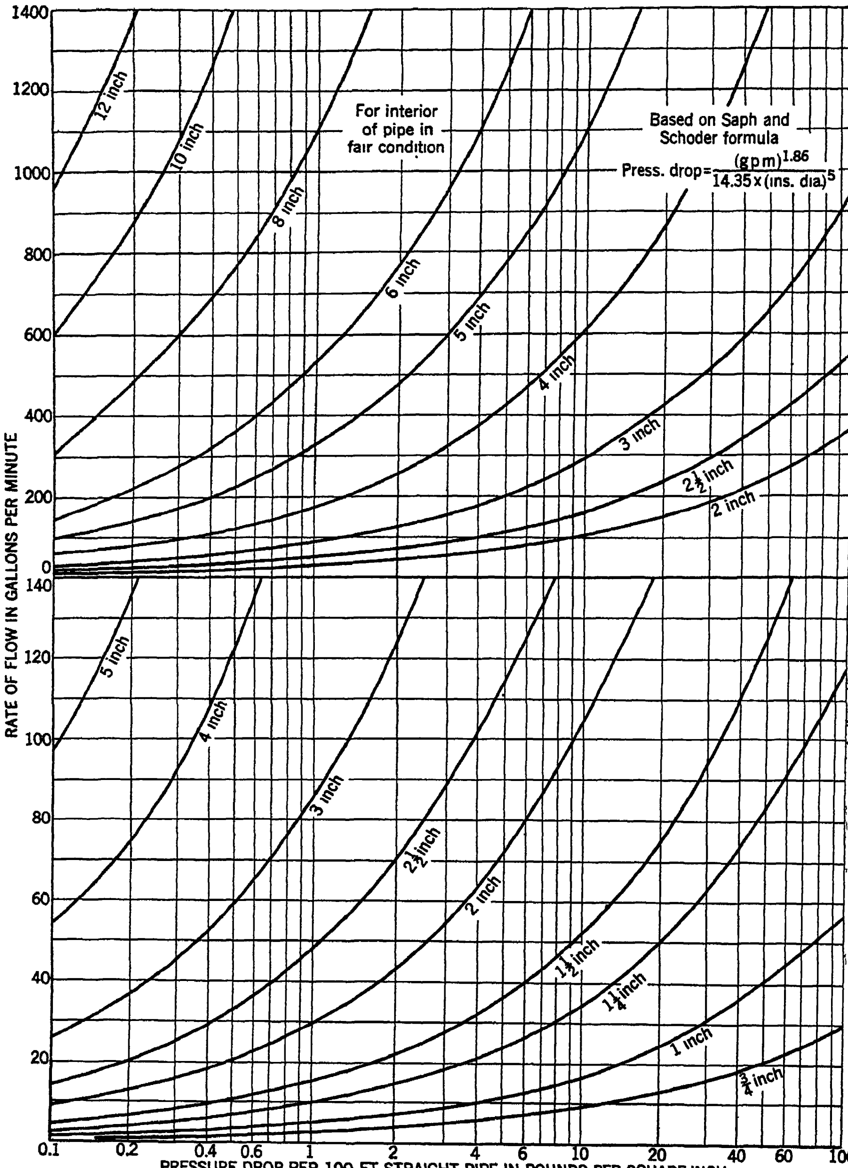


FIG. 2. CHART GIVING PRESSURE DROP FOR VARIOUS RATES OF FLOW OF WATER

ALLOWANCE FOR FITTINGS

Before applying charts for pipe friction, the resistance due to fittings and valves should be evaluated. Table 3 gives this resistance expressed in equivalent feet of straight pipe. To use Table 3, the size of the valve or fitting must be known. Table 3 is therefore of little use in the original design of a system, since the valve and fitting allowances must be made before the pipe size is known. Experience indicates, however, that an average increase of 50 per cent in the length of the longest measured pipe will account for the fittings, and thus a tentative length can be assumed for computing the pressure drop per 100 ft of run. The values of Table 3 should always be used to recheck the exact equivalent length of the pipe after an approximate diameter has been selected and after the number of fittings has been determined. The allowable pressure drop per 100 ft of pipe may be determined by dividing the total allowable drop for the system by the equivalent length of the system in hundreds of feet.

TABLE 3. APPROXIMATE ALLOWANCES FOR FITTINGS AND VALVES IN FEET OF STRAIGHT PIPE

Size of Pipe (Inches)	Type of Fitting or Valve					
	90 Deg Elbow	45 Deg Elbow	Tee in Run of Main	Gate Valve	Globe Valve	Angle Valve
½	2	1	1	1	17	9
¾	2	1	2	1	21	12
1	3	1	2	1	29	15
1¼	4	2	3	1	38	19
1½	5	2	3	1	45	22
2	5	3	3	1	58	28
2½	7	3	5	2	68	34
3	8	4	6	2	82	42
4	11	5	7	2	115	56
5	14	6	9	3	140	70
6	16	8	11	4	160	85

The pressure drop for various rates of water flow for standard size pipes is given in Fig. 2. This chart carries an allowance for reasonable roughness of the interior surface and for the effect of many years of service. The Saph and Schoder formulae have been proved conservative not only by the *American Water Works Association* but also by various tests conducted by the A.S.H.V.E. Committee on Research.

Example 2. Assume a street pressure of 70 lb, the height of the highest fixture 50 ft, the length of the longest run 200 ft, the pressure at the top fixture 15 lb, and the pressure loss through the meter 10 lb. Without knowing the additional length of pipe to be added for the fittings it will be assumed that this is about 100 ft. The surplus pressure which will be available for pressure drop will then be 70 lb − (15 lb + 10 lb + 50 ft × 0.43 lb) = 70 lb − (15 lb + 10 lb + 21.5 lb) = 23.5 lb.

To change this into drop per 100 ft: $\frac{23.5 \text{ lb} \times 100}{200 \text{ ft} + 100 \text{ ft}} = 7.8$ lb per 100 ft.

The pipe may then be sized from the probable flow by selecting a size that does not give a drop in excess of 7.8 lb per 100 ft.

METHOD OF SIZING UP-FEED SYSTEMS

Example 3. A typical layout of cold water lines for a 3-story, nine-family apartment house is shown in Fig. 3. The branch to each apartment supplies 1 lavatory, 1 bath tub, 1 flush valve closet, and 1 kitchen sink. Pressure in the street main is 70 lb per square inch and a minimum pressure of 15 lb per square inch must be maintained on the top floor. Find the sizes of all parts of the system.

Solution: The first step toward the solution of such a problem is the determination of the probable flow in the various parts of the system. In Section *A*, which supplies cold water to a single apartment, the maximum flow would be as follows:

1 water closet	45 gpm
1 lavatory	3 gpm
1 tub	5 gpm
1 sink	4 gpm
	57 gpm

From Fig. 1, the probable usage for 57 gpm maximum flow in a mixed system is 100 per cent. Therefore, Section *A* should be sized for 57 gpm.

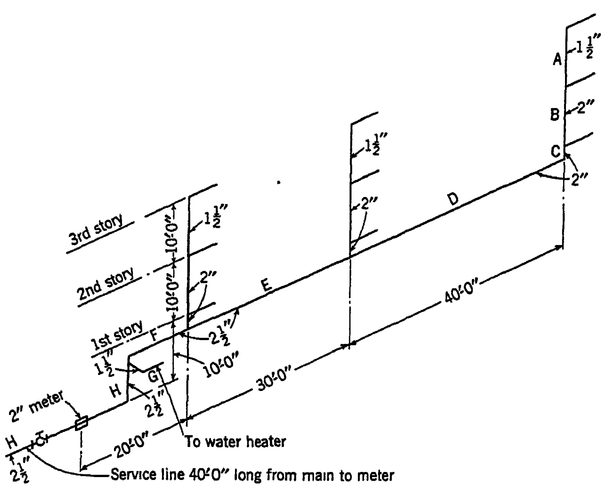


FIG. 3. UP-FEED COLD WATER SYSTEM WITH FLUSH VALVES

Section *B*, which supplies two apartments will have a maximum flow of 2 × 57 gpm = 114 gpm. From Fig. 1, the probable usage for 114 gpm is approximately 75 per cent and the probable flow in Section *B* = 114 × 0.75 = 86 gpm.

Similarly, the probable flow in Section *C* is found to be 98 gpm. Since all risers in this particular example are supplying the same number of fixtures, the probable flow in risers 1 and 2 is the same as determined for riser 3.

To determine the probable flow in Section *E*, add the maximum flow in risers 2 and 3, and multiply the sum by the probable usage for the sum, thus:

(171 + 171) × 0.35 = 120 gpm probable flow in *E*. Similarly, the probable flow in Section *F* is determined.

It should be noted that the probable flow in *E* cannot be determined by adding the probable flow in risers 2 and 3.

Initially it was decided to size only the cold water lines in this example, but it is also necessary to determine the maximum flow in line *G* to the water heater, since this is a

part of the water supplied by the cold water service line. Total hot water requirements are as follows:

9 lavatories	$9 \times 3 = 27$ gpm
9 tubs	$9 \times 5 = 45$ gpm
9 sinks	$9 \times 4 = 36$ gpm
Maximum flow	$= 108$ gpm

The probable flow in all sections of the system are determined as described previously, and tabulated in Table 4.

The next step in the solution is the determination of the allowable pressure drop:

Loss in a 2 in. meter for 149 gpm, from Table 2	= 22 lb per square inch
Hydrostatic head = 30 ft (30×0.43)	= 13 lb per square inch
Pressure at top fixture	= 15 lb per square inch
Total	50 lb per square inch
Allowable pressure loss = 70 − 50 lb	= 20 lb per square inch

To determine the allowable pressure loss per 100 ft of pipe, the longest run to the highest fixture must be used. In Fig. 3 this would be the length to the top fixtures on riser No. 3. The developed length from the meter to the top of riser 3 is 120 ft, and the

TABLE 4. SUMMARY OF RESULTS FOR EXAMPLE 3

Section	Maximum Flow Gpm	Probable Usage Per Cent	Probable Flow Gpm	Allowable Loss Lb per 100 Ft	Pipe Size In.
A	57	100	57	9.1	1½
B	114	75	86	9.1	2
C–D	171	57	98	9.1	2
E	342	36	123	9.1	2½
F	513	28	144	9.1	2½
G	108	43[a]	46	9.1	1½
H	621	24	149	9.1	2½

[a]From curve for fixtures having no flush valves.

equivalent length, allowing 50 per cent for fittings is 180 ft. The service line is 40 ft long, making a total equivalent length of 220 ft from the main to the farthest fixture. Since the service line is usually straight, no allowance has been made for fittings.

The total allowable loss is 20 lb per square inch, and the developed length of piping is 220 ft. Therefore, the allowable loss per 100 ft of pipe is $\frac{20 \times 100}{220} = 9.1$ lb.

Knowing the probable flow in all lines and the allowable loss per 100 ft of pipe, it is possible to determine the pipe sizes from Fig. 2 by reading the pipe size indicated at the intersection of the two known factors. Pipe sizes for all parts of the system are given in Table 4.

Ordinarily the size above the intersection on the chart is selected. However, it is permissible to select a pipe slightly undersize if the next section of the line is oversize. This is illustrated in the sizing of sections *A* and *B*. The pipe size of 1½ in. is slightly small for *A*, but 2 in. is enough oversize for *B*, so that the average loss in the two is less than 9.1 lb per 100 ft.

In this example, all risers have been sized for the same loss per 100 ft of pipe. Where the main is long it is frequently possible to increase the pressure drop per 100 ft of pipe in the risers near the meter, and thus reduce their size. For example, the total friction loss from the meter to the top of riser 1 in Fig. 3, could be as great as the total loss from the meter to the top of riser 3. However, all parts of the main must always be sized to assure sufficient pressure at the last riser. In a small system, such as shown in Fig. 3, no appreciable reduction in pipe sizes can be made by taking advantage of the possibility just described.

PIPE SIZES FOR DOWN-FEED COLD WATER SYSTEM

The risers for down-feed systems may be reduced considerably in size compared with those for up-feed systems because of the 0.43 lb per foot gain in pressure due to increasing hydrostatic head as the lowest story is approached. It has proved practicable to select down-feed riser sizes

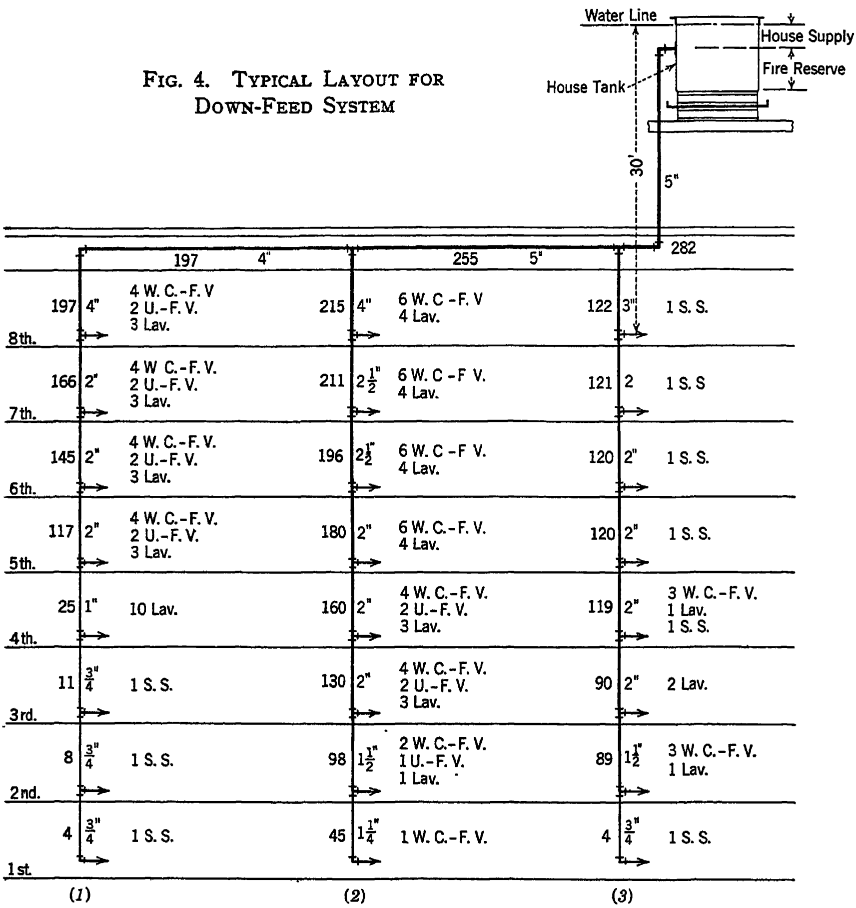


Fig. 4. Typical Layout for Down-Feed System

on the basis of a pressure drop of 30 lb per 100 ft. The 13 lb difference between 43 lb per 100 ft and 30 lb per 100 ft will usually take care of the friction in the fittings.

The overhead mains, however, must be selected conservatively, as the pressure at the top will be low and the pressure drop available for friction will necessarily be small. In nearly all tall buildings the pressure is limited to that due to the hydrostatic head between the house tank and the main, though sometimes this is increased by the use of a pneumatic house tank.

Where flush valves are used on top story closets the minimum practicable difference in elevation between the overhead mains and the bottom of an open type house tank is 20 ft, and flush valves specially adapted to operate on about 7 lb pressure must be used. In many installations gravity tank closets are used on the top story.

Example 4. Suppose an installation has a house tank in which the water line is 20 ft above the level of the top fixtures to be supplied and that the length of run to the farthest fixtures on this level is 400 ft with the pipe fittings adding another 200 ft, making an equivalent length of 600 ft. What would be the size of main coming out of the tank where a probable flow rate of 400 gpm may be expected, of the horizontal main where a probable flow rate of 200 gpm may be expected, and of the riser down to the fixture level where the probable flow rate is approximately 100 gpm?

TABLE 5. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS

(Riser No. 1. Fig. 4)

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXTURE	MAXIMUM GPM ON FLOOR	MAXIMUM GPM ON RISER	USAGE (PER CENT)	PROBABLE FLOW IN RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN.
1st	1 S. S.	4	4	4	100	4	30	¾
2nd	1 S. S.	4	4	8	100	8	30	¾
3rd	1 S. S.	4	4	12	92	11	30	¾
4th	10 Lav.	3	30	42	58	25	30	1
5th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	291	40	117	30	2
6th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	540	27	145	30	2
7th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	789	21	166	30	2
8th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	1038	19	197	2	4

Here the level of the water in the house tank is 20 ft above the faucet of the highest fixture and the gravity pressure will be 0.43 lb $\times$ 20 ft $=$ 8.6 lb and, if a total pressure drop of 1 lb is assumed, the pressure on the farthest fixture under times of peak load will be 8.6 lb $-$ 1 lb $=$ 7.6 lb while the drop per 100 ft of equivalent run will have to be

$$\frac{1 \text{ lb} \times 100}{600} = 0.1667 \text{ lb}.$$

Referring to Fig. 2 it will be noted that where the flow through the main is 400 gpm, an 8-in. pipe would be required; that where the flow is reduced to 200 gpm, a 6-in. pipe would be sufficient; and that where the flow is 100 gpm in the riser branch and riser, a 5-in. size would be correct. Of course these are somewhat excessive flows and the head from the tank is small so that large sizes are to be expected. It would be necessary to carry a 5-in. riser down to the branch of the top floor, then reduce to 4 in. for the branch to the floor below the top, and below this the pipes could be sized for a 30 lb drop per 100 ft. In such a case, tank closets should doubtless be used on the top floor.

Had the tank been set 10 ft higher, the head available for friction, while still giving the same pressure at the top fixtures, would have been 0.43 lb × 10 ft or 4.3 lb greater and this, with the 1 lb drop used previously, would give a total allowable drop of 1 lb + 4.3 lb = 5.3 lb which, divided by the 600 ft equivalent run gives a drop per 100 ft of $\frac{5.3 \times 100}{600} = 0.9$ lb.

With this drop, the sizes according to the chart (Fig. 2) are 6 in., 5 in., and 4 in., respectively. If the run is reduced to 200 ft instead of 600 ft, the allowable drop will

Table 6. Typical Calculation of Pipe Sizes on Down-Feed Riser with Flush Valve Water-Closets and Urinals

(Riser No. 2. Fig. 4)

Floor of Bldg.	Fixtures on Floor	Gpm per Fixture	Maximum Gpm on Floor	Maximum Gpm on Riser	Usage (per cent)	Probable Flow in Riser Gpm	Allowable Drop Lb per 100 Ft	Pipe Size In.
1st	1 W. C.	45	45	45	100	45	30	1¼
2nd	2 W. C. 1 U. 1 Lav.	45 30 3	90 30 3					
			123	168	58	98	30	1½
3rd	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9					
			249	417	31	130	30	2
4th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9					
			249	666	24	160	30	2
5th	6 W. C. 4 Lav.	45 3	270 12					
			282	948	19	180	30	2
6th	6 W. C. 4 Lav.	45 3	270 12					
			282	1230	16	196	30	2½
7th	6 W. C. 4 Lav.	45 3	270 12					
			282	1512	14	211	30	2½
8th	6 W. C. 4 Lav.	45 3	270 12					
			282	1794	12	215	2	4

be $\frac{5.3 \text{ lb} \times 100}{200} = 2.7$ lb per 100 ft. This gives 5 in., 4 in., and 3 in., respectively, for the flows of 400, 200, and 100 gpm.

From Example 4 it is evident that, while the down-feed system possesses certain economies in size for the riser portion, it is quite likely to involve large distribution main sizes, especially when the tank is not elevated to a considerable degree.

Example 5. Fig. 4 shows a typical down-feed layout with three risers extending eight stories and with the fixtures noted on each floor. This will be solved assuming that the level of the water in the house tank is 30 ft above the fixtures on the top floor, that the

length of run from the tank to the farthest fixture is 200 ft, equivalent length of fittings 100 ft, and the pressure required at the fixture is 7 lb.

The 30-ft head is equal to a static pressure of 0.43 × 30 or 12.9 lb per square inch and to maintain a pressure of 7 lb at the highest fixtures the drop allowable in pressure is 12.9 − 7.0 lb or 5.9 lb. As the total equivalent run is 300 ft, this is a drop per 100 ft of 1.97 lb, or practically 2 lb. Therefore, all risers and mains from the top floor back to the tank must be sized on the basis of a drop of 2 lb per 100 ft. Tables 5, 6, 7 and 8 show the schedule for Risers Nos. 1, 2 and 3 with the maximum flow taken from Table 1, the percentage of use at the peak taken from Fig. 1, and the probable flow at the peak worked

TABLE 7. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS

(Riser No. 3. Fig. 4)

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXTURE	MAXIMUM GPM ON FLOOR	MAXIMUM GPM ON RISER	USAGE (PER CENT)	PROBABLE FLOW IN RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN.
1st	1 S. S.	4	4	4	100	4	30	¾
2nd	3 W. C. 1 Lav.	45 3	135 3 138	142	63	89	30	1½
3rd	2 Lav.	3	6	148	61	90	30	1½
4th	3 W. C. 1 Lav. 1 S. S.	45 3 4	135 3 4 142	290	41	119	30	2
5th	1 S. S.	4	4	294	41	120	30	2
6th	1 S. S.	4	4	298	40	120	30	2
7th	1 S. S.	4	4	302	40	121	30	2
8th	1 S. S.	4	4	306	40	122	2	3

TABLE 8. SIZE OF DISTRIBUTION MAIN FOR DOWN-FEED SYSTEMS (SEE FIG. 4)

RISER NO.	MAXIMUM GPM RISER	MAXIMUM GPM MAIN	USAGE (PER CENT)	PROBABLE GPM	ALLOWABLE DROP LB PER 100 FT	SIZE OF MAIN IN.
1	1038	1038	18	187	2	4
2	1794	2832	9	255	2	4
3	306	3138	9	282	2	5

out for each portion of the riser. Riser sizes are taken from Fig. 2, using a drop of 30 lb per 100 ft except on the riser from the top story back to the tank where 2 lb per 100 ft is the allowable limit.

Since down-feed risers are nearly always sized for a pressure loss of 30 lb per 100 ft, it is possible to arrange useful sizing data in tabular form. Fig. 5 shows a typical down-feed riser for a 20 story building. Table 9 may be used for sizing such a riser of any height and for any probable flow up to 250 gpm.

It should be noted that the two top floors in Fig. 5 are sized for less than 30 lb per 100 ft. Regardless of the height of the riser being sized, the two top floors of it should be sized from values given for the top floors of Fig. 5.

TABLE 9. SCHEDULE OF SIZES FOR DOWN-FEED RISER (SEE FIG. 5)

Portion of Riser	Allowable Drop per Lb per 100 Ft	Probable Flow, Gallons per Minute: 5	10	15	20	25	30	40	50	60	70	80	90	100	125	150	200	250
T	3.5	¾	1	1¼	1¼	1½	1½	2	2	2	2½	2½	2½	3	3	3	3½	3½
S	20	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	2	2	2	2	2	2½	3
R	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
Q	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
P	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
O	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
N	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
M	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
L	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
K	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
J	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
I	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
H	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
G	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
F	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
E	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
D	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
C	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
B	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
A	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½

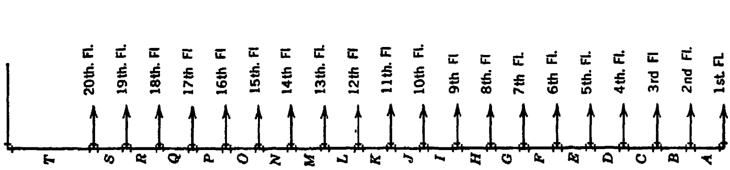


FIG. 5. TYPICAL RISER FOR 20-STORY BUILDING

HOT WATER SUPPLY PIPING

The same basic principles used in the design of cold water piping are also applicable to hot water systems. Hot water, like cold water, may be distributed by either up-feed or down-feed systems.

It is common practice to provide circulation in a hot water supply system so that hot water may be quickly available when the faucet is opened. If this is not done, it is necessary to drain all of the cold water

from the lines between the faucet and the heater, before hot water can be obtained.

Three common methods of arranging hot water circulating lines are illustrated in Fig. 6. Although the diagrams are for multi-story buildings, arrangements *a* and *b* are also used frequently in residences.

A check valve should be provided in the runout from each circulating riser to prevent temporary reversal of flow in the line when a faucet is opened.

Proper air venting of a circulated system is extremely important, particularly if gravity circulation is employed. In Fig. 6*a* and 6*b* this is accomplished by connecting the circulating line below the top fixture supply. Air is thus eliminated from the system each time the top fixture is opened. Where an overhead main is located above the highest fixture,

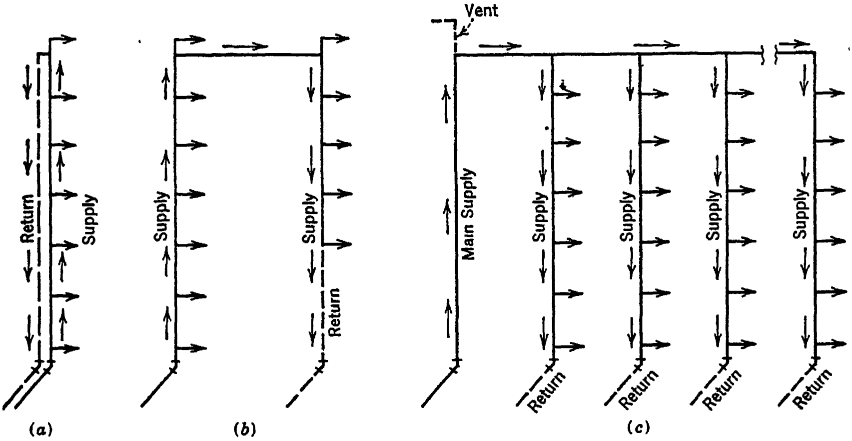


Fig. 6. Methods of Arranging Hot Water Circulation Lines

as in Fig. 6*c* an automatic float type air vent is installed at the highest point of the system.

The supply riser in Fig. 6*a* would be sized exactly the same as an up-feed cold water riser. The sizing of Fig. 6*b* and 6*c* would involve calculations for both up-feed and down-feed risers. Pressure loss in the overhead main would have to be considered in sizing the up-feed risers.

The return line for a gravity circulating system should never be less than ¾ in. Where supply risers are large or lines are long, larger circulating lines may be indicated. Pumps are frequently used in large systems to provide positive circulation.

It is sometimes necessary to make an allowance for pressure drop in the heater when sizing hot water lines. This is particularly true where instantaneous heaters are used.

STORAGE CAPACITY AND HEATING LOAD

In estimating the size of hot water storage tank required and the heating capacity to be provided either from the boiler or from an inde-

pendent domestic hot water heater, it is necessary to know the total quantity of water to be heated per day, and the maximum amount which will be used in any one hour, as well as the duration of the peak load.

In cases where the requirements for hot water are reasonably uniform, as in residences, apartment buildings, hotels, and the like, smaller storage capacity is required than in the case of factories, schools, office buildings, etc., where practically the entire day's usage of hot water occurs during a very short period. Correspondingly, the heating capacity must be proportionately greater with uniform usage of hot water than with intermittent usage where there may be several hours between peak demands during which the water in the storage tank can be brought up to temperature. As a general rule it is desirable to have a large storage capacity in

Table 10. Estimated Hot Water Demand per Person for Various Types of Buildings

Type of Building	Hot Water Required at 140 F	Max. Hourly Demand in Relation to Day's Use	Duration of Peak Load Hours	Storage Capacity in Relation to Day's Use	Heating Capacity in Relation to Day's Use
Res., apts., hotels, etc.	40 gal per person per day	1/7	4	1/5	1/7
Office buildings	2 gal per person per day	1/5	2	1/5	1/6
Factory buildings	5 gal per person per day	1/3	1	2/5	1/8
Restaurants $0.50 meals $1.00 meals $1.50 meals	 1.5 gal per meal 2.5 gal per meal 4.5 gal per meal			1/10	1/10
Restaurants 3 meals per day		1/10	8	1/5	1/10
Restaurants 1 meal per day		1/5	2	2/5	1/6

order that the heating capacity and consequently the size of the heater, or the load on the heating boiler may be as small as possible.

In estimating the hot water which can be drawn from a storage tank it should be borne in mind that only about 75 per cent of the volume of the tank is available, as by the time this quantity has been drawn off the incoming cold water has cooled the remainder down to a point where it can no longer be considered hot water.

Where steam from the heating boiler is used to heat domestic hot water, the computed load on the boiler should be increased by 4 sq ft EDR (equivalent direct radiation) for every gallon of water per hour heated through a 100 F rise. The actual requirement is $\frac{100 \times 8.33}{240} = 3.48$ sq ft

per gallon of water heated 100 F. The value of 4 allows for transmission losses, etc.

There are two ways in common use of estimating the hot water requirements of a building; first, by the number of people and second, by the number of plumbing fixtures installed. Where the number of people to be served is known or can be reasonably estimated, the data in Table 10 may be used.

Example 6. From Table 10, a residence housing five people would have a daily requirement of 5 × 40 = 200 gal per day, and a maximum hourly demand of 200 × $\frac{1}{7}$ = 28.5 gal. The heater should have a storage capacity of 200 × $\frac{1}{5}$ = 40 gal and a heating capacity of 200 × $\frac{1}{7}$ = 28.5 gal per hour.

The conditions given in Example 6 may be cited as average. It is possible to vary the storage and heating capacity by increasing and

TABLE 11. HOT WATER DEMAND PER FIXTURE FOR VARIOUS TYPES OF BUILDINGS

Gallons of water per hour per fixture, calculated at a final temperature of 140 F

	Apartment House	Club	Gymnasium	Hospital	Hotel	Industrial Plant	Office Building	Public Bath	Private Residence	School	Y.M.C.A.
Basins, private lavatory	2	2	2	2	2	2	2	2	2	2	2
Basins, public lavatory	4	6	8	6	8	12	6	12	.	15	8
Bathtubs	20	20	30	20	20	30		45	20	..	30
Dishwashers	15	50–150	..	50–150	50–200	20–100		.	15	20–100	20–100
Foot basins	3	3	12	3	3	12	.		3	3	12
Kitchen sink	10	20	.	20	20	20			10	10	20
Laundry, stationary tubs	20	28		28	28	.	..	.	20		28
Pantry sink	5	10		10	10	.	.	..	5	10	10
Showers	75	150	225	75	75	225		225	75	225	225
Slop sink	20	20	...	20	30	20	15	15	15	20	20
Hourly heating capacity factor	30%	30%	40%	25%	25%	40%	30%	50%	30%	40%	40%
Storage capacity factor	125%	90%	100%	60%	80%	100%	200%	120%	70%	100%	100%

decreasing one over the other. Such a condition is illustrated in Example 7.

Example 7. Assume an apartment house housing 200 people. From the data in Table 10: Daily requirements = 200 × 40 = 8000 gal. Maximum hours demand = 8000 × $\frac{1}{7}$ = 1140 gal. Duration of peak load = 4 hours. Water required for 4-hour peak = 4 × 1140 = 4560.

If a 1000 gal storage tank is used, hot water available from the tank = 1000 × 0.75 = 750. Water to be heated in 4 hours = 4560 − 750 = 3710 gal. Heating capacity per hour = $\frac{3710}{4}$ = 930 gal.

If instead of a 1000 gal tank, a 2500 gal tank had been installed, the required heating capacity per hour would be $\frac{4560 - (2500 \times 0.75)}{4}$ = 671 gal.

In cases where the number of fixtures only are known, the data in Table 11 have been found satisfactory.

Example 8. An apartment building has a hot water requirement as follows:

60 lavatories	× 2 =	120 gal per hour
30 bath tubs	× 20 =	600 gal per hour
30 showers	× 75 =	2250 gal per hour
60 kitchen sinks	× 10 =	600 gal per hour
15 laundry tubs	× 20 =	300 gal per hour
Maximum hourly requirement	=	3870 gal per hour
Hourly heating capacity	= 3870 × 0.30 =	1161 gal per hour
Storage capacity	= 1161 × 1.25 =	1450 gal per hour

METHODS OF HEATING WATER

Service water generally is heated either by direct combustion of fuel or by an intermediate carrier such as steam or hot water. A third method, also in use, is by contact with electrically heated surfaces. The oldest

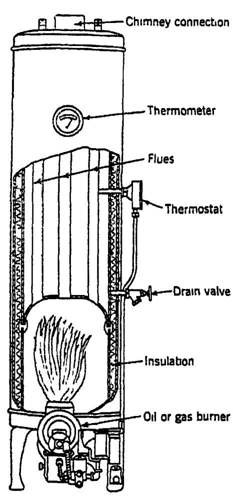


FIG. 7. OIL- OR GAS-FIRED STORAGE TYPE WATER HEATER

method is by fire on one side of a metal barrier and water on the other. If the water surfaces in such a method of heat transfer are small, and if the water carries a heavy proportion of precipitable salts, the water passages soon clog and then burn out. A familiar example of trouble is that with the water back of the firebox in the kitchen stove or the pipe coil inserted into the firebox of a warm air furnace. The critical water temperature at which the lime, magnesia, etc. collect on hot surfaces, varies with the character and proportions of the solids, but generally such deposits are not a serious trouble with water temperatures lower than 140 F.

Coal burning direct-fired water heaters frequently are of cored-out cast-iron, with water entirely surrounding the combustion chamber. They are also made of steel with water tubes which in some cases form racks to suspend garbage above the fire. These heaters are generally so small that low temperature combustion at poor efficiency ensues. Mud and scale may eventually close the water ways.

Oil burning direct-fired water heaters usually are of steel and operate with higher flame temperature and better efficiency then commensurate sized coal burning heaters. They have the same tendency to *lime up*, and the water passages should be large in cross-section and accessible for periodic cleaning.

A domestic hot water heater adapted to burn either oil or gas is shown in Fig. 7. The combustion chamber is rather large, with flues extending vertically through the hot water storage compartment. The thermostat is placed at about the center of the tank.

Gas burning direct-fired water heaters may also be of water tube type, usually having spiral copper coils around which the gas flames are directed to pass. While these heaters, when well insulated, reach high efficiency, the water tubes are especially subject to clogging if the water carries dissolved minerals and is heated above approximately 140 F.

Gas-fired water heaters may also be of the instantaneous type. Such a heater uses no storage tank, but contains sufficient length of copper coil

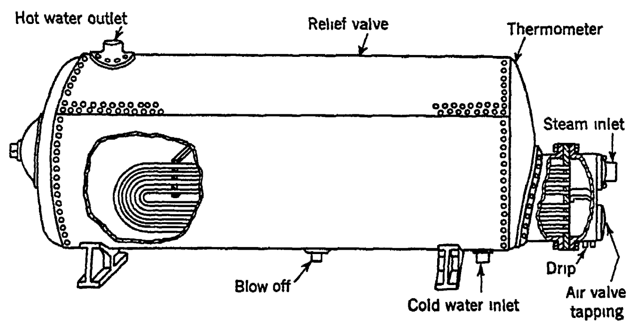


FIG. 8. INDIRECT HEATING COIL IN STEEL TANK

that the water may be heated to the desired temperature during a single passage through it.

The main gas valve on this type of heater is operated by water pressure, and opens wide whenever a faucet is opened. A second gas valve, operated by a thermostat in the outgoing water, throttles the gas to control the leaving water temperature.

Heaters of this type are economical in operation, since, except for the pilot flame, gas is consumed only when hot water is actually being used. These heaters should not be used with hard water, since the long coils become quickly clogged with scale.

Indirect water heaters employ a carrrier, such as steam, between the fire and the water. The domestic water preferably circulates around the outside of the steam tubes which are submerged within a tank.

A typical indirect heating coil in a steel tank is shown in Fig. 8. The coils usually are of copper and are *U* shaped to permit expansion and contraction. Where straight tubes are used, one end of the tube is usually expanded into a *floating* head to take care of expansion. In any type of construction the heat transfer surface should be capable of easy

withdrawal for inspection and for removal of scale. The heating element in Fig. 8 is adapted to use of either steam or hot water inside the tubes.

Another method of transferring heat from a house heating boiler to the domestic water is illustrated in Fig. 9. The water heater usually is a cast-iron vessel within which there is a spiral copper coil similar to the

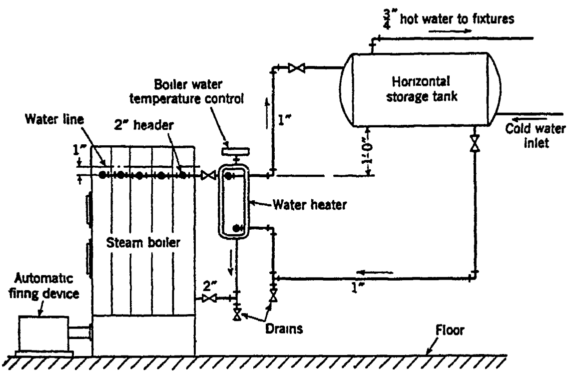


FIG. 9. INDIRECT WATER HEATER MOUNTED ON SIDE OF BOILER

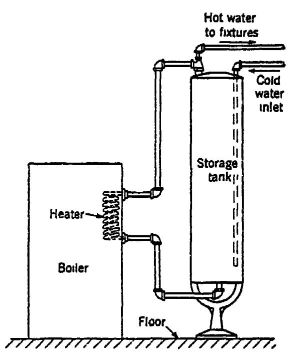


FIG. 10. INDIRECT WATER HEATER PLACED IN BOILER

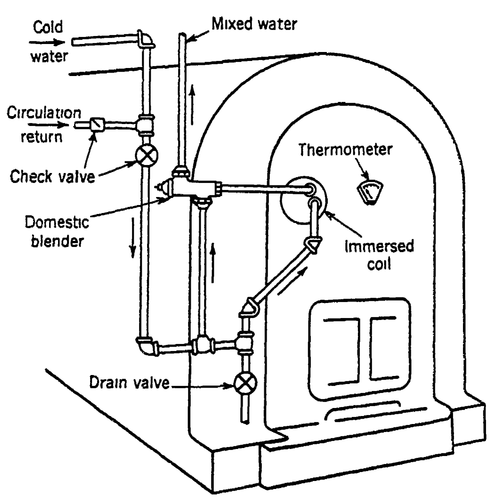


FIG. 11. HEATER INSERTED THROUGH HEAD OF BOILER

one shown in dotted lines in Fig. 10. Steam or water from the heating boiler circulates into the vessel around this coil and returns to the boiler, while domestic water from the storage tank circulates through the coil and receives the heat. The storage tank should be installed with the bottom of the tank as far above the boiler as possible. Horizontal storage tanks smaller than 18 or 20 in. diameter are not recommended because

of the difficulty of preventing the hot and cold water from mixing, and especially is this an important consideration when large quantities of water are withdrawn. In Fig. 10 the heat transfer surface is placed inside the boiler instead of in a separate vessel, but otherwise the operation is similar to that of Fig. 9. This arrangement with vertical tank is commonly used for small domestic installations.

An adaptation of the scheme shown in Fig. 10 suited to larger boilers is shown in Fig. 11. In this case the copper heat transfer surface is in the form of a number of straight tubes with rear *U* bends or a floating head, inserted through the front head of a horizontal fire-tube boiler. The heater may be above the water line of a steam boiler, though it operates more satisfactorily when below the water line since clogging of the water tubes may thereby be delayed.

In Fig. 11 a thermostatic three-way mixing valve or blender is arranged to maintain a uniform temperature of the hot water flowing to the fixtures. These heaters are widely used without storage tanks since the intimate contact and efficient circulation of the water in this arrangement permit utilizing to a great extent the storage of heat in the water of the boiler. A physical limitation for this type of heater is the desirability, if not the necessity, of having the head of the boiler free from any smoke box or breeching, since otherwise the water piping to the heater would be subject to the heat and corrosive influence of the products of combustion.

In order to reduce clogging by precipitated solids, water heating plants sometimes develop steam in a closed circuit, transferring the heat through a tubular heater to the domestic water. The water in the primary heater, exposed to the high temperature of the fire is repeatedly used and hence has no appreciable tendency to deposit scale, while the domestic water, heated by steam at a much lower temperature than that of the fire, also exhibits a much reduced tendency to separate its dissolved salts.

COMPUTING GRATE AREA FOR COAL-FIRED HEATER

The grate area required for a small coal-fired water heater may be calculated by Equation 1.

$$G = \frac{W\,(t_2 - t_1) \times 100}{H \times E \times C} \tag{1}$$

where

G = grate area, square feet.
W = weight of water, pounds per hour.
$t_2 - t_1$ = temperature difference between entering and leaving water, degrees Fahrenheit.
H = heating value of coal, Btu per pound.
C = weight of coal burned, pounds per hour per square foot of grate.
E = efficiency, per cent.

In a small heater 4.5 lb is a conservative value for C, and an efficiency of 60 per cent would represent excellent performance.

Example 9. What grate area is required for a coal-burning water heater warming 100 gal per hour of water from 50 to 180 F, when the combustion rate is 4.5 lb per hour

per square foot of grate, if the heating value of the fuel is 12,500 Btu per pound, and the efficiency is 60 per cent?

$$\text{Substituting: } \frac{100 \times 8.3 \times (180 - 50)}{12{,}500 \times 0.60 \times 4.5} = 3.2 \text{ sq ft.}$$

The quantity of gas, oil, or other fuel required per hour for water heating may be calculated by the Equation 2.

$$F = \frac{W\,(t_2 - t_1) \times 100}{H \times E} \tag{2}$$

where

F = units of fuel (lb, cu ft, gal, etc).
H = heating value of fuel, Btu per unit.
W = weight of water, pounds per hour.
$t_2 - t_1$ = temperature difference between entering and leaving water, degrees Fahrenheit
E = efficiency, per cent.

Efficiencies for oil and gas may be taken as 75 and 80 per cent respectively. The heating value of the fuel and the temperature rise should be determined to suit local conditions.

COMPUTING AREA OF HEAT TRANSMITTING SURFACE

The area of the inside surface of a heating coil may be determined from the following Equation 3.

$$A = \frac{Q \times 8.33\,(t_2 - t_1)}{K_o \times t_m} \tag{3}$$

where

A = surface area of coil, square feet.
Q = quantity of water heated, gallons per hour.
t_2 = hot water outlet temperature, degrees Fahrenheit.
t_1 = cold water inlet temperature, degrees Fahrenheit.
K_o = coefficient of heat transmission, Btu per hour per square foot surface. For copper or brass coils K_o = 240 (steam) and 100 (hot water).
For iron coils K_o = 160 (steam) and 67 (hot water).
t_m = logarithmic mean of the difference between the temperature of the heating medium and the average water temperature. t_m is approximately = $t_s - \left[\frac{(t_o + t_i)}{2}\right]$
t_s = temperature of the coil surface, degrees Fahrenheit.

Equation 3 may be used to check the heating coil ratings under temperature conditions differing from those stated in the manufacturer's published ratings.

Example 10. What area of copper transfer surface will be required to heat 70 gal per hour from 40 to 180 F with boiler water at 220 F?

$$t_m = \left[220 - \frac{(180 + 40)}{2}\right] = 110$$

$$A = \frac{70 \times 8.33\,(180 - 40)}{100 \times 110} = 7.39 \text{ sq ft.}$$

The rate of heat transfer between steam or water as the carrier and the domestic water is influenced by the rate of movement of both the carrier and the water which receives the heat. For this reason, where the transfer is from heating system water to domestic water, it is good practice to install a circulating pump to insure rapid movement of the boiler water.

In view of the high condensation rates when steam is used with gravity circulation from the boiler and when there is a sudden demand followed by an inflow of cold water, the bottom of a steam heating transfer element always should be at least 30 in. above the boiler water line, and the steam and condensate return pipes should be of liberal size. Otherwise water hammer and reduced capacity may result due to imperfect drainage of condensate.

When connecting a transfer-type hot water heater below the water line of a cast-iron steam boiler having vertical sections, there should be a separate tapping for water circulation into every section of the boiler, as shown in Fig. 9. Ordinarily in steam boilers of this type the top connecting nipples between the sections are in the steam space and thus no full internal circulation of water can occur. If a connection to any section is omitted, steaming may take place in that section during summer operation when steam generation is undesirable. Water heating capacity would also be reduced.

CONTROL OF SERVICE WATER TEMPERATURE

With coal-fired heaters the usual arrangement of control is by an aquastat in the heated water, which opens or closes draft dampers at the heater to adjust the rate of combustion. When oil or gas is burned the aquastat controls the oil burner motor or the magnetic gas valve and the pilot flame usually burns continuously. When electric heaters are used the aquastat operates a switch on the source of energy.

When steam or hot water is the carrier of heat to transfer surface submerged in the water of a tank, the aquastat controls a valve in the transmitting line. In small residence installations using water as the carrier a combined aquastat and butterfly valve all in one simple fitting may be installed in the transmitting circuit to prevent overheating of the service water.

In many residences heated by mechanically circulated hot water, the house temperature is controlled by operating the circulating pump intermittently, while domestic hot water is warmed by transfer from the house heating water, independent of the pump operation. The domestic water may be heated by the main boiler the year around. Under such an arrangement, to prevent overheating the house by thermal circulation when the pump is not running, it is usual to insert a weighted check-valve in the house heating main, so that no circulation to the house heating system can occur unless the pump operates. In summer the fire may be controlled to maintain a limited water temperature, generally about 20 F warmer than that desired in the out-going domestic hot water.

In buildings which have restaurants it is generally desirable to install two separate service hot water systems so that water at about 180 F minimum may be available for dish washing, while water at 140 F maximum may be used for lavatory and bath purposes.

The temperature-controlling aquastat in a hot water storage tank ought to be no higher than the center of the tank, and possibly should be even closer to the bottom since water in a tank stratifies proportionally to the temperature. When hot water is removed, the cold water entering to replace it quickly reduces the temperature in the lower parts of the tank.

SWIMMING POOL HEATING REQUIREMENTS

Swimming pools present a problem of hot water heating demand which is frequently overestimated. Few outdoor swimming pools require water heating, and in some cases they require the addition of cold water to regulate the temperature. The recirculation system of a swimming pool consists of the pumps, hair and lint catchers, and filters together with all necessary pipe connections to the inlets and outlets of the pool. The water heater, the sterilizing equipment and suction cleaner are usually installed or connected to the recirculation system and may be considered as integral parts of the system.

The recirculation system and all its component parts should be designed to provide the required volume of circulation so that the water turnover ratio is at least two times per day and where heavy loads are anticipated the turnover ratio should be increased to three times or more. Many states have regulations prescribing the circulation turnover.

The water heaters for swimming pools are usually instantaneous steam coil heaters. These heaters should be sized so that they will have sufficient capacity to heat the water delivered by the circulating pump 15 F per hour.

The water temperature in a pool is usually maintained at about 72 F. A few states have regulations prohibiting higher water temperatures than 70 F. The room temperature should be approximately 5 F higher, but not more than 8 F higher nor less than 2 F lower, than the water temperature.

Example 11. Assume a swimming pool 75 ft long, 30 ft wide with an average depth of 6 ft. If the water is to be heated from a temperature of 50 to 65 F, what capacity heater and steam consumption is required with a turnover ratio of two times per day?

Pool volume: $75 \times 30 \times 6 \times 7.5 = 100{,}000$ gal.

With a turnover ratio of twice in 24 hr, the heating capacity is: $\frac{100{,}000 \times 2}{24} = 8333$ gal per hour.

The steam consumption would be: $\frac{8333 \times 8.33\,(65 - 50)}{970} = 1080$ lb steam per hour.

Regulation of swimming pool temperatures is essential for successful operation and economy. It is therefore recommended that the steam supply to the heater be provided with a by-pass which may be used for pool filling and initial heating and that a smaller by-pass be installed with an automatic control valve having the capacity to heat the circulation water approximately 5 F per hour.

Chapter 47

TERMINOLOGY

Glossary of Physical and Heating, Ventilating and Air Conditioning Terms Used in the Text, Standard Abbreviations, Conversion Equations, Drafting Symbols, Specific Heat Table

Absolute Humidity: *See Humidity.*

Absolute Pressure: The pressure referred to that of a perfect vacuum. It is the sum of gage pressure and barometric pressure.

Absolute Temperature: A reading on the absolute temperature scale. Absolute temperature is obtained by adding 459.70 degrees to the Fahrenheit temperature.

Absolute Zero: The zero point on the absolute scale 459.70 F below the zero of the Fahrenheit scale.

Acceleration: The rate of change of velocity. In the fps system this is expressed in units of one foot per second. $a = V \div t$.

Acceleration Due to Gravity: The rate of gain in velocity of a freely falling body, the value of which varies with latitude and elevation. The international gravity standard has the value of 980.665 cm per second per second or 32.174 ft per second per second, which is the actual value of this acceleration at sea level and about 45 deg latitude.

Adiabatic: An adjective descriptive of a process in which no heat is added to or extracted from the system executing the process.

Air Cleaner: A device designed for the purpose of removing air-borne impurities such as dusts, fumes and smokes. (Air cleaners include air washers and air filters.)

Air Conditioning: The simultaneous control of all or at least the first three of those factors affecting both the physical and chemical conditions of the atmosphere within any structure. These factors include temperature, humidity, motion, distribution, dust, bacteria, odors and toxic gases, most of which affect in greater or lesser degree human health or comfort. (See *Comfort Air Conditioning*.)

Air Washer: An enclosure in which air is forced through a spray of water in order to cleanse, humidify, or dehumidify the air.

Anemometer: An instrument for measuring the velocity of moving air.

Atmospheric Pressure: The pressure indicated by a barometer. *Standard atmospheric pressure* is a pressure of 76 cm mercury (density 13.5951 grams per cubic centimeter, gravity 980.665 cm per second per second). It is equivalent to 14.6959 lb per square inch or 29.921 in. of mercury at 32 F.

Baffle: A plate or wall for deflecting gases or fluids.

Blast: This word was formerly used to denote forced air circulation, particularly in connection with central fan systems using steam or hot water as the heating medium. As applied in this sense, the word *blast* is now obsolete.

Boiler: A closed vessel in which steam is generated or in which water is heated.

Boiler Heating Surface: That portion of the surface of the heat-transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other, in which the fluid being heated forms part of the circulating system; this surface shall be measured on the side receiving heat. This includes the boiler, water walls, water screens, and water floor. (*A.S.M.E.* Power Test Codes, Series 1929.)

Boiler Horsepower: The equivalent evaporation of 34.5 lb of water per hour from and at 212 F. This is equal to a heat output of $970.3 \times 34.5 = 33{,}475$ Btu per hour.

British Thermal Unit: A unit of energy defined in terms of the international steam-table calorie through the convenient relation 1 Btu per pound per degree Fahrenheit = 1 cal per gram per degree Centigrade. It is approximately the quantity of heat required to raise the temperature of 1 lb of liquid water from 63 to 64 F.

By-pass: A pipe or duct, usually controlled by valve or damper, for short-circuiting fluid flow.

Calorie: (large calorie or kilogram calorie) is equal to 1000 international steam-table calories = 1/860 international kilowatthour. For practical purposes it may be considered as 1/100 of the heat required to raise the temperature of 1 kilogram of water from 0 to 100 C.

Central Fan System: A mechanical indirect system of heating, ventilating, or air conditioning, in which the air is treated or handled by equipment located outside the rooms served, usually at a central location, and is conveyed to and from the rooms by means of a fan and a system of distributing ducts. (See Chapter 21.)

Chimney Effect: The tendency in a duct or other vertical air passage for air to rise when heated, owing to its decrease in density.

Coefficient of Transmission: The amount of heat (Btu) transmitted *from air to air* in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 F *between the air on the inside and that on the outside of the wall, floor, roof or ceiling.*

Comfort Air Conditioning: The process by which simultaneously the temperature, moisture content, movement and quality of the air in enclosed spaces intended for human occupancy may be maintained within required limits. (See *Air Conditioning.*)

Comfort Line: The effective temperature at which the largest percentage of adults feel comfortable.

Comfort Zone (*Average*): The range of effective temperatures over which the majority (50 per cent or more) of adults feel comfortable. *Comfort Zone* (*Extreme*): The range of effective temperatures over which one or more adults feel comfortable. (See Chapter 2.)

Concealed Radiator: A heating device located within, adjacent to, or exterior to the room being heated but so covered or enclosed or concealed that the heat transfer surface of the device, which may be either a radiator or a convector, does not *see* the room. Such a device transfers its heat to the room largely by convection air currents.

Conductance: The amount of heat (Btu) transmitted from surface to surface in one hour through one square foot of a material or construction, whatever its thickness, when the temperature difference is 1 F between the two surfaces.

Conduction: The transmission of heat through and by means of matter unaccompanied by any obvious motion of the matter.

Conductivity: The amount of heat (Btu) transmitted in one hour through one square foot of a homogeneous material *1 in. thick* for a difference in temperature of 1 F between the two surfaces of the material.

Conductor (*Heat*): A material capable of readily conducting heat. The opposite of an insulator or insulation.

Constant Relative Humidity Line: Any line on the psychrometric chart representing a series of conditions which may be evaluated by one percentage of relative humidity; there are also *constant* dry-bulb lines, wet-bulb lines, effective temperature lines, vapor pressure lines, and lines showing other physical properties of air mixed with water vapor.

Convection: The transmission of heat by the circulation of a liquid or a gas such as air. Convection may be *natural* or *forced.*

Convector: A heat transfer surface designed to transfer its heat to surrounding air largely or wholly by convection. Such a surface may or may not be enclosed or concealed. When concealed and enclosed the resulting device is sometimes referred to as a concealed radiator. (See also definition of *Radiator.*) (See also Chapter 13.)

Decibel: A unit commonly used for expressing sound or noise intensities referred to an arbitrary reference level. It is defined by the relation db $= 10 \log_{10} \frac{P_1}{P_0}$, where P_1 is the unknown intensity, and P_0 is the reference level which is commonly taken as 10^{-16} watts per square centimeter.

Degree-Day: A unit, based upon temperature difference and time, used in specifying the nominal heating load in winter. For any one day there exists as many degree-days as there are degrees Fahrenheit difference in temperature between the mean temperature for the day and 65 F.

Degree of Saturation or Per Cent Saturation: The ratio of actual humidity ratio W to the saturation humidity ratio W_s corresponding to the actual temperature and the observed pressure. $\mu = \frac{W}{W_s}$ (Approximately the same as but not identical with *relative humidity.* See Chapter 1).

Dehumidification: The condensation of water vapor from air by cooling below the dew-point.

Dehydration: The removal of water vapor from air by the use of adsorbing or absorbing materials.

Density: The weight of a unit volume, expressed in pounds per cubic foot. $d = W \div V$.

Dew-Point Temperature: The temperature corresponding to saturation (100 per cent relative humidity) for a given moisture content.

Direct-Indirect Heating Unit: A heating unit located in the room or space to be heated and partially enclosed, the enclosed portion being used to heat air which enters from outside the room.

Direct Radiator: Same as *Radiator*.

Direct-Return System (*Hot Water*)*:* A hot water system in which the water, after it has passed through a heating unit, is returned to the boiler along a direct path so that the total distance traveled by the water is the shortest feasible, and so that there are considerable differences in the lengths of the several circuits composing the system.

Down-Feed One-Pipe Riser (*Steam*)*:* A pipe which carries steam downward to the heating units and into which the condensation from the heating units drain.

Down-Feed System (*Steam*)*:* A steam heating system in which the supply mains are above the level of the heating units which they serve.

Draft Head (*Side Outlet Enclosure*)*:* The height of a gravity convector between the bottom of the heating unit and the bottom of the air outlet opening. (*Top Outlet Enclosure*)*:* The height of a gravity convector between the bottom of the heating unit and the top of the enclosure.

Drip: A pipe, or a steam trap and a pipe, considered as a unit, which conducts condensation from the steam side of a piping system to the water or return side of the system.

Dry Air: In psychrometric work, *dry air* is defined as air without water vapor. This state, though not obtained practically, is used as the basis of calculations.

Dry-Bulb Temperature: The temperature indicated by a standardized thermometer after correction for radiation, etc.

Dry Return: A return pipe in a steam heating system which carries both water of condensation and air. The dry return is above the level of the water line in the boiler in a gravity system. (See *Wet Return.*)

Dust: Solid material in a finely divided state, the particles of which are large and heavy enough to fall with increasing velocity, due to gravity in still air. For instance, particles of fine sand or grit, the average diameter of which is approximately 0.01 centimeter, such as are blown on a windy day, may be called dust.

Dynamic Head or Pressure: Same as *Total Pressure.*

Effective Temperature: An arbitrary index which combines into a single value the effect of temperature, humidity, and movement of air on the degree of warmth or cold felt by the human body. The numerical value is that of the temperature of still, saturated air which would induce an identical sensation of warmth.

Enthalpy: A thermodynamic property which serves as a measure of the quantity of thermal energy *convected* by a fluid in steady flow. In a non-flow process the increase of enthalpy equals the quantity of heat absorbed *provided pressure is constant.* Enthalpy was formerly called *heat content,* sometimes *total heat.* Specific enthalpy is the ratio of total enthalpy to total weight, that is, enthalpy per unit weight of substance, Btu per pound.

Entropy: A thermodynamic property which, for practical purposes, is best defined by stating its principal functions: (1) during a reversible *adiabatic* change of state, entropy is constant; (2) during a reversible *isothermal* change of state, the heat absorbed is equal to absolute temperature times change of entropy. *Specific entropy* is the ratio of total entropy to total weight, that is, entropy per unit weight, Btu per degree Fahrenheit per pound.

Equivalent Evaporation: The amount of water a boiler would evaporate, in pounds per hour, if it received feed water at 212 F and vaporized it at the same temperature and atmospheric pressure.

Estimated Design Load: The sum of the heat emission of the equivalent direct radiation to be installed plus the allowance for heat loss of the connecting piping plus the heat requirements of any auxiliary apparatus connected with the system.

Estimated Maximum Load: The load stated in Btu per hour or equivalent direct radiation that has been estimated to be the greatest or maximum load that the boiler will be called upon to carry.

Extended Heating Surface: See *Heating Surface.*

Extended Surface Heating Unit: A heating unit having a relatively large amount of extended surface which may be integral with the core containing the heating medium or assembled over such a core, making good thermal contact by pressure or by being

soldered to the core or by both pressure and soldering. An extended surface heating unit is usually placed within an enclosure and therefore functions as a convector.

Fan Furnace System: See *Warm Air Heating System.*

Force: The action on a body which tends to change its relative condition as to rest or motion. $F = (WV) \div (gt)$.

Free Enthalpy: A thermodynamic property which serves as a measure of the available energy of a system with respect to surroundings at the same temperature and same pressure as that of the system. No process involving an increase in available energy can occur spontaneously. (See example on *Free Enthalpy* in Chapter 1.)

Fumes: Particles of solid matter resulting from such chemical processes as combustion, explosion, and distillation, ranging from 0.1 to 1.0 micron in size.

Furnace: That part of a boiler or warm air heating plant in which combustion takes place. Also, a fire-pot.

Furnace Volume *(Total):* The total furnace volume for horizontal-return tubular boilers and water-tube boilers is the cubical contents of the furnace between the grate and the first plane of entry into or between tubes. It therefore includes the volume behind the bridge wall as in ordinary horizontal-return tubular boiler settings, unless manifestly ineffective (*i.e.*, no gas flow taking place through it), as in the case of waste-heat boilers with auxiliary coal furnaces, where one part of the furnace is out of action when the other is being used. For Scotch or other internally fired boilers it is the cubical contents of the furnace, flues and combustion chamber, up to the plane of first entry into the tubes. (*A.S.M.E.* Power Test Codes, Series 1929.)

Gage Pressure: Pressure measured from atmospheric pressure as a base. Gage pressure may be indicated by a manometer which has one leg connected to the pressure source and the other exposed to atmospheric pressure.

Grate Area: The area of the grate surface, measured in square feet, to be used in estimating the rate of burning fuel. This area is construed to mean the area measured in the plane of the top surface of the grate, except that with special furnaces, such as those having magazine feed, or special shapes, the grate area shall be the mean area of the active part of the fuel bed taken perpendicular to the path of the gases through it. For furnaces having a secondary grate, such as those in double-grate down draft boilers, the effective area shall be taken as the area of the upper grate plus one-eighth of the area of the lower grate, both areas being estimated as previously defined.

Gravity Warm Air Heating System: See *Warm Air Heating System.*

Heat: Heat is that form of energy which transfers from one system to a second system at lower temperature by virtue of the temperature difference, when the two are brought into communication.

Heating Medium: A substance such as water, steam, air, or furnace gas used to convey heat from the boiler, furnace or other source of heat or energy to the heating unit from which the heat is dissipated.

Heating Surface: The exterior surface of a heating unit. *Extended heating surface (or extended surface):* Heating surface having air on both sides and heated by conduction from the prime surface. *Prime Surface:* Heating surface having the heating medium on one side and air (or extended surface) on the other. (See also *Boiler Heating Surface.*)

Heat of the Liquid: This can usually be interpreted as the specific enthalpy of saturated liquid.

Hot Water Heating System: A heating system in which water is used as the medium by which heat is carried through pipes from the boiler to the heating units.

Humid Heat: Ratio of increase of enthalpy per pound of dry air to rise of temperature under conditions of constant pressure and constant humidity ratio.

Humidify: To add water vapor to the atmosphere; to add water vapor or moisture to any material.

Humidistat: A regulatory device, actuated by changes in humidity, used for the automatic control of relative humidity.

Humidity: Water vapor when mixed with dry air or other dilutent gases. *Absolute humidity* is the weight of water vapor per unit volume of moist air, pounds per cubic foot. It can be calculated by dividing the humidity ratio, weight of water vapor per pound of dry air, by the volume of the mixture per pound of dry air. *Relative humidity* is the ratio of the partial pressure of the water vapor in the air to the saturation pressure of pure water corresponding to the actual temperature. (See Chapter 1.)

Humidity Ratio: Weight of water vapor per pound of dry air. (Formerly called specific humidity.)

Hygrostat: Same as *Humidistat.*

Inch of Water: The pressure due to a column of liquid water one inch high at a temperature of 60 F.

Insulation (*Heat*)*:* A material having a relatively high heat-resistance per unit of thickness.

Isobaric: An adjective used to indicate a change taking place at constant pressure.

Isothermal: An adjective used to indicate a change taking place at constant temperature.

Latent Heat: The most general interpretation is heat absorbed at constant temperature. More specifically the *latent heat of vaporization* is the difference between the specific enthalpies of saturated vapor and saturated liquid at the same temperature (and, for a pure substance, the same pressure). *Latent heat of sublimation* is the difference between the specific enthalpies of saturated vapor and saturated solid at the same temperature. *Latent heat of fusion* is the difference between the specific enthalpies of saturated liquid and saturated solid at the same temperature.

Laws of Thermodynamics: The Law of Conservation of Energy states that energy, in any of its forms, can neither be created nor destroyed. As a corollary to this, the *First Law* of Thermodynamics states that in any power cycle or refrigeration cycle the net heat absorbed by the working substance is exactly equal to the net work done. The *Second Law* of Thermodynamics states that a power cycle which absorbs heat at a single temperature and converts it wholly into work, as required by the First Law, is impossible; hence it is absolutely necessary to reject heat at some lower temperature if any work is to be done. The Second Law further prescribes the least possible quantity of heat that must be so rejected depending on the two temperatures involved.

Manometer: An instrument for measuring pressures; essentially a U-tube partially filled with a liquid, usually water, mercury, or a light oil, so the amount of displacement of the liquid indicates the pressure being exerted on the instrument.

Mass: The quantity of matter, in pounds, to which the unit of force (one pound) will give an acceleration of one foot per second per second. $m = W \div g$.

Mb, Mbh: Symbols which represent, respectively, 1000 Btu and 1000 Btu per hour.

Mechanical Equivalent of Heat: The conversion factor from Btu to foot pounds; $J = 778.26$ foot pounds per Btu. This is also referred to as Joule's Equivalent.

Micron: A unit of length, the thousandth part of one millimeter or the millionth of a meter.

Mol (*Pound Mol*)*:* A weight in pounds numerically equal to the molecular weight of a substance. In the case of gases, and at not too high pressures, the volume of 1 mol is approximately the same for any gas at the same temperature and pressure. At 32 F and standard atmospheric pressure this volume is 358.65 cu ft.

One-Pipe Supply Riser (*Steam*)*:* A pipe which carries steam upward to a heating unit and which also carries the condensation from the heating unit in a direction opposite to the steam flow.

One-Pipe System (*Hot Water*)*:* A hot water system in which the water flows through more than one heating unit before it returns to the boiler; consequently, the heating units farthest from the boiler are supplied with cooler water than those near the boiler in the same circuit.

One-Pipe System (*Steam*)*:* A steam heating system consisting of a main circuit in which the steam and condensate flow in the same pipe, usually in opposite directions. Ordinarily to each heating unit there is but one connection which must serve as both the supply and the return, although separate supply and return connections may be used.

Overhead System: Any steam or hot water system in which the supply main is above the heating units. With a steam system the return must be below the heating units; with a water system, the return *may* be above the heating units.

Panel Radiator: A heating unit placed on or flush with a flat wall surface and intended to function essentially as a radiator.

Panel Warming: A method of heating involving the installation of the heating units (pipe coils) within the wall, floor or ceiling of the room, so that the heating process takes place mainly by radiation from the wall, floor or ceiling surfaces to the objects in the room.

Plenum Chamber: An air compartment maintained under pressure and connected to one or more distributing ducts.

Potentiometer: An instrument for measuring or comparing small electromotive forces.

Power: The rate of performing work; usually expressed in units of horsepower, Btu per hour, or watts.

Prime Surface: See *Heating Surface.*

Psychrometer: An instrument for ascertaining the humidity or hygrometric state of the atmosphere. *Psychrometric:* Pertaining to psychrometry or the state of the atmosphere as to moisture. *Psychrometry:* The branch of physics that treats of the measurement of degree of moisture, especially the moisture mixed with the air.

Pyrometer: An instrument for measuring high temperatures.

Radiation: The transmission of heat through space by wave motion.

Radiator: A heating unit exposed to view within the room or space to be heated. A radiator transfers heat by radiation to objects *it can see* and by conduction to the surrounding air which in turn is circulated by natural convection; a so-called *radiator* is also a *convector* but the single term *radiator* has been established by long usage.

Recessed Radiator: A heating unit set back into a wall recess but not enclosed.

Refrigerant: A substance which produces a refrigerating effect by its absorption of heat while expanding or vaporizing.

Relative Humidity: See *Humidity;* also discussion relative humidity, Chapter 1.

Return Mains: The pipes which return the heating medium from the heating units to the source of heat supply.

Reversed-Return System (*Hot Water*)*:* A hot water heating system in which the water from several heating units is returned along paths arranged so that all circuits composing the system or composing a major sub-division of the system are practically of equal length.

Roof Ventilator: A device placed on the roof of a building to facilitate egress of air.

Saturated Air: A mixture of dry air and saturated water vapor, all at the same dry-bulb temperature. It may also be considered as air containing the maximum possible amount of water vapor at a given temperature without becoming supersaturated.

Saturation: The condition for coexistence in stable equilibrium of two or more distinct phases, such as steam over the water from which it is being generated.

Saturation Pressure: The saturation pressure for a pure substance for any given temperature is that pressure at which vapor and liquid or vapor and solid can coexist in stable equilibrium.

Sensible Heat: Heat which manifests itself by temperature change.

Smoke: Carbon or soot particles less than 0.1 micron in size which result from the incomplete combustion of carbonaceous materials such as coal, oil, tar, and tobacco.

Smokeless Arch: An inverted baffle placed in an up-draft furnace toward the rear to aid in mixing the gases of combustion and thereby to reduce the smoke produced.

Specific Enthalpy: The ratio of total enthalpy to total weight. The specific enthalpy of air is its enthalpy, Btu per pound, measured above 0 F and 29.921 in. Hg as a reference point. The specific enthalpy of water is its enthalpy, Btu per pound, measured from the reference point of saturated liquid at 32 F. (See *Enthalpy.*)

Specific Gravity: The ratio of the weight of a body to the weight of an equal volume of water at some standard temperature, usually 39.2 F.

Specific Heat: The ratio of heat absorbed per unit weight of substance to temperature rise. For gases, both specific heat at constant pressure, c_p, and specific heat at constant volume, c_v, are frequently given. In air conditioning, c_p is usually used.

Specific Volume: The volume, expressed in cubic feet, of one pound of a substance. $v = 1 \div d = V \div W$.

Split System: A system in which the heating and ventilating are accomplished by means of radiators or convectors supplemented by mechanical circulation of air (heated or unheated) from a central point.

Square Foot of Heating Surface (*Equivalent*)*:* Equivalent Direct Radiation (EDR). That amount of heating surface which will give off 240 Btu per hour. The *equivalent* square feet of heating surface may have no direct relation to the actual surface area.

Stack Height: The height of a gravity convector between the bottom of the heating unit and the top of the outlet opening.

Standard Air: Air weighing 0.075 lb per cubic foot. (The density of air at 29.921 in. of mercury barometric pressure, 68 F dry-bulb and 50 per cent relative humidity is 0.07497; and dry air at 70 F dry-bulb is 0.07496.)

Static Pressure: The normal force per unit area that would be exerted by a moving fluid on a small body immersed in it if the body were carried along with the fluid. Practi-

cally, it is the normal force per unit area at a small hole in a wall of the duct through which the fluid flows (piezometer) or on the surface of a stationary tube at a point where the disturbances created by inserting the tube cancel. It is supposed that the thermodynamic properties of a moving fluid depend on static pressure in exactly the same manner as those of the same fluid at rest depend upon its uniform hydrostatic pressure.

Steam: Water in the vapor phase. *Dry Saturated Steam* is steam at the saturation temperature corresponding to the pressure, and containing no water in suspension. *Wet Saturated Steam* is steam at the saturation temperature corresponding to the pressure, and containing water particles in suspension. *Superheated Steam* is steam at a temperature higher than the saturation temperature corresponding to the pressure.

Steam Heating System: A heating system in which heat is transferred from the boiler or other source of steam to the heating units by means of steam at, above, or below atmospheric pressure.

Steam Trap: A device for allowing the passage of condensate and preventing the passage of steam, or for allowing the passage of air as well as condensate.

Superheated Steam: See *Steam*.

Supply Mains (*Steam*): The pipes through which the steam flows from the boiler or source of supply to the run-outs and risers leading to the heating units.

Surface Conductance: The amount of heat (Btu) transmitted by radiation, conduction, and convection *from a surface to the air or liquid surrounding it*, or vice versa, in one hour per square foot of surface for a difference in temperature of 1 deg between the surface and the surrounding air or liquid.

Therm: 100,000 Btu. (Used in the gas industry.)

Thermal Resistance: The reciprocal of *conductance*.

Thermal Resistivity: The reciprocal of *conductivity*.

Thermostat: An instrument which responds to changes in temperature and which directly or indirectly controls the source of heat supply.

Ton of Refrigeration: The removal of 12,000 Btu of heat per hour at a low temperature.

Ton Day of Refrigeration: The removal of 288,000 Btu of heat at a low temperature.

Total Heat: This can usually be interpreted as increase of enthalpy at constant pressure. It is often regarded as snynonymous with enthalpy.

Total Pressure: In the theory of the flow of fluids; the sum of the static pressure and the velocity pressure at the point of measurement.

Tube (or Tubular) Radiator: A cast-iron heating unit used as a radiator and having small vertical tubes.

Two-Pipe System (*Steam or Water*): A heating system in which one pipe is used for the supply of the heating medium to the heating unit and another for the return of the heating medium to the source of heat supply. The essential feature of a two-pipe system is that each heating unit receives a direct supply of the heating medium which medium cannot have served a preceding heating unit.

Underfeed Distribution System (*Hot Water*): A hot water heating system in which the main flow pipe is below the heating unit.

Underfeed Stoker: A stoker which feeds the coal underneath the fuel bed.

Unit: As applied to heating, ventilating and air conditioning equipment this word means a factory-built and assembled equipment with apparatus for accomplishing some specified function or combination of functions. (See Chapters 22 and 23.)

It is loosely applied to a great variety of equipment. Usually the function is included in the name, and hence come terms like Unit Heater, Unit Ventilator, Humidifying Unit, and Air Conditioning Unit.

Units are said to be ***direct*** or ***room***, when intended for location, or located in, the treated space; ***indirect*** or ***remote***, when outside or adjacent to the treated space. They are ***ceiling*** units when suspended from above, and ***floor*** when supported from below. Other descriptive words include ***free delivery*** when the unit is not intended to be attached to ducts or similar resistance-producing devices, and ***pressure*** when for use with such ducts. Complete description requires the use of several of these qualifying words or phrases. (See Chapter 23.)

Up-Feed System (*Steam*): A steam heating system in which the supply mains are below the level of the heating units which they serve.

Vacuum Heating System: A two-pipe steam heating system equipped with the necessary accessory apparatus which will permit operating the system below atmospheric pressure when desired.

Vapor: Any substance in the gaseous state.

Vapor Heating System: A steam heating system which operates under pressures at or near atmospheric and which returns the condensation to the boiler or receiver by gravity. Vapor systems have thermostatic traps or other means of resistance on the return ends of the heating units for preventing steam from entering the return mains; they also have a pressure-equalizing and air-eliminating device at the end of the dry return. *Direct Vent Vapor System:* A vapor heating system with air valves which do not permit re-entry of air.

Vapor Pressure: Synonymous with saturation pressure in the case of a pure substance.

Velocity: The time rate of motion of a body in a fixed direction. In the fps system it is expressed in units of one foot per second. $V = \frac{s}{t}$.

Velocity Pressure: The difference due to velocity between total pressure and static pressure. It is supposed to equal the kinetic energy per unit volume of the fluid at the point of measurement.

Ventilation: The process of supplying or removing air by natural or mechanical means, to or from any space. Such air may or may not have been conditioned. (See *Air Conditioning.*)

Warm Air Heating System: A warm air heating plant consists of a heating unit (fuel-burning furnace) enclosed in a casing, from which the heated air is distributed to the various rooms of the building through ducts. If the motive head producing flow depends on the difference in weight between the heated air leaving the casing and the cooler air entering the bottom of the casing, it is termed a *gravity* system. A booster fan may, however, be used in conjunction with a gravity-designed system. If a fan is used to produce circulation and the system is designed especially for fan circulation, it is termed a *fan furnace* system or a *central fan furnace* system. A fan furnace system may include air washers and filters.

Wet-Bulb Temperature: *Thermodynamic wet-bulb temperature* is the temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature. *Wet-bulb temperature* (without qualification) is the temperature indicated by a wet-bulb psychrometer constructed and used according to specifications. (*A.S.M.E.* Power Test Codes, Series 1932, Instruments and Apparatus, Part 18.)

Wet Return: That part of a return main of a steam heating system which is filled with water of condensation. The wet return usually is below the level of the water line in the boiler, although not necessarily so. (See *Dry Return.*)

ABBREVIATIONS [1]

Absolute	abs
Acceleration, due to gravity	*g*
Acceleration, linear	*a*
Air horsepower	air hp
Alternating-current (as adjective)	a-c
Ampere	amp
Ampere-hour	amp-hr
Area	*A*
Atmosphere	atm
Average	avg
Avoirdupois	avdp
Barometer	bar.
Boiler pressure	bp
Boiling point	bp
Brake horsepower	bhp
Brake horsepower-hour	bhp-hr
British thermal unit	Btu
Calorie	cal
Centigram	cg
Centimeter	cm

[1]From compilations of abbreviations approved by the *American Standards Association*, Z 10.1-1941 and Z 10 a, c, f, and i.
As a general rule the period is omitted in all abbreviations except where the omission results in the formation of an English word.

Centimeter-gram-second (system) .... cgs
Change in specific volume during vaporization .... v_{fg}
Cubic .... cu
Cubic foot .... cu ft
Cubic feet per minute .... cfm
Cubic feet per second .... cfs
Decibel .... db
Degree[2] .... deg or °
Degree centigrade C
Degree Fahrenheit .... F
Degree Kelvin .... K
Degree Réaumur .... R
Density, Weight per unit volume, Specific weight .... d or ρ (rho)

$$d = \frac{1}{v}$$

Diameter .... D or diam
Direct-current (as adjective) .... d-c
Distance, linear .... s
Dry saturated vapor, Dry saturated gas at saturation pressure and temperature, vapor in contact with liquid .... *Subscript* g
Entropy. (The capital should be used for any weight, and the small letter for unit weight) .... S or s
Feet per minute .... fpm
Feet per second .... fps
Foot .... ft
Foot-pound .... ft-lb
Foot-pound-second (system) .... fps
Force, total load .... F
Freezing point .... fp
Gallon .... gal
Gallons per minute .... gpm
Gallons per second .... gps
Gram .... g
Gram-calorie .... g-cal
Head .... H or h
Heat content, Total heat, Enthalpy. (The capital should be used for any weight and the small letter for unit weight) .... H or h
Heat content of saturated liquid, Total heat of saturated liquid, Enthalpy of saturated liquid, sometimes called heat of the liquid .... h_f
Heat content of dry saturated vapor, Total heat of dry saturated vapor, Enthalpy of dry saturated vapor .... h_g
Heat of vaporization at constant pressure .... L or h_{fg}
Horsepower .... hp
Horsepower-hour .... hp-hr
Hour .... hr
Inch .... in.
Inch-pound .... in.-lb
Indicated horsepower .... ihp
Indicated horsepower-hour .... ihp-hr
Internal energy, Intrinsic energy. (The capital should be used for any weight and the small letter for unit weight) .... U or u
Kilogram .... kg
Kilowatt .... kw
Kilowatthour .... kwhr
Length of path of heat flow, thickness L
Load, total .... W
Mass .... mass
Mechanical efficiency .... e_m
Mechanical equivalent of heat .... J
Melting point .... mp
Meter .... m

[2]It is recommended that the abbreviation for the temperature scale, F, C, K, be included in expressions for numerical temperatures but, wherever feasible, the abbreviations for *degree* be omitted; as 68 F.

Micron .... μ (mu)
Miles per hour .... mph
Millimeter .... mm
Minute .... min
Molecular weight .... mol. wt
Mol .... mol
Ounce .... oz
Pound .... lb
Power, Horsepower, Work per unit time .... P
Pressure, Absolute pressure, Gage pressure, Force per unit area .... p
Quantity (total) of fluid, water, gas, heat; Quantity by volume; Total quantity of heat transferred .... Q
Quality of steam, Pounds of dry steam per pound of mixture x
Revolutions per minute .... rpm
Saturated liquid at saturation pressure and temperature, Liquid in contact with vapor .... *Subscript* f
Second .... sec
Specific gravity .... sp gr
Specific heat .... sp ht or c
Specific heat at constant pressure .... c_p
Specific heat at constant volume .... c_v
Specific volume, Volume per unit weight, Volume per unit mass v
Square foot .... sq ft
Square inch .... sq in.
Temperature (ordinary) F or C. (*Theta* is used preferably only when t is used for Time in the same discussion) .... t or θ (*theta*)
Temperature (absolute) F abs or K. (Capital *theta* is used preferably only when small *theta* is used for ordinary temperature) .... T or Θ (*capital theta*)
Thermal conductance[3] (heat transferred per unit time per degree) C

$$C = \frac{1}{R} = \frac{kA}{L} = \frac{q}{t_1 - t_2}$$

Thermal conductance per unit area, Unit conductance (heat transferred per unit time per unit area per degree) .... C_a

$$C_a = \frac{C}{A} = \frac{1}{RA} = \frac{q}{A(t_1 - t_2)} = \frac{k}{L}$$

Thermal conductivity (heat transferred per unit time per unit area, and per degree per unit length) .... k

$$k = \frac{\frac{q}{A}}{\frac{(t_1 - t_2)}{L}}$$

Surface coefficient of heat transfer, Film coefficient of heat transfer, Individual coefficient of heat transfer (heat transferred per unit time per unit area per degree) .... f

$$f = \frac{\frac{q}{A}}{t_1 - t_2}$$

(In general f is not equal to k/L, where L is the actual thickness of the fluid film.)

Over-all coefficient of heat transfer, Thermal transmittance per unit area (heat transferred per unit time per unit area per degree over-all) .... U

$$U = \frac{\frac{q}{A}}{t_1 - t_2}$$

[3]Terms ending *ivity* designate properties independent of size or shape, sometimes called *specific properties*. Examples: conductivity, resistivity. Terms ending *ance* designate quantities depending not only on the material, but also upon size and shape, sometimes called *total quantities*. Examples: conductance, transmittance. Terms ending *ion* designate rate of heat transfer. Examples: conduction, transmission.

Thermal transmission (heat transferred per unit time) .... q

$$q = \frac{Q}{t}$$

Thermal resistance (degrees per unit of heat transferred per unit time) .... R

$$R = \frac{t_1 - t_2}{q} = \frac{L}{kA}$$

Thermal resistivity .... $1/k$
Vaporization values at constant pressure, Differences between values for saturated vapor and saturated liquid at the same pressure .... *Subscript* fg
Velocity V
Volume (total) V
Volume per unit time, Rate at which quantity of material passes through a machine, Quantity of heat per unit time, Quantity of heat per unit weight .... q
Watt .... w
Watthour .... whr
Weight of a major item, Total weight .... W
Weight rate, Weight per unit of power, Weight per unit of time .... w
Work (total) .... W

CONVERSION EQUATIONS

Heat, Power and Work

1 ton refrigeration	= { 12,000 Btu per hour; 200 Btu per minute
Latent heat of ice	= 143.4 Btu per pound
1 Btu	= { 778.26 ft-lb; 0.293 whr; 252 mean calories
1 watthour	= { 2,655 ft-lb; 3.413 Btu; 3600 joules; 860 mean calories
1 kilowatthour	= { 3,413 Btu; 3.517 lb water evaporated from and at 212 F
1 kilowatt (1000 watts)	= { 1.341 hp; 56.88 Btu per minute; 44,253 ft-lb per minute
1000 mean calorie } 1 kilogram calorie }	= { 3.969 Btu; 3087 ft-lb; 1.1627 whr
1 horsepower	= { 0.746 kw; 42.42 Btu per minute; 33,000 ft-lb per minute; 550 ft-lb per second
1 boiler horsepower	= { 33,475 Btu per hour; 9.808 kw

Weight and Volume

1 gal (U. S.)	= { 231 cu in.; 0.1337 cu ft
1 British or Imperial gallon	= 277.42 cu in.
1 cu ft	= { 7.48 gal; 1728 cu in.
1 cu ft water at 60 F	= 62.37 lb
1 cu ft water at 212 F	= 59.83 lb
1 gal water at 60 F	= 8.34 lb
1 gal water at 212 F	= 7.998 lb
1 lb (avdp)	= { 16 oz; 7000 grains
1 bushel	= 1.244 cu ft
1 short ton	= 2000 lb

Pressure

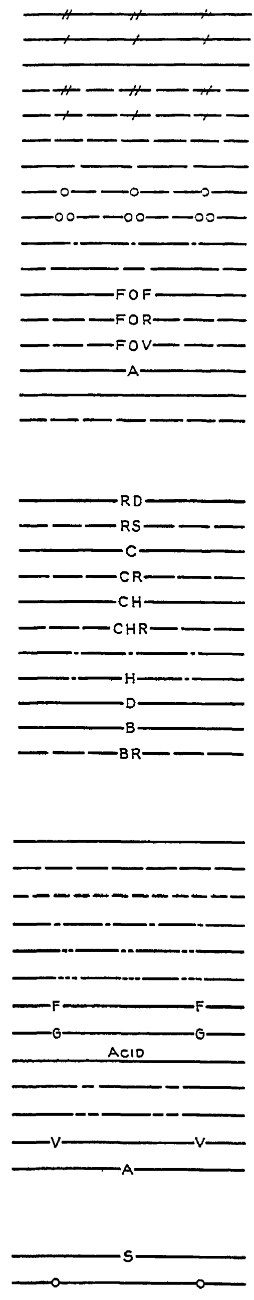
1 lb per square inch	=	144 lb per square foot 2.0421 in. mercury at 62 F 2.309 ft water at 62 F 27.71 in. water at 62 F
1 oz per square inch	=	0.1276 in. mercury at 62 F 1.732 in. water at 62 F
1 atmosphere	=	14.6959 lb per square inch 2117 lb per square foot 33.9 ft water at 62 F 30 in. mercury at 62 F 29.921 in. mercury at 32 F
1 in. water at 62 F	=	0.03609 lb per square inch 0.5774 oz per square inch 5.196 lb per square foot
1 ft water at 62 F	=	0.433 lb per square inch 62.35 lb per square foot
1 in. mercury at 62 F	=	0.491 lb per square inch 7.84 oz per square inch 1.131 ft water at 62 F 13.58 in. water at 62 F

Metric Units

1 cm	=	0.3937 in.
1 in.	=	2.540 cm
1 m	=	3.281 ft
1 ft	=	0.3048 m
1 sq cm	=	0.155 sq in.
1 sq in.	=	6.452 sq cm
1 sq m	=	10.76 sq ft
1 sq ft	=	0.0929 sq m
1 cu cm	=	0.06102 cu in.
1 cu in.	=	16.39 cu cm
1 cu m	=	35.31 cu ft
1 cu ft	=	0.02831 cu m
1 liter	=	1000 cu cm = 0.2642 gal
1 kg	=	2.205 lb (avdp)
1 lb	=	0.4536 kg
1 metric ton	=	2205 lb (avdp)
1 gram	=	0.002205 lb (avdp)
1 kilometer per hour	=	0.6214 mph
1 gram per square centimeter	=	0.02905 in. mercury at 62 F 0.3944 in. water at 62 F
1 kg per sq cm (metric atmosphere)	=	14.22 lb per square inch
1 gram per cubic centimeter	=	0.03613 lb per cubic inch 62.43 lb per cubic foot
1 dyne	=	0.00007233 poundals
1 joule	=	10,000,000 ergs 0.7376 ft-lb
1 metric horsepower	=	75 kg-m per second 0.986 hp (U. S.)
1 kilogram-calorie per kilogram	=	1.8 Btu per pound
1 gram-calorie per square centimeter	=	3.687 Btu per square foot
1 gram-calorie per square centimeter per centimeter	=	1.452 Btu per sq ft per inch
1 gram-calorie per second per square centimeter for a temperature gradient of 1 deg C per centimeter.	=	2903 Btu per hour per square foot for a temperature gradient of 1 deg F per inch of thickness.

GRAPHICAL SYMBOLS FOR DRAWINGS

Piping



HEATING

1. High Pressure Steam
2. Medium Pressure Steam
3. Low Pressure Steam
4. High Pressure Return
5. Medium Pressure Return
6. Low Pressure Return
7. Boiler Blow Off
8. Condensate or Vacuum Pump Discharge
9. Feedwater Pump Discharge
10. Make Up Water
11. Air Relief Line
12. Fuel Oil Flow
13. Fuel Oil Return
14. Fuel Oil Tank Vent
15. Compressed Air
16. Hot Water Heating Supply
17. Hot Water Heating Return

AIR CONDITIONING

18. Refrigerant Discharge
19. Refrigerant Suction
20. Condenser Water Flow
21. Condenser Water Return
22. Circulating Chilled or Hot Water Flow
23. Circulating Chilled or Hot Water Return
24. Make Up Water
25. Humidification Line
26. Drain
27. Brine Supply
28. Brine Return

PLUMBING

29. Soil, Waste or Leader (Above Grade)
30. Soil, Waste or Leader (Below Grade)
31. Vent
32. Cold Water
33. Hot Water
34. Hot Water Return
35. Fire Line
36. Gas
37. Acid Waste
38. Drinking Water Flow
39. Drinking Water Return
40. Vacuum Cleaning
41. Compressed Air

SPRINKLERS

42. Main Supplies
43. Branch and Head
44. Drain

GRAPHICAL SYMBOLS FOR DRAWINGS *Ductwork*

45. Duct (1st Figure, Width; 2nd, Depth)

46. Direction of Flow

47. Inclined Drop in Respect to Air Flow

48. Inclined Rise in Respect to Air Flow

49. Supply Duct Section

50. Exhaust Duct Section

51. Recirculation Duct Section

52. Fresh Air Duct Section

53. Other Duct Sections

54. Register

55. Grille

56. Supply Outlet

57. Exhaust Inlet

58. Top Register or Grille

59. Center Register or Grille

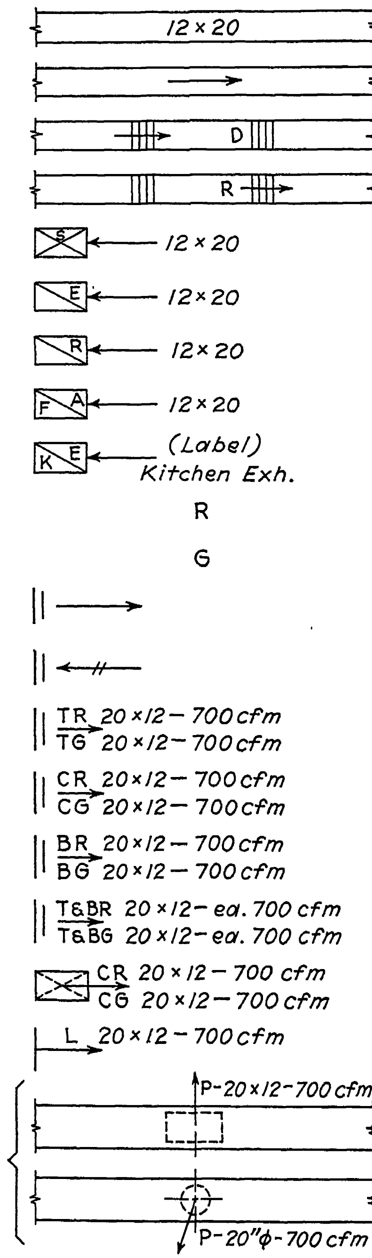
60. Bottom Register or Grille

61. Top and Bottom Register or Grille

62. Ceiling Register or Grille

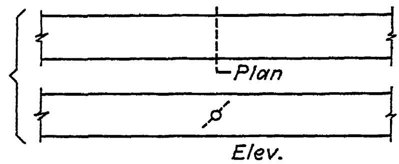
63. Louver Opening

64. Adjustable Plaque

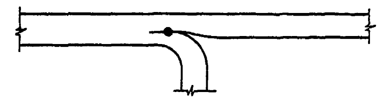


Graphical Symbols for Drawings *Ductwork*

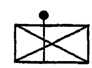
65. Volume Damper



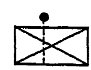
66. Deflecting Damper



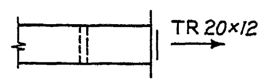
67. Deflecting Damper, Up



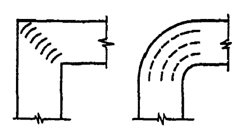
68. Deflecting Damper, Down



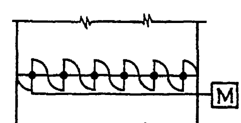
69. Adjustable Blank Off



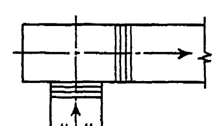
70. Turning Vanes



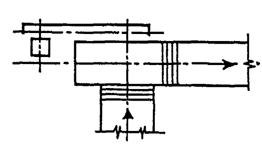
71. Automatic Dampers



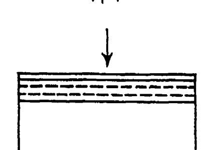
72. Canvas Connections



73. Fan and Motor With Guard

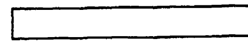


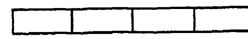
74. Intake Louvers and Screen



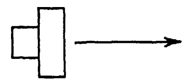
Graphical Symbols for Drawings

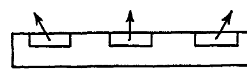
Heating and Ventilating

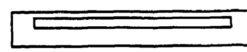
75. Heat Transfer Surface, Plan


76. Wall Radiator, Plan


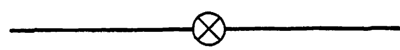
77. Wall Radiator on Ceiling, Plan

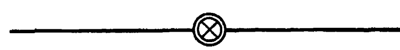

78. Unit Heater (Propeller), Plan


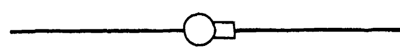
79. Unit Heater (Centrifugal Fan), Plan


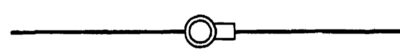
80. Unit Ventilator, Plan


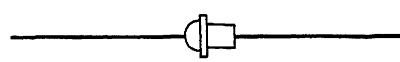
Traps

81. Thermostatic


82. Blast Thermostatic


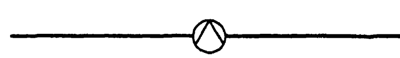
83. Float and Thermostatic


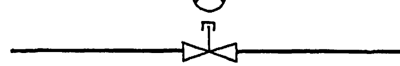
84. Float


85. Boiler Return


Valves

86. Reducing Pressure


87. Air Line


88. Lock and Shield


89. Diaphragm


90. Air Eliminator

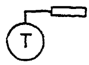

91. Strainer


92. Thermometer


93. Thermostat


Graphical Symbols for Drawings *Refrigerating*

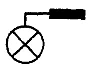
94. Thermostat (Self Contained)

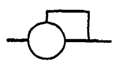

95. Thermostat (Remote Bulb)


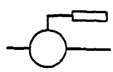
96. Pressurestat

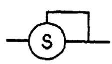

97. Hand Expansion Valve

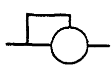

98. Automatic Expansion Valve


99. Thermostatic Expansion Valve


100. Evaporator Press. Regulating Valve, Throttling Type


101. Evaporator Press. Regulating Valve, Thermostatic Throttling Type


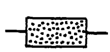
102. Evaporator Press. Regulating Valve, Snap-Action Valve


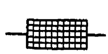
103. Compressor Suction Pressure Limiting Valve, Throttling Type


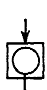
104. Hand Shut Off Valve


105. Thermal Bulb

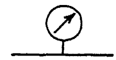

106. Scale Trap

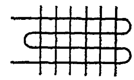

107. Dryer


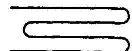
108. Strainer


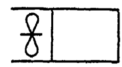
109. High Side Float


110. Low Side Float

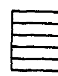

111. Gage


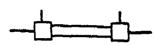
112. Finned Type Cooling Unit, Natural Convection


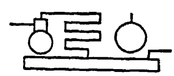
113. Pipe Coil


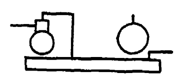
114. Forced Convection Cooling Unit


115. Immersion Cooling Unit

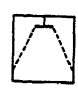

116. Ice Making Unit


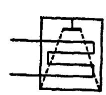
117. Heat Interchanger


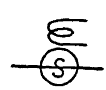
118. Condensing Unit, Air Cooled


119. Condensing Unit, Water Cooled


120. Compressor


121. Cooling Tower


122. Evaporative Condenser


123. Solenoid Valve


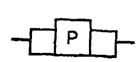
124. Pressurestat With High Pressure Cut-Out


Table 1. Specific Heat of Solids

Materials	Temperature F	Specific Heat	Authority
Alloys			
Brass, Red	32	0 0899	S
Brass, Yellow	32	0 0883	S
Bronze (80*Cu*, 20*Sn*)	57–208	0.0862	S
Monel Metal	68–2370	0 127	S
Aluminum	80–212	0.212	S
Asbestos	68–208	0 195	S
Brickwork		0.195	H
Carbon (Graphite)	104–1637	0 314	I
Coal		0.278	H
Coke		0 201	H
Concrete		0.270	H
Copper	64–212	0.0928	S
Fire Clay Brick	77–1832	0.258	I
Glass			
Crown	50–122	0 161	S
Flint	50–122	0 117	S
Gold	64	0.0312	S
Gypsum		0 259	H
Ice	32	0.487	S
Ice	−40	0.434	S
Iron, Pure	32	0.1043	S
Iron, Pure	32–600	0.127	M
Iron, Cast	68–212	0.1189	H
Iron, Wrought	59–212	0.1152	H
Lead	32	0.0297	S
Nickel	32	0.1032	S
Masonry		0.2159	H
Plaster		0 2	H
Platinum	58–212	0.0319	S
Rocks			
Gneiss	63–210	0.196	S
Granite	54–212	0.192	S
Limestone	59–212	0.216	S
Marble	32–212	0.21	S
Sandstone		0.22	S
Silver	32	0 0536	S
Steel		0.1175	H
Sulphur	240–320	0.220	S
Silica Brick	77–1832	0.263	I
Tin	77	0.0548	S
Woods (Average)	68	0.327	S
Zinc	32	0.0913	S

Table 2. Specific Heat of Liquids

Liquid	Temperature F	Specific Heat	Authority
Alcohol, Ethyl	32	0 548	S
Alcohol, Methyl	59–68	0.601	S
Glycerine	59–122	0 576	S
Lead (Molten)	360	0.041	H
Mercury	68	0.03325	S
Petroleum	70–136	0.511	S
Sea Water			
Sp. Gr. 1.0043	64	0.980	S
Sp. Gr. 1.0463	64	0.903	S
Water	59	1.000	S

Table 3. Specific Heat of Gases and Vapors

Substance	Temperature F	Specific Heat at Constant Pressure	Ratio of Specific Heat C_p/C_v	Specific Heat at Constant Volume (Computed)	Authority
Air	32–392	0.2375	1.405	0.169	S
Ammonia	80–392	0.5356	1.277	0.419	S
Carbon Dioxide	52–417	0.2169	1.3003	0.1668	S
Carbon Monoxide	79–388	0.2426	1.395	0.1736	S
Coal Gas	68–1900	0.3145			S
Flue Gas		0.24 (Approx.)			H
Hydrogen	70–212	3.41	1.419	2.402	S
Nitrogen	32–392	0.2438	1.41	0.1729	S
Oxygen	55–404	0.2175	1.3977	0.155	S
Water Vapor	212	0.421	1.305	0.322	S
Water Vapor	356	0.51			S

Notes: When one temperature is given the true specific heat is given, otherwise the value is the mean specific heat between the given limits.

Authorities: S—Smithsonian Physical Tables, 1933; I—International Critical Tables; H— Heating, Ventilation and Air Conditioning, by L. A. Harding and A. C. Willard; M—Engineers' Handbook, by Lionel S. Marks.

TABLE 4. CIRCUMFERENCES AND AREAS OF CIRCLES

Diameter in Inches	Area		Circumference		Diameter in Inches	Area		Circumference	
	Sq In.	Sq Ft	Inches	Feet		Sq In.	Sq Ft	Inches	Feet
¼	0.049	0.0003	0.785	0.0652	28	615.8	4.276	87.97	7.330
½	0.196	0.0014	1.571	0.1309	28½	637.9	4.430	89.54	7.462
¾	0.442	0.0031	2.356	0.1964	29	660.52	4.587	91.11	7.592
1	0.785	0.0054	3.142	0.2618	29½	683.5	4.747	92.63	7.725
1¼	1.227	0.0085	3.927	0.3273	30	706.8	4.909	94.25	7.854
1½	1.767	0.0123	4.712	0.3927	31	754.8	5.241	97.39	8.116
1¾	2.405	0.0167	5.498	0.4582	32	804.3	5.585	100.5	8.378
2	3.142	0.0218	6.283	0.5236	33	855.3	5.940	103.7	8.639
2¼	3.976	0.0276	7.069	0.5891	34	907.9	6.305	106.8	8.901
2½	4.909	0.0341	7.854	0.6546	35	962.1	6.681	109.9	9.163
2¾	5.939	0.0412	8.639	0.7200	36	1018.0	7.069	113.1	9.425
3	7.069	0.0491	9.425	0.7854	37	1075.0	7.467	116.2	9.686
3¼	8.296	0.0576	10.21	0.8510	38	1134.0	7.876	119.4	9.948
3½	9.621	0.0668	10.99	0.9160	39	1195.0	8.296	122.5	10.21
3¾	11.04	0.0767	11.78	0.9818	40	1256.0	8.727	125.6	10.47
4	12.57	0.0873	12.57	1.047	41	1320.0	9.168	128.8	10.73
4¼	14.19	0.0986	13.35	1.113	42	1385.0	9.621	131.9	10.99
4½	15.90	0.1104	14.14	1.178	43	1452.0	10.08	135.1	11.26
4¾	17.72	0.1231	14.92	1.243	44	1521.0	10.56	138.2	11.52
5	19.64	0.1364	15.71	1.309	45	1590.0	11.04	141.4	11.78
5¼	21.65	0.1504	16.49	1.374	46	1662.0	11.54	144.5	12.04
5½	23.76	0.1650	17.28	1.440	47	1735.0	12.05	147.7	12.30
5¾	25.97	0.1840	18.06	1.505	48	1810.0	12.51	150.8	12.57
6	28.27	0.1964	18.85	1.571	49	1886.0	13.09	153.9	12.83
6¼	30.68	0.2131	19.64	1.637	50	1963.0	13.64	157.1	13.09
6½	33.18	0.2304	20.42	1.702	51	2043.0	14.19	160.2	13.35
6¾	35.79	0.2486	21.21	1.768	52	2124.0	14.75	163.4	13.61
7	38.49	0.2673	21.99	1.833	53	2206.0	15.32	166.5	13.88
7¼	41.28	0.2867	22.78	1.899	54	2290.0	15.90	169.6	14.14
7½	44.18	0.3068	23.56	1.964	55	2376.0	16.50	172.8	14.40
7¾	47.17	0.3276	24.35	2.029	56	2463.0	17.10	175.9	14.66
8	50.27	0.3491	25.13	2.094	57	2552.0	17.72	179.1	14.92
8¼	53.46	0.3713	25.92	2.160	58	2642.0	18.35	182.2	15.18
8½	56.75	0 3942	26.70	2.225	59	2734.0	18.99	185.4	15.45
8¾	60.13	0.4175	27.49	2.291	60	2827.0	19.63	188.5	15.71
9	63.62	0.4418	28.27	2.356	61	2922.0	20.29	191.6	15.97
9¼	67.20	0.4668	29.06	2.422	62	3019.0	20.97	194.8	16.23
9½	70.88	0.4923	29.85	2.488	63	3117.0	21.65	197.9	16.49
9¾	74.66	0.5185	30.63	2.553	64	3217.0	22.34	201.1	16.76
10	78.54	0.5454	31.42	2.618	65	3318.0	23.04	204.2	17.02
10½	86.59	0.6010	32.99	2.750	66	3421.0	23.76	207.3	17.28
11	95.03	0.6600	34.56	2.880	67	3526.0	24.48	210.5	17.54
11½	103.9	0.7215	36.13	3.011	68	3632.0	25.22	213.6	17.80
12	113.1	0.7854	37.70	3.142	69	3739.0	25.97	216.8	18.06
12½	122.7	0 8520	39.27	3.273	70	3848.0	26.73	219.9	18.33
13	132.7	0.9218	40.84	3.403	71	3959.0	27.49	223.1	18.59
13½	143.1	0.9937	42.41	3.535	72	4072.0	28.27	226.2	18.85
14	153.9	1.069	43.98	3.665	73	4185.0	29.07	229.3	19.11
14½	165.1	1.146	45.55	3.796	74	4301.0	29.87	232.5	19.37
15	176.7	1.227	47.12	3.927	75	4418.0	30.68	235.6	19.63
15½	188.7	1.310	48.69	4.058	76	4536.0	31.50	238.8	19.90
16	201.1	1.396	50.27	4.189	77	4657.0	32.34	241.9	20.16
16½	213.8	1.485	51.84	4.321	78	4778.0	33.18	245.0	20.42
17	226.9	1.576	53.41	4.451	79	4902.0	34.04	248.2	20.68
17½	240.5	1.670	54.98	4.582	80	5027.0	34.91	251.3	20.94
18	254.5	1.767	56.55	4.712	81	5153.0	35.78	254.5	21.21
18½	268.8	1.867	58.12	4.845	82	5281.0	36.67	257.6	21.47
19	283.5	1.969	59.69	4.974	83	5411.0	37.57	260.8	21.73
19½	298.6	2.074	61.26	5.105	84	5542.0	38.48	263.9	21.99
20	314.2	2.182	62.83	5.236	85	5675.0	39.41	267.0	22.25
20½	330.1	2.293	64.40	5.367	86	5809.0	40.34	270.2	22.51
21	346.4	2.405	65.97	5.498	87	5945.0	41.28	273.3	22.78
21½	361.1	2.508	67.54	5.629	88	6082.0	42.24	276.5	23.04
22	380.1	2.640	69.12	5.760	89	6221.0	43.20	279.6	23.30
22½	397.6	2.761	70.69	5.891	90	6362.0	44.18	282.7	23.56
23	415.5	2.885	72.26	6.021	91	6504.0	45.17	285.9	23.82
23½	433.7	3.012	73.83	6.153	92	6648.0	46.16	289.0	24.09
24	452.4	3.142	75.40	6.283	93	6793.0	47.17	292.2	24.35
24½	471.4	3.274	76.97	6.415	94	6940.0	48.19	295.3	24.61
25	490.9	3.409	78.54	6.545	95	7088.0	49.22	298.4	24.87
25½	510.7	3.547	80.11	6.676	96	7238.0	50.27	301.6	25.13
26	530.9	3.687	81.68	6.807	97	7390.0	51.32	304.7	25.39
26½	551.6	3.832	83.25	6.938	98	7543.0	52.38	307.9	25.66
27	572.6	3.976	84.82	7.069	99	7698.0	53.46	311.0	25.92
27½	593.9	4.125	86.39	7.199	100	7854.0	54.54	314.2	26.18

CATALOG DATA SECTION

The HEATING VENTILATING AIR CONDITIONING GUIDE 1942

INDEX TO ADVERTISERS
PAGE 859

INDEX TO MODERN EQUIPMENT
PAGE 1137

TWENTIETH ANNIVERSARY EDITION

1922-1942

In 1922, when the first edition of THE GUIDE was published, a group of 80 leading manufacturers provided equipment data of great value to the 5000 engineers, architects, contractors, and others in the profession and the industry who received this reference volume. It is noteworthy that in every succeeding edition—now the Twentieth Anniversary Edition—14 of the firms originally using THE GUIDE have continued their service to its readers. Here is the Honor Roll:

American Blower Corporation
American Radiator & Standard Sanitary Corporation
Buffalo Forge Company
Carrier Corporation
C. A. Dunham Company
Hoffman Specialty Company
Johnson Service Company
Kewanee Boiler Corporation
Nash Engineering Company
Powers Regulator Company
Sarco Company, Incorporated
The Trane Company
United States Radiator Corporation
Warren Webster & Company

ACKNOWLEDGEMENT

For their valued assistance in the development and expansion of THE GUIDE to its present place as the recognized authority in the field of heating, ventilating and air conditioning, the Guide Publication Committee offers sincere thanks to these manufacturers—and to the 162 others who participate in this annual edition. Annually, since 1937, more than 10,000 men in the profession and the industry procure THE GUIDE and utilize the information supplied by these manufacturers.

HEATING VENTILATING AIR CONDITIONING GUIDE, 1942

ALPHABETICAL INDEX TO ADVERTISERS

A

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Y

AIR CONDITIONING

Equipment for complete air conditioning systems, consisting of an assembly of apparatus for air circulation, air cleaning and heat transfer, with control apparatus for maintaining temperature and humidity within prescribed limits, has many commercial, comfort, and industrial applications. Systems for all-year, winter and summer service, and special processing work are presented in four divisions . . . Pages 867-916.

CENTRAL SYSTEMS (p. 867-877)

Complete assembly of supply and return ducts serving one or more spaces, connected with some or all of the following equipment: fans, motors, heat transfer surfaces, humidifiers, dehumidifiers, refrigeration machinery, air cleaning devices and control equipment.

An outline of the design procedure generally used to create a modern central air conditioning system is given in Chapter 21 of the Technical Data Section.

DIRECT FIRED UNITS (p. 878-887)

Automatic heating and comfort air conditioning apparatus suitable for residential and small commercial applications designed to give results similar to the larger central systems provide direct fired oil, gas or coal heating units, filtration, fan controls, etc.

The Technical Data Section, Chapters 10, 12 and 20 cover this type of equipment.

FAN-FURNACE SYSTEMS (p. 888-891)

Winter air conditioning and summer ventilation for residences are provided by Automatic fired fan-furnace systems. As in the larger central systems these installations clean, heat and humidify the air, and if desired, auxiliary units will provide cooling.

In Chapter 20 on Mechanical Warm Air Furnace Systems will be found details of the design of this type of system.

UNIT HEATERS, COOLERS (p. 892-916)

For complete or partial air conditioning there are a variety of self-contained units. Such units may be complete in themselves, employing their own direct means of air cleaning, heating distribution and source of refrigeration.

The various functional elements of unitary equipment are given in Chapters 22 and 23, for Unit Heaters, Ventilators, Humidifiers, Conditioning and Cooling Units and Attic Fans.

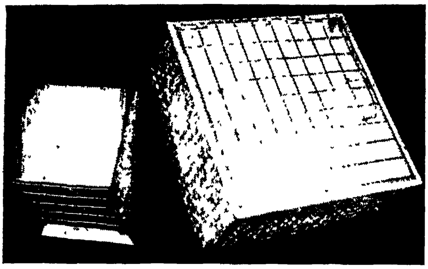
Manufacturer's products shown in this division are designed for specific applications. Consult the Index to Modern Equipment for additional products of these manufacturers.

Air & Refrigeration Corporation

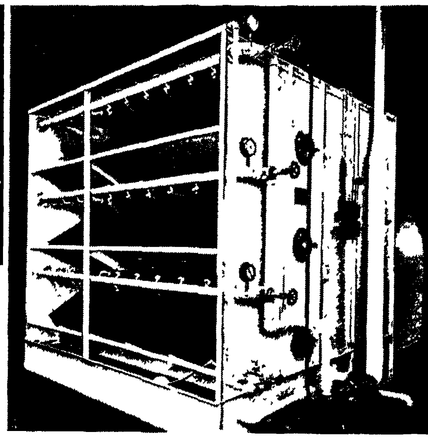
475 Fifth Avenue, **New York City**

Atlanta, Ga. **Detroit, Mich.**

CAPILLARY AIR CONDITIONERS



A standard Capillary cell with cut-away section showing oriented glass filaments. Size: 20 in. x 20 in. x 8 in.



A standard Size 6-5 Class I Capillary Central Station Installation.

Every Air Conditioning Engineer and all Industrial Engineers responsible for air conditioning should be familiar with the uses of this advanced equipment.

The standard Capillary cell is the basic element in all Capillary conditioners. The patented arrangement of glass filaments, essentially parallel to the flow of air and water through the cell, accounts for the highly efficient heat transfer between air and water. At the same time, the cells act as an efficient air cleaner and the arrested dirt is continuously flushed from the cell.

As a simple air washer, humidifier or evaporative cooler, Class I Capillary conditioners call for the recirculation of only 3 gpm per 1000 cfm distributed over the cells at 6 lbs nozzle pressure. The saturation efficiency is 97 per cent. Less efficient spray washers require 15 gpm or more at 20 lbs nozzle pressure.

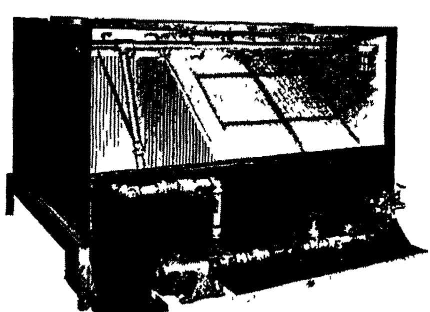
A single stage of Capillary cells equals or exceeds in cooling and dehumidifying capacity a 2 bank spray type dehumidifier.

Increased cleaning efficiency and an approach of less than 1 deg F between leaving air and leaving water is obtained through a Class II Capillary wherein the water flows counter to the air through the cell.

A 2-stage Capillary Class I-II offers true counterflow performance with leaving cooling water temperature exceeding that of leaving air.

Where a closed system for the cooling medium is required (direct expansion, brine or cold water), a Class III Capillary is offered with suitable coils after the Capillary cells. No filters are required. Coils are kept clean and evaporative cooling is available whenever entering wet-bulb conditions permit.

Capillary conditioners of all classes are made in central station units ranging from 2200 cfm to 132,000 cfm or larger. Assembled units including fans, heaters, coils, pump, insulated casing, etc., suitable for suspension or floor mounting range from 4000 to 16,000 cfm.



A standard Size 3-4 Capillary unit air conditioner complete in insulated casing. Capacity 16,000 cfm.

Submit design and capacity conditions to receive specific recommendations or write for complete catalogs and engineering data.

American Blower Corporation

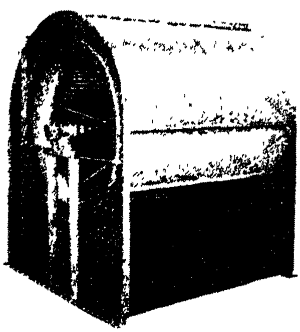
Division of American Radiator and Standard Sanitary Corporation

General Offices and Factory

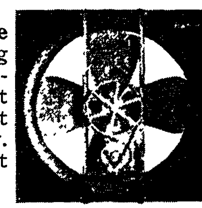
Detroit, Mich.

Branches in All Principal Cities

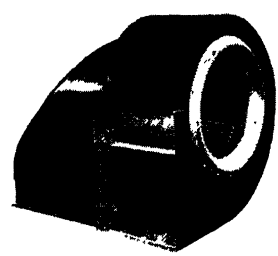
AIR CONDITIONING — HUMIDIFYING — DEHUMIDIFYING — COOLING — VENTILATING — HEATING — VAPOR-ABSORPTION — DRYING — AIR WASHING AND PURIFICATION — EXHAUSTING EQUIPMENT AND MECHANICAL DRAFT APPARATUS



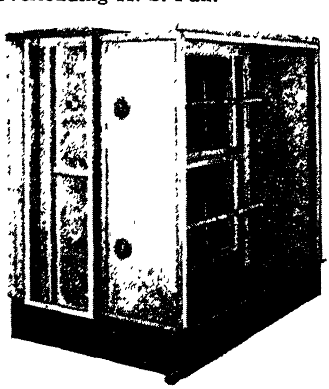
Double Inlet "ABC" Multiblade Fan—above, is a heavy duty ventilating fan. Its wheel has narrow, forward pitched blades. Low tip speeds assure quiet operation. Request Bulletin A-701. Write for Bulletin A-403 for backwardly inclined, non-overloading H. S. Fan.



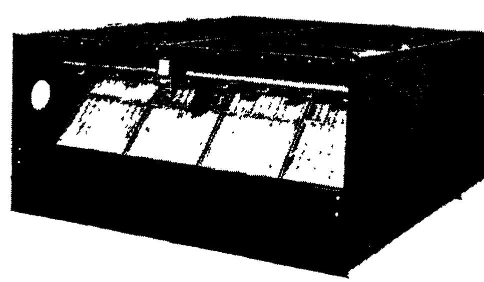
Commercial V-Belt Drive Ventura Fans—for ventilating applications without duct systems, where extremely quiet operation is desired. Inlet-outlet streamlined for high efficiency. Also direct connected. Request Bulletin B-2529.



"ABC" Utility Sets—right, complete packaged units, direct connected or V-Belt short coupled drive, for duct applications. Famous "ABC" Multiblade Wheel operates at low tip speeds. Quiet, compact. Bulletin B-2529.



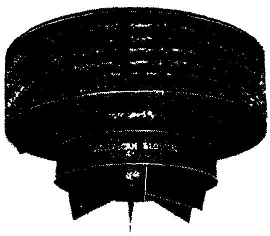
American Blower Air Washer—above, cleans, purifies and freshens the air, removes dust, odors and bacteria, cools if desired and provides an effective method of controlling humidity. Bulletin 3623.



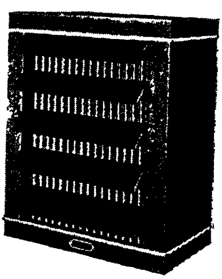
American Blower Capillary Air Washers—above, for high efficiency in cleaning, humidification, cooling and dehumidification of air. A highly efficient surface contact mechanism, the capillary cell, is used. Air is forced at low resistance through long, irregular passages of small size formed by a large amount of thoroughly wetted glass surface. Unit includes a substantial metal casing and tank of air washer design, capillary cells, improved low head sprays, metal or glass fibre low resistance moisture eliminators, non-ferrous, extended surface cooling or heating coils. Write for Bulletin 3723.

TYPES OF AMERICAN BLOWER CORPORATION AIR HANDLING AND CONDITIONING EQUIPMENT

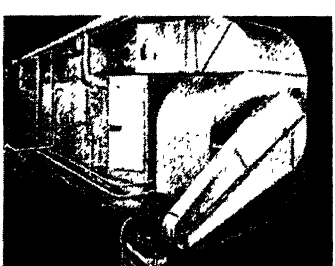
All types of air handling and air conditioning equipment for industrial applications, process work, drying, cooling; also equipment for stores, offices, shops, public buildings, power plants, etc., and attic ventilation for homes.



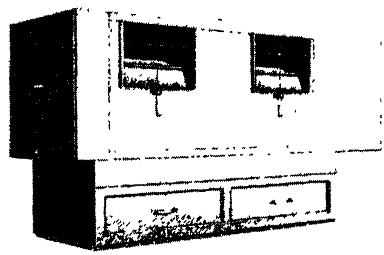
"ABC" Vertical Heaters—for ceiling applications, give an even, wide floor area distribution of heat. For either steam or hot water heating systems. Variable speed, 2-speed and constant speed models. Write for Bulletin A-9418.



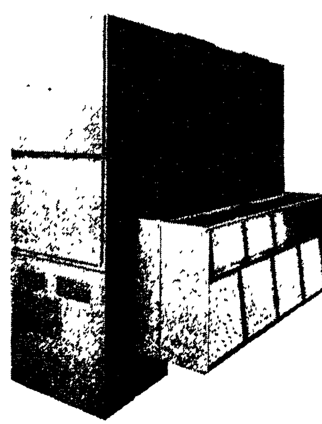
Venturafin Unit Heaters—for many general purpose heating jobs. Wall or ceiling mounting. Streamline construction, rugged heating elements. Steam or hot water. Write for Bulletin A-8218.



Air Conditioning Central Systems—provide an effective way of cooling, heating, humidifying, dehumidifying and purifying air in all classes of business and public buildings where a dust system is desirable. Write for Special Data.



"HV" General Purpose Units—with air filters and Aileron control. Ideal wherever attractive, quiet and economical heating and ventilating units are required. Wall, floor or ceiling mounting. Offer great flexibility of design and arrangement to meet specific needs. Write for Bulletin 5927.



American Blower Series "H" Air Conditioners with Sprayed Coils—are usually applied for industrial uses where air washing and evaporative cooling are required. Sprayed coils give cleaner air, cut coil maintenance and refrigeration costs, reduce necessary air volumes, permit use of smaller ducts and grilles. Horizontal or floor types (as shown). Aileron control provides simple method of regulating flow of air from the fans. Write for Bulletin 6027.

Carrier Corporation

HOME OFFICE AND FACTORIES:
SYRACUSE, N. Y.

INTERNATIONAL DIVISION:
SYRACUSE, N. Y.

DEALERS IN PRINCIPAL CITIES

MARINE DIVISION:
405 LEXINGTON AVE.
NEW YORK CITY

BRANCH SALES OFFICES:

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BOSTON
CHICAGO
CINCINNATI
CLEVELAND
DALLAS
DETROIT
KANSAS CITY
LOS ANGELES
NEW ORLEANS
NEW YORK CITY
PHILADELPHIA
ST. LOUIS
SAN FRANCISCO
WASHINGTON

AIR CONDITIONING

Room Air Conditioning and Refrigerating Unit—Self Contained.
Four sizes: Cooling range 5,700 to 14,500 Btu.

Commercial Air Conditioning and Refrigerating Unit—Self Contained.
Four sizes: Cooling range 2½ to 8 tons.

Air Conditioning and Refrigerating Assembly—With ducts.
Four sizes: Cooling range 4 to 18 tons. Air capacity 2000 to 9000 cfm.

Humidifier. Individual Unit Supplementing Heating System.
Two models: Evaporating capacity 2.3 to 5 lb per hour.

Room Ventilating Unit. For double-hung and casement windows.
Two sizes: Air capacity 200 to 480 cfm.

Room Air Conditioning Unit—for Central Station System—Normal Ducts.
Six sizes: Air capacity totals 96 to 1100 cfm.

Room Air Conditioning Unit—for Central Station System—Conduit air Distribution.
Six sizes: Cooling range .4 to 1.3 tons. Heating range 2000 to 34,500 Btu per hour.

Room Air Conditioning Unit—Ductless Supply of hot and chilled water required.
Three sizes: Cooling range .5 to 1 ton. Heating range 3600 to 39,700 Btu per hour.

Room Outlet—for exposed or concealed ducts.
Twenty-two sizes: Air capacity 40 to 4900 cfm per outlet.

Commercial Air Conditioning Units—Suspension—No Ducts.
Six sizes: Cooling range .5 to 5.5 tons. Air capacity 310 to 1850 cfm.

Commercial Air Conditioning Units—Floor Mounted—With Ducts.
Four sizes: Cooling range 2 to 45 tons. Heating range 100,000 and up Btu per hour. Air capacity 700 to 8000 cfm.

Commercial Air Conditioning Units—Suspension—With ducts.
Four sizes: Cooling range 2 to 45 tons. Heating range 100,000 and up Btu per hour. Air capacity 700 to 8000 cfm.

Industrial Air Conditioning Unit. May be installed with or without ducts.
Four sizes: Cooling range 2 to 40 tons. Heating range 30,000 to 750,000 Btu per hour. Air capacity 2000 to 8000 cfm.

Industrial Humidifier—humidifiers, filters, distributes air.
One size: Air capacity 5000 cfm.

Dehumidifier—Cells used singly or in multiple.
Eighteen sizes: Cooling range 6 to 60 tons per cell. Air capacity 2000 to 36,000 cfm per cell.

Dehydration—silica gel. For either residential or industrial applications.
Four sizes: Moisture removal capacity 23 to 126 lbs per hour. Air capacity 460 to 3150 cfm.

Clarage Fan Company

Kalamazoo, Michigan

Sales Engineering Offices in All Principal Cities

(Consult Telephone Directory)

CLARAGE AIR HANDLING AND CONDITIONING EQUIPMENT

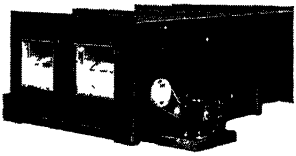
For Over a Quarter-Century Clarage has been a leading manufacturer of air handling and conditioning equipment. There is a Clarage fan or blower, conditioning unit or system to meet every need, from the simplest ventilating or cooling job to the most exacting temperature and humidity control installation.

Whatever your ventilating, unit heating, cooling, drying, air cleaning, humidifying, dehumidifying or complete air conditioning problem, we can meet your requirements successfully and economically.

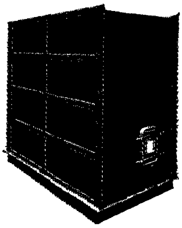
Clarage Experience covers every conceivable type of installation, commercial, industrial and public building. Clarage equipment is used in the largest industrial plants, office buildings, auditoriums, theatres, hotels, restaurants, retail stores, hospitals, churches and schools.

Architects, Engineers and Contractors find our service specially helpful. This Company is an independent manufacturer selling through regular trade channels, and cooperating fully with those who specify and those who install. Your inquiry for data on any Clarage product is invited. Write for Bulletins.

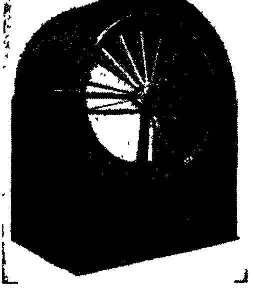
Clarage Systems for complete air conditioning in public buildings and industrial plants.



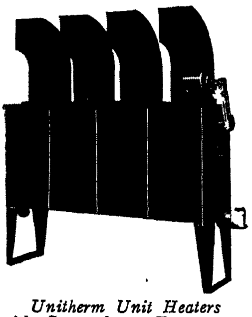
Multitherm Units for complete conditioning, summer cooling, or winter heating.



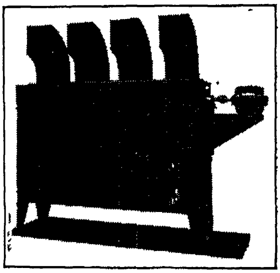
Unicoil Units used in conditioning systems for air cleaning, cooling, heating and humidity control.



Clarage Fan with Vortex (constant speed) Volume Control for ventilation and air conditioning.



Unitherm Unit Heaters with Syncrotherm Temperature Control for factory heating.



Unitherm Unit Coolers for product cooling and refrigeration.

Parks-Cramer Company

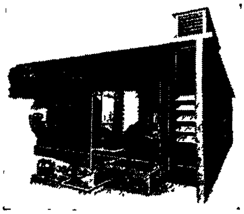
Fitchburg, Mass. **Charlotte, N. C.**

CERTIFIED CLIMATE

Complete Air Conditioning Systems including Heating, Cooling, Humidifying or De-humidifying, Air Changing, Refrigeration, Air Filtering, Air Washing

AUTOMATIC REGULATION

Merrill Process System of Hot Oil Circulation for Heating Industrial Materials



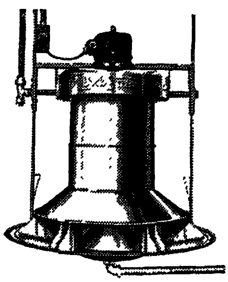
Central Station

Central Station Air Conditioning

Centrally located AIR WASHER. Proper moisture. Positive, pre-determined air removal or re-circulation. Heating coils and refrigeration optional. Helps such industries as Celluloid; Cement; Ceramics; Cereals; Cigars, Cigarettes and Tobacco; Clothing; Confectionery; Glassine; Leather; Paper and Envelopes; Printing and Lithographing; Shoes; Starch and Dextrine; Storage of Perishables; Textiles; Wood Products. Similar installations effective in Hospitals, Art Galleries, Auditoriums, Restaurants.

Air Washer or Central Station Units.

Nozzles for Central Station Air Washers.



High Duty Humidifier

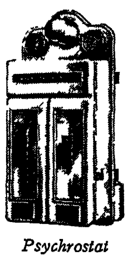
High Duty Humidifier

Water under pressure generates spray. Excess water returns to filter tank and re-circulates. Evaporation per unit high; two sizes of heads each with three sizes of nozzles give flexible capacity for varying conditions. Circulation increased by individual motor-driven fan. Spray thoroughly diffused and distributed over wide area.

Turbomatic Humidifier

(not illustrated)

Efficient humidifier of the atomizer type. For direct humidification, as humidity boosters for Central Station systems of all makes. Self-cleaning.



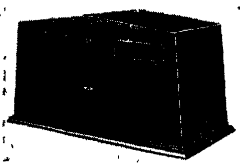
Psychrostat

Parks Automatic Airchanger

For use with High Duty or Turbomatic Humidifiers. Insures fixed humidity and maximum evaporative cooling.

Automatic Regulation

The Psychrostat for accuracy, durability, sensitivity. Employs the principle of the Sling Psychrometer, used in all U. S. Weather Bureau Stations. Hygrostat (not illustrated) where requirements are not so exacting. An Air Conditioning System is no better than its Regulation.



Pettifogger

The Pettifogger

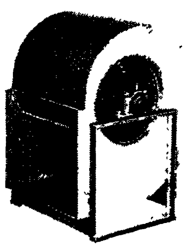
A compact humidifier for offices, stores, storerooms, laboratories, or other isolated departments. Self-contained in lacquered copper casing. Permanently though flexibly connected to water and electrical supplies. Automatic control. Adjustable capacity. Reduces dust. Neutralizes drying effect of heating.

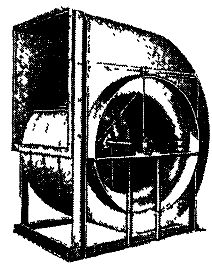
United States Air Conditioning Corporation

Heating, Cooling, Ventilating and Air Conditioning Equipment

For Industrial, Commercial and Residential Applications

General Offices and Factory: Northwestern Terminal,
Minneapolis, Minn.

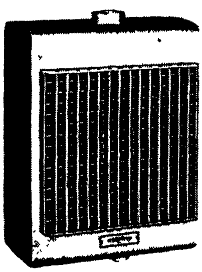


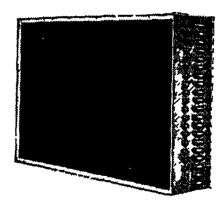


USAirCo Blowers
Heavy and light duty blowers, single or double inlet, in sizes and capacities for any heating, cooling, ventilating and air conditioning application.

USAirCo Air Washers
Single, double or triple stage 2,500 to 100,000 cfm for cleansing, cooling by cold water or refrigerant, humidifying or dehumidifying.

USAirCo Heating Units
Suspended types with Deflecto diffusing grilles. Floor or wall type blower heaters. Sizes and types for every heating need.



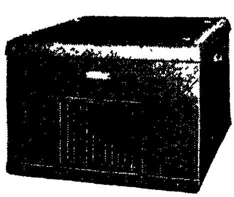


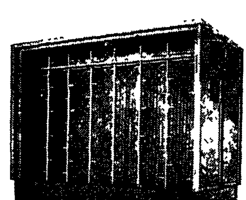
USAirCo Cooling or Heating Cores
Five standard series for central station heating or cooling applications.

USAirCo Cooling Units
Suspended type for cold water or direct expansion applications.

USAirCo Blower Filters
Complete assemblies for warm-air furnace applications.

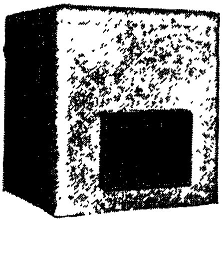
Kooler-aire Package Units
Complete self-contained units for refrigerative, cold water and evaporative cooling. Also room coolers and humidifiers.

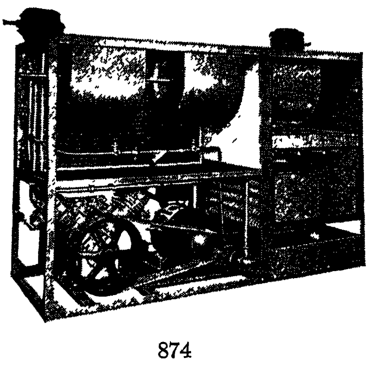


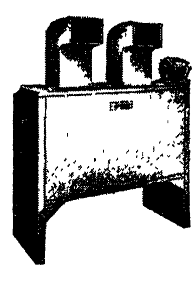


USAirCo Deflecto Grilles
Patented diffusing grilles for controlled directional distribution of air.

Write for Latest USAirCo Catalog





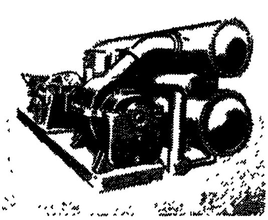


York Ice Machinery Corporation

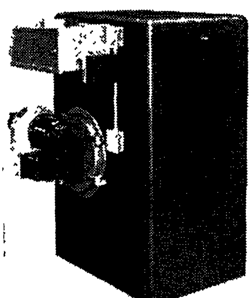
York, Pennsylvania

Factory Branches and Distributor Engineering and Sales Offices throughout the World.

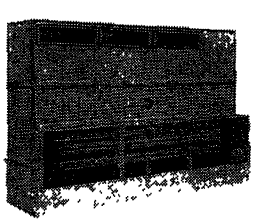
Air Conditioning and Refrigeration for maintaining proper atmospheric conditions for human comfort and industrial processes. Installations of unit and central systems in a complete range of capacities and types for every design requirement.



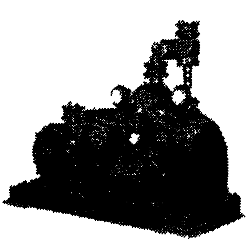
York Turbo Compressor



Yorkaire Heating Unit



York Sectional Economizer



York V-W Condensing Unit

Air Conditioning Units: A complete line of finned coil, dry coil, wetted surface and spray type sectional air conditioners for horizontal or vertical applications, designed to facilitate installation and the distribution of air. Standard units can be equipped with by-pass feature and arranged for cooling and dehumidifying, heating and humidifying, for year-round comfort.

Yorkaire Heat and Winter Air Conditioning—A complete line of York equipment is available for residential or small commercial heating and winter air conditioning installations. Direct fired furnaces, and boilers for steam or hot water can be furnished for burning oil, gas, or coal. Stokers and conversion oil burners complete the catalog. Related apparatus for use with YORKAIRE HEAT units provides complete equipment for year-round air conditioning systems for homes and small business establishments.

Dehumidifiers—For central station systems where a large volume of air is to be handled and where control of humidity is an essential requirement, the York dehumidifier is especially applicable. Construction features insure a minimum space demand and maximum performance conditions. Standard washers are available in a full range of capacities for human comfort or industrial installation. Air washers can be furnished also for use as indoor condensing water cooling towers when specified.

The York Economizer—A combined forced-draft cooling tower and refrigerant condenser, is available for installations where prohibitive water costs or inadequate drainage facilities preclude the use of a water cooled condenser. Standard factory constructed and built-up units may be used singly or in multiple for applications of any specified capacity. Economizers for use with Freon as the refrigerant are furnished, as standard, with a liquid sub-cooling coil.

Condensing and Water Cooling Systems—Standard systems are available for every application requirement up to 1,200 hp capacity using a single compressor. Self-contained units up to 300 hp feature the YORK line. These units are furnished with water cooled condensers or without condensers for economizer applications.

Automatic or manual capacity reduction by-pass valves can be provided for economical operation at reduced load.

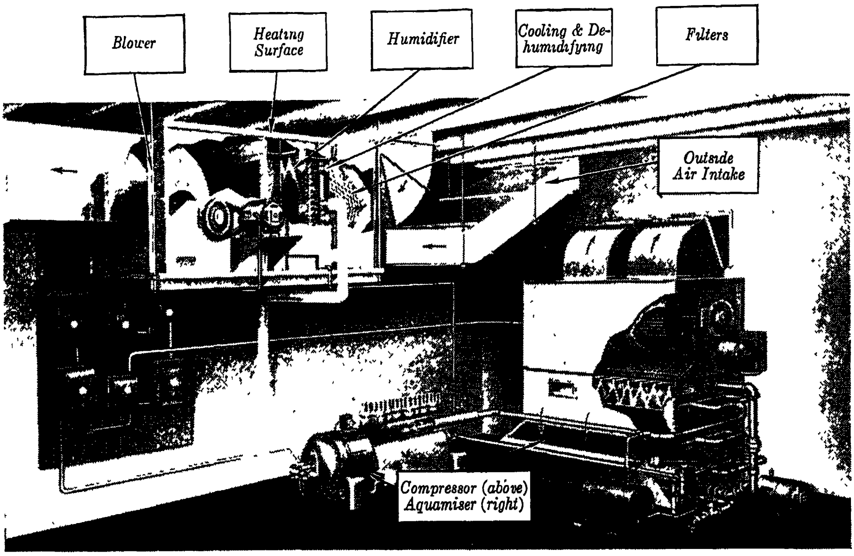
All materials and manufacturing methods employed in the construction of the York Freon Condensing and Water Cooling Systems conform to the high standards and efficient operating characteristics of all YORK products and carry performance guarantees based on *Refrigeration Manufacturers' Association* and *Air Conditioning Manufacturers' Association* ratings.

Westinghouse Electric & Manufacturing Co.

653 Page Blvd., **Springfield, Mass.**

Sales, engineering and service available through Authorized Engineering Contractors in all principal cities

UNIFIED Air Conditioning Equipment for Commercial Building Applications



APPLICATION—Westinghouse equipment provides air conditioning for every building application. Every unit is engineered by Westinghouse and applied in carefully engineered installations.

COMPRESSORS AND CONDENSING UNITS—Hermetically-sealed to conserve power and efficiency. Dust, dirt and trouble sealed out. No seal leaks. All operating parts accessible. Lightweight, compact.

WATER CHILLING UNITS—Available in five sizes from 5 to 110 tons of refrigeration. Highly efficient vertical, semi-flooded type, designed for air conditioning with chilled water.

AIR CONDITIONING UNITS—Incorporate blower, heat transfer surfaces, humidifiers and filters. Horizontal and vertical types.

AQUAMISERS (Evaporative Condensers)—Reduce water consumption 90 per cent to 95 per cent. Sizes to match all compressors in which water consumption is a factor.

Hermetically-Sealed Compressors and Condensing Units*

Type	Hp	Capacity Btu per Hour		Dimensions—Inches†			Approx. Net Weight Lb‡
		A.S.R.E. Rating**	Maximum†	Length	Width	Height	
CLD- 45	1	12000	15700	23½	18	16⅛	235
CLD- 90	2½	26900	41000	21	20½	36½	440
CLD- 135	3¾	41000	60800	21	20½	36½	460
CLD- 205	5	69600	94400	32	22	36	572
CLD- 275	7½	90000	124000	32¼	21¾	32¾	590
CLD- 415	10	120000	163000	34½	22¾	35¼	630
CLS - 550	15	181000	255000	64	19½	40½	1720
CLS - 640	20	222000	302000	80¾	19½	40½	2080
CLS - 850	25	294000	393000	80¾	19½	40½	2080
CLS -1320	40	455000	615000	88⅛	30⅝	44¼	3090
CLS -1980	60	653000	896000	88⅛	30⅝	44¼	3090
CLS -2550	75	915000	1155000	93¾	34	61	6000
CLS -3400	100	1240000	1529000	95¼	34	64	6590

*For 60 Cycle Alternating Current.
**Rating at 37 lb suction, 65°F suction gas temperature, condenser water 75°F entering, 95°F leaving.
†Rating at 50°F evaporating temperature, 25°F superheat and 100 lb per square inch condensing pressure.
‡Dimensions and weights are for complete water-cooled condensing units.

(*See also Page 932*)

Air Conditioning Units

Type	Nominal Capacity Air Delivery Cfm	Dimensions—Inches			Approx. Net Weight Lb
		Length	Width	Height	
AF- 16	480- 1120	58 5/8	29	21	300
AF- 27	810- 1890	50	46	19	400
AF- 37	1110- 2590	50	58	21	525
AH- 55	2580- 3870	72 13/16	79 1/2	27 1/4	612
AV- 55	2580- 3870	48 1/16	79 1/2	62 1/4	615
AH- 83	4080- 6120	81 9/16	88 15/16	33 1/4	794
AV- 83	4080- 6120	56 13/16	88 15/16	74 1/4	826
AH-103	4930- 7394	75 3/4	102 9/16	33 1/4	915
AV-103	4930- 7394	51	102 9/16	74 1/4	951
AH-124	5800- 8700	82 3/4	111 3/4	35 1/4	1127
AV-124	5800- 8700	58	111 3/4	78 1/4	1177
AH-154	7874-11812	93 5/8	112 5/16	42 1/2	1450
AV-154	7874-11812	68 3/4	112 5/16	92 3/4	1550

Aquamisers (Evaporative Condensers)

Type	†Capacity Btu per Hour	Dimensions—Inches			Approx. Net Weight Lb
		Length	Width	Height	
EV- 10	99200	68 1/8	31 1/4	62 1/2	719
EV- 13	117800	68 1/8	31 1/4	62 1/2	832
EV- 20	187000	89 1/8	38 1/4	69 1/2	1266
EV- 25	247000	89 1/8	38 1/4	69 1/2	1376
EV- 35	312600	115 1/8	42 7/8	74 1/2	1863
EV- 40	399600	115 1/8	43 1/4	74 1/2	2065
EV- 60	572700	117 3/8	55 1/4	88 1/2	2931
EV- 80	799200	117 3/8	83 9/16	104	3902
EV- 90	859000	127	62 1/2	130 1/8	4180
EV-120	1145000	128 3/8	82 1/2	132 3/8	5990
EV-150	1433000	131 1/2	102 1/2	134 3/8	6770

†Net refrigerant capacity at 75°F WB entering air and 110°F condensing temperature.

HOW TO SELECT—Fit equipment to meet total Btu load.

WHERE TO BUY—Consult classified telephone directory or nearest Westinghouse district office for name of Authorized Contractor.

SELF CONTAINED SYSTEMS

Mobilaire

Compact, self-contained units for individual room cooling in homes, offices, hotels, apartments. Attractive cabinets of modern design and finish. Details available on request.

Unitaire

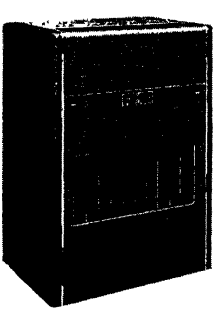
Compact, self-contained units for retail stores, restaurants, and other businesses, also for home use. Available in either "within-the-space" or "central plant" types. Quick economical installation either singly or in combination. Nine sizes.

Unitaire Specifications

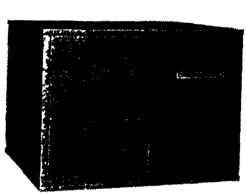
Type	*Hp	Dimensions—Inches			Approx. Net Weight Lb
		Depth	Width	Height	
CU- 45	1	24	38	26	450
SU- 20	2	24	36	45	750
SU- 30	3	24	40	59 3/4	900
SU- 50	5	23 3/4	46 1/2	92 3/8	1455
LU-275	7 1/2	†34	54	69 3/4	1970
LU-415	10	†34	68 7/8	72	2300
LU-550	15	†34	82 1/2	66 1/2	3350
LU-640	20	*34 5/16	100	73 1/4	4000
LU-850	25	‡34 5/16	100	73 1/4	4200

*Net capacity with normal air flow, entering 80 deg DB, 67 deg WB and condenser water inlet 75 F, outlet 95 F.
†Add 14 1/2 in. for overall dimensions with filters.
*Add 21 1/4 in. for overall dimensions with filters.
‡Add 25 1/2 in. for overall dimensions with filters.

TYPICAL UNITAIRE PACKAGED SYSTEMS



Model SU-30
FLOOR TYPE
(3 hp)
(Also available in 2 and 5 hp sizes)

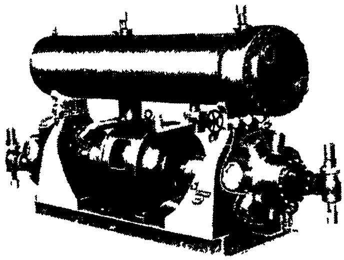


Model CU-45
CEILING TYPE
(1 hp)

Model LU-275
CENTRAL-PLANT TYPE
(7 1/2 hp)
(Also available in 10, 15, 20, and 25 hp sizes)

CHRYSLER AIRTEMP

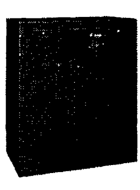
AIRTEMP DIVISION OF CHRYSLER CORPORATION, DAYTON, OHIO



RADIAL COMPRESSOR

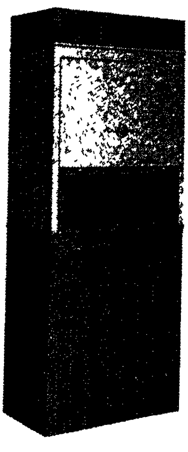
Sizes range from 10 H.P. to 75 H.P. capacity, for use with Freon for air conditioning or heavy duty industrial applications. Airtemp radial compressors are directly connected, have force-feed lubrication. Automatic starting unloader and automatic capacity-reduction unloader give high operating efficiency. Shipped ready to run—easy to install—no special foundation necessary.

CONDENSING UNITS

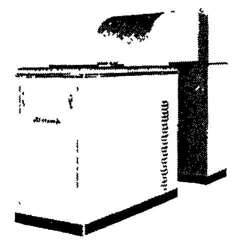


Designed for efficient, low-temperature control. Radial compressor saves weight. Directly connected motor and compressor sealed in oil. Force-feed lubrication and automatic starting unloader. Available in 1½ H.P. to 5 H.P. capacities. Open type units available in ¼ H.P., ⅓ H.P., ½ H.P. and ¾ H.P. sizes.

"PACKAGED" AIR CONDITIONER



Cools, dehumidifies, cleans and circulates the air. Free discharge or duct distribution. Heating coil (steam or hot water) and humidifier can be added to provide year-'round service. Sealed radial compressor has automatic starting unloader. Water-cooled condenser for use with city water or cooling tower. Shipped with all controls—ready to install. Enameled "Bonderized" steel cabinet. Available in 3 and 5 ton capacities.



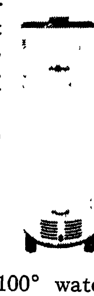
YEAR-'ROUND AIR CONDITIONER WITH WARM AIR FURNACE

This combination heating and cooling unit for homes consists of an Airtemp air-conditioning furnace and a "packaged" cooling unit of 3 hp capacity. It gives complete year-'round air conditioning with same system of ducts, same blower and filters. Heating unit can be any Airtemp winter air conditioner shown on opposite page, except the smallest sizes.

YEAR-'ROUND AIR CONDITIONER WITH BOILER

For home and small store installation, Airtemp "Percolator" boiler and 3 hp or 5 hp packaged air conditioning unit may be connected to provide both heating and cooling. Steam coil and humidifier in cooling unit. Units may be installed together or separated in various convenient arrangements.

VAPORIZING OIL-BURNING WATER HEATER



Uses efficient vaporizing burner with a special, economical pilot flame. Completely insulated. Thermostatically controlled. 20-gallon size provides plenty of hot water for average family. 30- and 45-gallon sizes also available. No gas or electrical connection necessary. Installs anywhere. Heater is approved by Underwriters' Laboratories, Inc.—the mark of safety. All sizes have a 30-gallon-per-hour heating capacity, based on 100° water temperature rise. Economy pilot flame.

CHRYSLER AIRTEMP

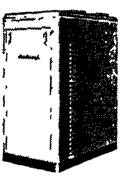
AIRTEMP DIVISION OF CHRYSLER CORPORATION, DAYTON, OHIO

VAPORIZING OIL-BURNING WINTER AIR CONDITIONER

55,000 Btu output. Completely automatic. Models for gravity—forced-air or complete winter air conditioning. Sure-Draft fan assures highest overall efficiency. Bonderized and insulated jacket. Approved for closet installation, Underwriters Laboratories, Inc.

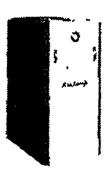
OIL-FIRED WINTER AIR CONDITIONER

Heats, humidifies, filters and circulates the air. Five models, from 70,000 to 160,000 Btu output. "Bonderized" and insulated jacket. Metal combustion chamber, seam-welded firebox of copper-bearing steel; large, slow-speed, rubber-mounted fan. Airtemp conventional or Twin Airflow oil burners on larger models.



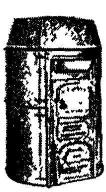
GAS-FIRED WINTER AIR CONDITIONER

Heats, humidifies, filters and circulates the air. Steel-firebox models from 70,000 to 130,000 Btu output. Cast-iron firebox models from 50,000 to 200,000 Btu output. "Bonderized" and insulated jacket. The Airtemp "Silent Flame" Gas Burner starts, stops and operates quietly, has many exclusive features—no popping or flash-backs. Approved, A.G.A. Laboratories.



PERCOLATOR BOILERS—OIL- OR GAS-FIRED

Three oil-fired models from 400 to 1700 E.D.R. (steam). Three gas-fired models from 400 to 1050 E.D.R. (steam). Fire chamber surrounded with water on all sides and bottom. High efficiency and faster heating result from "percolator" principle. Steam or hot water. "Bonderized" and insulated jacket.



COAL FIRED FURNACES—FORCED-AIR—GRAVITY

Forced-air models in cast iron and steel. Sizes from 226 to 346 sq in. grate area in steel—and from 258 to 384 sq in grate area in cast iron. Oversized blower motor mounted in rubber, has automatic overload and low-voltage protection. Gravity models in cast iron and steel—sizes from 226 to 452 sq in. grate area, steel; 204 to 452 sq in. grate area, cast iron.

OIL BURNERS

Model A-8: A conventional oil burner, .75 to 1.35 gallons No. 3 furnace oil per hour.

Model B-9: The exclusive Airtemp Twin Airflow Oil Burner, 1.35 to 3.0 gallons No. 3 furnace oil per hour. Air for combustion is furnished by two fans and adjusted at the outlet, not the inlet of the fans.

Model B-10 and C-10: A conventional pressure-atomizing oil burner, 1.35 to 4.5 gallons No. 3 furnace oil per hour.

All Airtemp Oil Burners are approved by Underwriters' Laboratories, Inc., and bear the seal of the Official Inspection Agency of the Oil Burner Industry as evidencing compliance with commercial Standard CS75-39, as issued by the National Bureau of Standards of the U. S. Department of Commerce.

Delco Appliance Division

General Motors Corporation

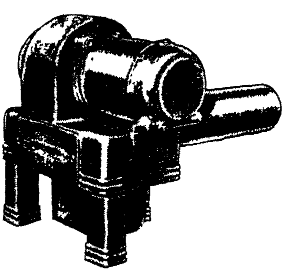
BACKED BY
BUILT AND
GM
GENERAL MOTORS

Rochester, N. Y.

Delco Offers a Complete Line of Automatic Heating Products, including: Oil burners, bituminous coal stokers, oil-fired boilers, oil and gas-fired winter air conditioners, automatic water heaters, thermostats and master controls. Write to Delco Appliance Division, General Motors Corporation, Rochester, N. Y., for latest information and detailed specifications, or consult your local Delco-Heat distributor whose address is listed in the classified section of your telephone directory.

DELCO AUTOMATIC HEAT

OIL BURNERS

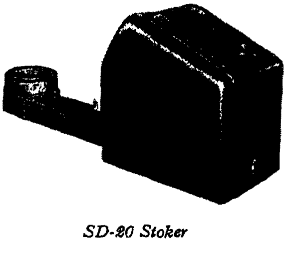


Model "A" Oil Burner with Rotopower Unit

Delco Oil Burners employ the highly efficient pressure atomizing method of breaking the liquid fuel into fine particles for complete combustion. In the Delco Oil Burner with the Rotopower Unit, the motor, air blower, fuel pump, filter and fuel control valve are all contained within a single, easy-to-remove unit on an integrated shaft. The high-precision pump connecting directly to the shaft of the Rotopower Unit motor has only two moving parts, fitted together within 2/10,000ths of an inch.

Built into the Rotopower Unit is the Thin Mix Fuel Control which regulates the pressure of the fuel oil for proper mixing with air to assure the most economical flame. Delco Oil Burners are available in 4 sizes in standard voltage characteristics with combustion rates from 1 to 15 gal per hour or a capacity range from 440 to 5,400 sq ft of steam, EDR.

BITUMINOUS COAL STOKERS



SD-20 Stoker

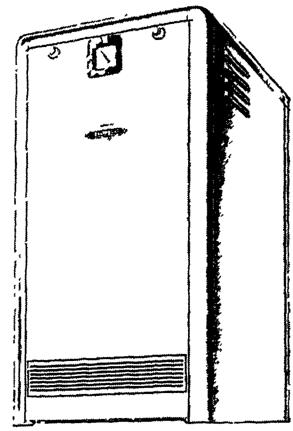
Delco Stokers which are designed to provide automatic firing for coal-fired domestic heating plants are of the underfeed, screw type with intermittent coal feed. A 20-pound, 30-pound and 50-pound stoker, burning bituminous coal, make up the Delco line. Features incorporated in deluxe models include: automatic controls, air control, Rhino-Hide lined hopper, smokeback eliminator, oversize feed worm, and sound insulation.

Fingertip transmission control permits ease of regulation of coal flow to suit the weather. Coal control consists of two speeds and neutral in 20-pound model; three speeds and neutral in 30-pound and 50-pound models. Using 12,000 Btu per pound coal, capacity of 20-pound model is 600 sq ft steam EDR, 30-pound models, 900 sq ft steam EDR, and 50-pound models, 1500 sq ft steam EDR.

DELCO AUTOMATIC HEAT

BOILERS

Delco Automatic Boilers coordinate the Delco Oil Burner with a boiler of special design and construction for application on hot water, steam, or vapor-vacuum heating systems. Most oil-fired models incorporate the famous Delco Oil Burner with the Rotopower Unit. Boiler sections are dotted with heat-absorbing fins. When sections are fitted together, fins form a series of passes, exposing a maximum of water-backed surface to the heated gases. Provision is made in all models for incorporating a built-in domestic water heater, and a coil for this purpose is included with each boiler. Five oil-fired automatic boilers, with capacities ranging from 350-1,335 sq ft of steam, EDR are available.



The DB-3 Boiler

WATER HEATER

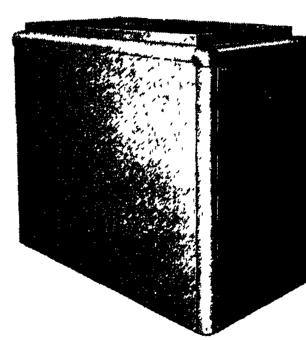
The Model DS Delco Automatic Water Heater resembles the Delco boilers in construction and operation. It employs the Quik-Action Heat Transmitter which conveys radiant heat with greatly increased speed; the simplified Rotopower Unit Delco Oil Burner that contains all moving parts within a single, easily removed unit on an integrated shaft; the Thin Mix Fuel Control which meters fuel oil economically and prevents waste; and the Heat Trap, a special baffle that conserves heat ordinarily lost. The square case, which is assembled and added to the unit after tank has been placed in position, makes installation easier and adds to the appearance of the finished job.

The DS Water Heater

CONDITIONAIRS

The Delco Conditionair is a compact, completely automatic unit, oil or gas-fired, which provides true winter air conditioning by circulating cleaned, humidified and properly heated air. Model DAOO Delco Oil Conditionair incorporates the new, exclusive Quik-Action Heat Transmitter. Air flow resistance is reduced to a minimum by tear drop design. Heat is transferred to the flowing air from a large heating surface dotted with heat projectors, and moisture is then added by a pan or spray type humidifier. A cooling attachment can be added for year 'round use.

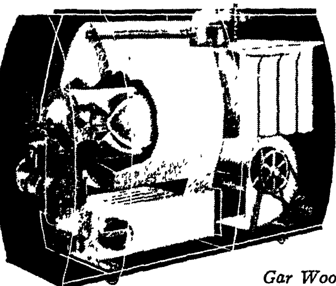
In Delco Gas Conditionairs there is a sufficient range in sizes to permit selection of the proper unit for applications ranging from 46,750 Btu heat loss to 102,000 Btu heat loss. Delco Conditionairs, oil-fired, range in size from 72,250 Btu heat loss to 170,000 Btu heat loss.



The Delco DAO Conditionair

Gar Wood Industries, Inc.

7924 Riopelle Street
Detroit, Michigan



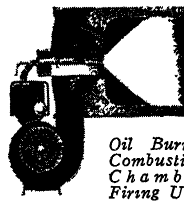
Gar Wood Tempered-Aire Home Unit

TEMPERED-AIRE UNITS

A high efficiency, direct fired heating unit incorporating a large-volume firebox, an integral economizer, a coordinated oil burner combustion chamber firing unit, a flash type humidifier, washable cloth filters, and a low speed rubber mounted blower.

The primary transfer of heat occurs in the large firebox, having its outlet at the bottom, through which the hot gases pass down into the economizer. Here the gases divide into long thin slices, within the economizer tubes. Each tube is swept at high velocity by cold return air from the blower. The rapidly flowing cool air absorbs a maximum of heat from the economizer surfaces.

The burner and firebowl combustion chamber are built together as an actual unit, with the back end of the firebowl forming a windbox containing air for combustion at sufficient pressure to offset the effect of draft variations. Air from the windbox passes through metering chutes, set at an angle to cause rotation. Both air and oil rotate and intermingle in a conical spray, resulting in a definitely controlled flame entirely contained within the firebowl.



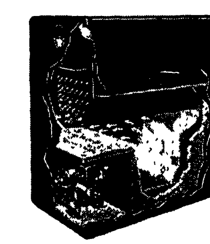
Oil Burner Combustion Chamber Firing Unit

Ratings and Dimensions	No. 001	No. 101	No. 201	No. 301	No. 401	No. 501
B.T.U. 1 Hour at Bonnet	85,000	100,000	135,000	200,000	300,000	400,000
B.T.U. 1 Hour at Grilles	68,000	80,000	110,000	160,000	240,000	320,000
*Air Delivery C.F.M.	850	1,000	1,350	2,000	3,000	4,000
Heating Surface, Sq. Ft.	40	54½	64	86	124	164
Firebox Volume, Cu. Ft.	6½	11	13	15	22	29½
Filter Area, Sq. Ft.	30	30	48	52	78	78
Motor H.P.—Blower	1/6	1/5	1/4	1/3	3/4	1
Overall Length, Inches	71	77	83	89	116	131
Overall Width, Inches	29¼	32	32	36⅝	45	45
Overall Height, Inches	48	54	54	57	65	65
Dimension—Supply Opening	28¾"x14¾"	35"x16¾"	36½"x16¾"	36¾"x16¾"	50"x21¼"	65"x21¼"
Dimension—Return Opening	28½"x14¾"	28½"x16¾"	32½"x16¾"	36"x16¾"	47½"x21¼"	47½"x21¼"
Blower Wheel, Size	10"x10"	12"x12"	14"x14"	16"x16"	20"x20"	20"x20"
Shipping Weight, Pounds	760	900	1,000	1,200	1,500	1,800
Stack Connection, Diameter	7"	7"	8"	9"	2-10"	2-10"
Recommended Chimney Size	8"x12"x30'	8"x12"x30'	8"x12"x30'	12"x12"x35'	12"x16"x40'	12"x16"x40'

*Ratings based on 160° bonnet temperature and 68° return temperature. Greater air deliveries with correspondingly lower bonnet temperatures may be had by the use of larger blower motors.

SERIES "R" BOILER-BURNER UNITS

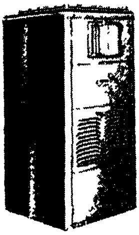
Series "R"	R1000	R1400	R1800
Burner Model	D	D	D
Oil Jet Size—Gal. per Hr.	3.50	5.00	6.50
Max. Net Steam Load, Sq. Ft.	1000	1400	1800
Max. Net Hot Water Load, Sq. Ft.	1600	2240	2880
Max. Gross Steam Load, Sq. Ft.	1500	2100	2700
Max. Gross Hot Water Load, Sq. Ft.	2400	3360	4320
Heating Surface, Sq. Ft.	84	118	154
Size Steam Nozzle	4"	2-4"	2-4"
Size Return Connections	2-2½"	2-2½"	2-3"
Overall Height, Jacket	68⅝	68⅝	68⅝
Overall Width, Jacket	37 13/16	37 13/16	37 13/16
Overall Length, Jacket	59¾	66	80
Height Steam Nozzle	68⅜	68⅜	68⅜
Height Return Connections	29⅞	29⅞	29⅞
Height External Heater Connection	43⅝	43⅝	43⅝
Height Mean Water Line	47⅜	49 9/16	49 9/16
Height Smoke Pipe Adapter	40	40	40
Hor. Distance Between Returns	7	7	7
Hor. Distance Between Steam Nozzles		18	18
Stack Size	12"	12"	12"
Recommended Chimney Size	12"x16"x40'	12"x16"x45'	16"x16"x45'
Water Capacity, Gals.	52	69	91
Shipping Weight, Lbs.	2200	2500	2950



SERIES "R"

The series "R" Boiler-Burner Units are designed for larger installations where their separate boiler shells and fireboxes facilitate rigging and erection.

GAS-FIRED AIR CONDITIONING UNITS



Gas-Fired Vertical Unit

Three basic, high efficiency heat exchanger sections are used singly and in multiple to produce the nine different furnace units listed below. The horizontal types, approximately 4 feet in height, are ideal for basements with low ceilings. The vertical types occupy small floor space, and are particularly suitable for installation in heater utility rooms located on the first floor, where space is limited.

The horizontal units feature the Gar Wood washable cloth filters for economy and collection of fine dust. An exclusive built-in diverter protects against varying and erratic drafts. A new humidifier provides adequate humidity. Canvas couplings for the air bonnets prevent sound telegraphing to the duct work. The large sized, slow speed, ball bearing, rubber mounted and canvas connected blower is driven by a hinge mounted motor with an adjustable speed pulley and built-in overload protection. These features assure adequate delivery of warm air throughout the house, and silent operation.

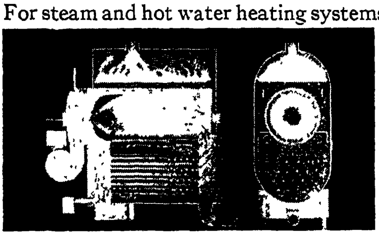
Important—When ordering, specify B.T.U. content of gas. Vertical types arranged for bottom inlet of return air when specified on original order.

Model	1 Input	1 Output	1 Room Delivery	2 Heating Surface	3 Air Delivery	4 Blower Motor	5 Blower Size	5 Flue Size	L In.	W In.	H In.	5 Supply Opening	5 Return Opening	6 Ship. Weight
60V	60	48	40	21½	480	1/6	10x10	6	27½	24⅜	68¼	22⅜x17	22⅜x25	320
84V	84	67	56	29	675	1/6	10x10	6	33½	24⅜	68¼	22⅜x23	22⅜x32	345
120V	120	96	80	40½	960	1/5	12x12	7	42½	24⅜	68¼	22⅜x31	22⅜x40	375
84H	84	67	56	29	675	1/6	10x10	6	61⅝	22½	54¾	20½x22	20½x27½	385
120H	120	96	80	40½	960	1/5	12x12	7	70⅝	22½	54¾	20½x31	20½x27½	415
168H	168	134	112	58	1340	1/4	14x14	8	66½	38½	54¾	36½x22	36½x31	515
240H	240	192	160	81	1920	1/3	16x16	9	75½	38½	54¾	36½x31	36½x31	570
360H	360	288	240	121½	2880	2-1/4	2-14x14	10	77½	54½	54¾	52½x31	52½x31	750
480H	480	384	320	162	3840	2-1/3	2-16x16	11	77½	70½	54¾	70½x31	70½x31	900

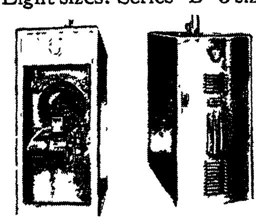
1–1000 BTU/HR 2–Sq. ft. 3–CFM 4–HP 5–Inches 6–Pounds.

SERIES "B" BOILER-BURNER UNITS

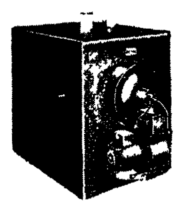
For steam and hot water heating systems. Eight sizes: Series "B" 5 sizes; Series "R" 3 sizes



Cross-sectional view of Models 135, 180 and 270



Models 135, 180 and 270



Models 120 and 165

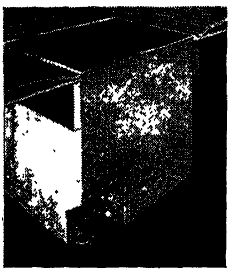
Series "B"	B-120-W	B-135-S or W	B-165-W	B-180-S or W	B-270-S or W
Boiler Nozzle Output B.T.U./Hr.	120,000	135,000	165,000	180,000	270,000
Net Load Rating B.T.U./Hr.	80,000	90,000	110,000	120,000	180,000
Boiler Nozzle Output Sq. Ft. 240 B T.U./Hr. Steam Radiation	—	562	—	750	1125
Boiler Nozzle Output, Sq. Ft. 150 B.T.U./Hr. Hot Water Radiation	800	900	1100	1200	1800
Net Load Capacity 240 B.T.U./Hr. Steam Radiation	—	375	—	500	750
Net Load Capacity 200 B.T.U./Hr. Hot Water Radiation	400	450	550	600	900
Net Load Capacity 150 B.T.U./Hr Hot Water Radiation	533	600	733	800	1200
Heating Surface, Sq. Ft.	30	35	40	46	64
Oil Jet Size, G.P.H.	1.20	1.35	1.65	1.80	2.70
Burner Type	"O"	"O"	"O"	"O"	"O"
Size Steam Nozzle	1-3″	1-3″	1-3″	1-3″	1-4″
Size Return Connection	1-3″	1-3″	1-3″	1-3″	1-4″
Stack Size	7″	8″	8″	8″	9″
Recommended Chimney Size	8″x12″x30′	8″x12″x30′	8″x12″x35′	12″x12″x35′	12″x16″x40
Water Capacity, Gallons—Steam	—	40	—	46	64
Water	14	51	20	64	88
Internal Tankless Water Heater (Dual Coil)	—	—	—	200*	200*
Internal Tank Type Water Heater (Single Coil)	—	30*	—	30*	30*
Shipping Weight—Pounds	885	1000	1100	1300	1585

*Gallons per Hour 80° Temperature Rise at 180° Boiler Temperature.

NORGE HEATING and CONDITIONING DIVISION

BORG-WARNER CORPORATION

12345 Kercheval Ave., Detroit, Mich.

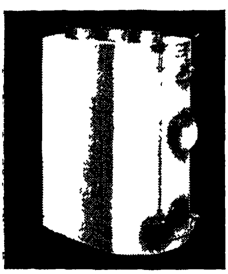


NORGE *Fastemp* OIL-BURNING FLOOR FURNACE

Model OC-60—60,000 Btu, Bonnet Output

Modern "under-floor" vaporizing unit, needs no furnace room. Extends down only 36 inches, controlled from the room above through floor grille. Low first cost (about the same as a space heater), no duct work or pipes.

"L"-shaped heat distributor with 25-40 per cent more heating surface and Permolain finish inside and outside to resist heat, acids, soot, rust; Down-Draft Whirlator feeds air into heart of flame; automatic chimney draft regulator; constant-level oil meter; 36 in. high, 28 in. wide, 40 in. deep.

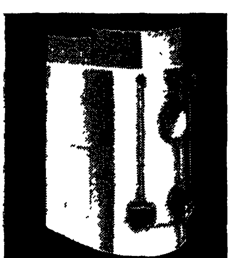


NORGE *Fastemp* OIL-BURNING FURNACE

Model OB-60—65,000 Btu, Bonnet Output

Gravity circulation vaporizing unit for basement installation. A "package" unit, ready to set in place and connect to ducts, and oil supply. Low first cost (about the same as a space heater) and low installation cost. Economical.

"L"-shaped heat distributor with 25-40 per cent more heating surface and Permolain finish inside and outside to resist heat, acids, soot, rust. Down-Draft Whirlator feeds air into heart of flame; automatic chimney draft regulator; constant-level oil meter; thermostatic control; 44 in. high, 26 in. wide, 40 in. deep.

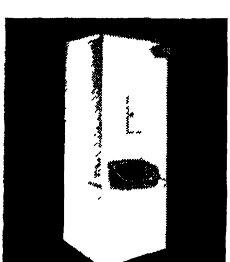


NORGE *Fastemp* OIL-BURNING FURNACE

Model OA-63—65,000 Btu, Bonnet Output Basement, 70,000 B. T. U. Output First Floor

Modern, semi-automatic vaporizing unit for installation in ground floor utility room, pit or basement. Actually costs less than many an old-fashioned furnace.

Silent, rubber-mounted blower, thermostatic control, 800 C. F. M.; "L"-shaped heat distributor with 25-40 per cent more heating surface and Permolain finish inside and outside to resist heat, acids, soot, rust; Down-Draft Whirlator: automatic chimney draft regulator; constant-level oil meter; 44 in. high, 24 in. wide, 40 in. deep.



NORGE OIL-BURNING FURNACE

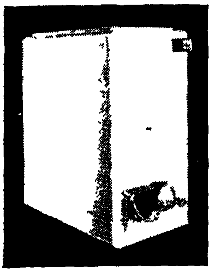
Model OD-70—70,000 Btu, Bonnet Output Pressure Vaporizing

Specificially designed for Government Agency contracts.

Modern, vertical unit; 500-900 C. F. M. Only 26 in. square, can be installed 2 in. from wall either side. Converts to winter air conditioner by adding filters and humidifier. Long life Permolain heat exchanger. Automatic draft supplied by 2-stage fan; low voltage thermostat; combination fan and limit control; safety float valve; fast, low-cost installation; factory wired; all controls fully automatic; meets every government code regulation including CS-75-39; 67 in. high, 26 in. square.

NORGE HEATING AND CONDITIONING DIVISION
BORG-WARNER CORPORATION . . . DETROIT, MICH.

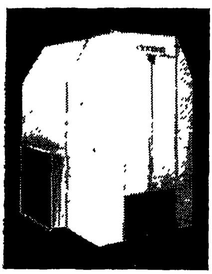
PRODUCT OF BORG-WARNER



NORGE OIL-BURNING WINTER AIR CONDITIONER

Model OE-80—80,000 Btu, Bonnet Output

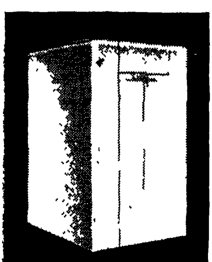
Horizontal pressure—vaporizing unit, 900 C. F. M.; for jobs requiring greater capacity than average small home. Basement or utility room installation within 2 in. of one sidewall and 9 in. at back. Permolain heat exchanger. Automatic draft from 2-stage fan; low voltage thermostat; combination fan and limit control; safety float valve; safety warp switch; fast, low cost installation; all controls fully automatic; meets every government code regulation including CS 75-39; 50 in. high, 26 in. wide, 47 in. deep.



NORGE OIL-BURNING WINTER AIR CONDITIONER

Model O-90—90,000 Btu, Bonnet Output

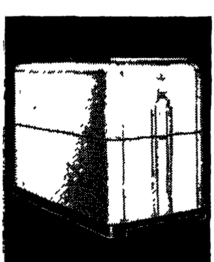
Fully automatic pressure atomizing type unit. 950 C. F. M.; automatic draft by "pull-through" fan; 1— 20 in. filter, single power unit; centralized visible servicing, low operating cost; quick low-cost installation; factory wired; automatic humidifier; horizontal firing tunnel; spiral "ramp" economizer; electric ignition; 53½ in. high x 33 in. wide x 49 in. deep.



NORGE OIL-BURNING WINTER AIR CONDITIONER

Model 120—120,000 Btu, Bonnet Output

Fully automatic pressure-atomizing type, oil-burning unit. 1200 C. F. M.; automatic draft by "pull-through" fan; 2—16 x 25 in. filters; with automatic safety by-pass single power unit; centralized visible servicing; low operating cost; quick low-cost installation; factory wired; factory tested; automatic humidifier; horizontal fire tunnel; spiral "ramp" economizer; electric ignition; basement or utility room installation; 62 in. high, 40 in. wide, 57 in. deep.



NORGE OIL-BURNING WINTER AIR CONDITIONER

Model BO-9—165,000 Btu, Bonnet Output

Modern, brilliantly engineered, deluxe unit for the larger home. Efficiency rating 84 per cent; 1500 C. F. M.; 2 filters with automatic safety by-pass; pressure atomizing oil burner; low operating cost; quick low-cost installation; factory tested; automatic humidifier; horizontal fire tunnel; spiral "ramp" economizer; electric ignition; 64 in. high, 40 in. wide, 68 in. deep.

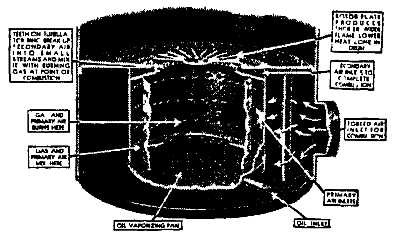
SEE NORGE BEFORE YOU BUY!

Send for detailed information regarding these Norge Units.

The Quincy Stove Manufacturing Co.

Quincy, Illinois—U. S. A.

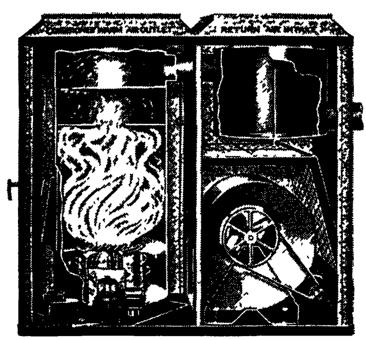
MONOGRAM Automatic Oil Burning Furnaces



MONOGRAM Turbulent-Flame Vaporizing Oil Burner

An important factor in the high efficiency of all MONOGRAM Heating Equipment is the MONOGRAM Turbulent-Flame Vaporizing Oil Burner—a new method of oil burning. No moving parts, nor frequent cleaning or servicing. Products of combustion are confined to the heating drum, and oil vapors, gas, or products of combustion can not escape into the building. The burner is continuously in operation—either on low or high flame, and maintains sufficient burner temperature to completely vaporize oil so that combustion occurs immediately when oil enters the burner—quick, efficient.

The cycle of operation from low fire to high fire and again reducing to low fire is gradual and without puff or explosion. In case of electric power failure the burner operates on low fire without danger of flooding, and can be operated manually by adjusting the oil flow and the draft regulator as required. Burner approved by the Underwriters' Laboratories, Inc.



Ten Furnace Models

Three sizes of Booster Gravity Units for quick and inexpensive change from coal to automatic oil heating, equipped with limit control, mechanical draft, and thermostat—these three models may be equipped with blowers for Full-Forced Circulation. Three sizes of Full-Forced Winter Air Conditioning units complete with air filters, automatic humidifiers, blowers, economizers, mechanical draft, limit control, blower switch and thermostat. An upright Full-Forced Winter Air Conditioner for basement or utility room installation equipped with separate blower for draft and air, limit control, blower switch and thermostat. Heating drum 12 to 14 gauge—all cabinets insulated with one inch fibreglas. Mechanical draft only on high fire. Low fire operates on .02 inch draft.

Special oil service station models from 50,000 to 72,000 Btu at bonnet—efficiency obtainable 80 to 81.7 per cent cfm 550 to 850.

Specifications for MONOGRAM Oilfire Furnaces

Type of Furnace	Booster Gravity			Full-Forced Warm Air			Full-Forced Winter Air Conditioner			Upright
Model No...	75	100	200	76	103	203	125	150	250	102
Btu Rating at Bonnet..	75,000	90,000	125,000	75,000	90,000	125,000	90,000	120,000	150,000	75,000
Per Cent Efficiency Obtainable	80	82	82	80	82	82	84	84	84	80
Air Delivery—cfm	. . .	.	. . .	400–700	800–1400	1000–1600	800–1400	800–1400	1000–1600	800–1400
Cabinet Floor Size—In.	26¼x25¼	30x30	36x36	26¼x52¼	30x56	36x66	26¼x49½	30x56½	36x68½	27x27
Cabinet Height—In..	50¼	60	60	50¼	60	60	50¼	60	60	71
Height With Bonnet ..	64½	74¼	74¼	. . .	. .	. .	. . .	. . .	. . .	. . .
Max. Gal. Oil per Hr.	0.69	0.81	1.12	0.69	0.81	1.12	0.80	1.05	1.32	.69
Min. Gal. Oil per 24 Hr.	1.50	1.00	1.50	1.50	1.00	1.50	1.50	1.00	1.50	1.50

New MONOGRAM Patented Burner vaporizes oil quicker—more completely, then induces both primary and secondary air in proper relation to oil, producing the highest known operating efficiency. Double baffle heat unit made possible by shorter, wider flame, stops rush of heat out the flue, creates a lower heating zone which combined with the clean, perfect combustion produces unheard of operating efficiencies.

Williams Oil-O-Matic Heating Corporation

Manufacturers of Automatic and Manually Controlled Fuel Oil Burners, Year 'Round Air Conditioning Systems, Williams Ice-O-Matic Compressors

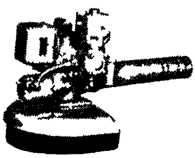
Bloomington, Illinois

WILLIAMS OIL-O-MATIC HEATING COOLING PRODUCTS

2014 Graybar Building
NEW YORK, N. Y.

OIL BURNERS

Williams Oil-O-Matic Lo-Pressure oil burners are offered in 6 sizes ranging in capacity from ½ to 25 gallons of fuel oil per operating hour. Oil-O-Matic Lo-Pressure burners can be installed in any type or size heating plant within this range, or larger where multiple units are used.



Williams Oil-O-Matic Lo-Pressure oil burners feature exclusive patented Thrift Meter fuel control. This device actually measures oil drop by drop to fit the exact requirements of any heating plant.

Williams designed, precision-built Hi-Pressure burners were developed for those jobs where first cost is most important. Three sizes range from .65 to 7 gallons per hour.

How to Decide Size of Burner

For low pressure domestic boilers, 1 gal of fuel oil per hour (140,000 Btu) is required for approximately:

300 sq ft of steam radiation or its equivalent.

480 sq ft of hot water radiation or its equivalent.

70,000 Btu when using hot air furnace ratings.

24 sq ft steam boiler heating surface (or 2.2 hp).

For exact detail data, see Oil-O-Matic Installation and Service Manual.

Heating Capacities of Oil-O-Matic Burners

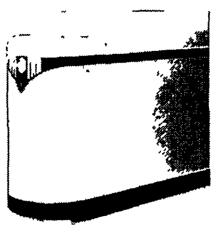
Model	Domestic Shipping Weight, Lb	Length, in.	Width, in.	Height, in.	HP	RPM	Gals. Fuel Oil per Operating Hour	
							Minimum	Maximum
K-150	126	29	15¼	19¾	1/10	1750	.50	1.50
K-3	180	30	15⅝	20¾	1/10	1750	1.00	3.00
K-4.5	190	33½	20¼	22¼	1/5	1750	1.35	4.50
K-7	200	26½	22¼	21¾	1/5	1750	4.00	7.00
J-1800	220	45	19½	23½	1/2	1800	8.00	15.00
JJ	295	50	32	24	1	1800	12.00	25.00
HP-1	90	20¼	14⅞	X	1/16	1725	.65	1.00
HP-3A	90	22¾	16½	X	1/12	1730	1.35	3.00
HP-7	150	26½	20	21¾	1/5	1750	3.50	7.00

X—Adjustable.

Standard draft pipe 11 in. 17-in. length draft pipe optional.

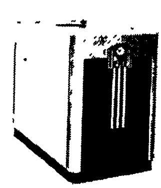
Standard electric current is 110-volt, 60-cycle.

WINTER AIR CONDITIONERS



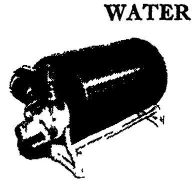
Williams Oil-O-Matic Winter Air Conditioners automatically clean, freshen, ventilate, humidify, heat and circulate air throughout the house. Electrically welded heat exchangers preclude possibility of air or gas leaks. Exclusive Anti-Pulsator eliminates starting vibration. Triple Thrift Lo-Pressure operation. Also available with Hi-Pressure burner. Built in 5 sizes to meet every heating load.

BOILER BURNER UNITS



Williams Oil-O-Matic oil-fired boilers are available in 3 models. The Lo-Boy (illustrated), in 3 sizes for steam, 3 for hot water; Residential Steel—4 sizes for steam, 4 sizes for hot water; and the Water Base—3 sizes for steam, 3 for hot water. Oil-O-Matic oil-fired boilers furnish low cost winter heating and economical year 'round domestic hot water. These complete units furnished with either Oil-O-Matic Lo-Pressure or Williams Hi-Pressure burners.

WATER HEATERS



Williams Oil-O-Matic horizontal type water heaters are compact, completely automatic, self-contained units. Lo-Pressure operation makes it possible to burn heavy domestic fuel oil for utmost economy. Unique Triple Flame Travel combustion chambers aids still further in economy. Where home is already heated with oil, fuel may be drawn from the same tank.

Water Heaters

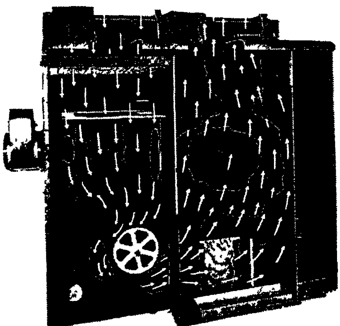
WHA*	600	57	22½	28	1/10	1800	...	½
WHB†	885	75	23	28	1/10	1800	½	1
WHC‡	1385	90	29	34	1/10	1800	1	1¾

*Output: 90 F rise, 60 gal per hour. †Output: 90 F rise, 120 gal per hour. ‡Output: 90 F rise, 210 gal per hour.

Kaustine Company, Inc.

Manufacturers of Kaustine Winter Air Conditioning Equipment

Main Office and Factory **Perry, N. Y.** **Sales Agencies in all Principal Cities**



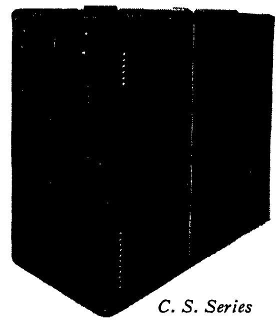
Deluxe Series

DELUXE SERIES

The Deluxe Series of winter air conditioners include patented Kaustine "Airflometers" which regulate quantity of air delivered to registers according to plenum chamber temperature. Wiring necessary for all controls can be completed at the factory. Chrome steel combustion chamber makes for rapid heating and high efficiencies. Improved pan type humidifier properly conditions the air. Spun glass filters clean air thoroughly. Multiple blade blower insures quiet operation. Service door and extended limit control increase ease in adjustment and safety. Heavy steel casings in gray wrinkled baked enamel with contrasting blue trim at top and base, or in two tone green baked enamel.

C. S. SERIES

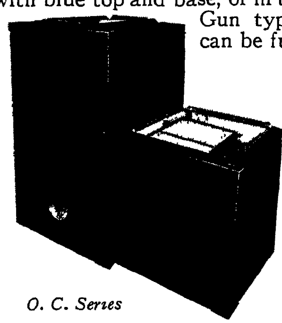
The C. S. Series of winter air conditioning units are constructed in capacities ranging from 80,000 to 270,000 Btu at the bonnet, and consist of quiet centrifugal type fan, spun glass filters, pan type humidifier, pre-cast combustion chamber liner, correctly engineered furnace and casing. Casings furnished either in gray wrinkled baked enamel with blue top and base, or in two tone green baked enamel. Gun type oil burners and controls can be furnished upon request.



C. S. Series

O. C. SERIES

The O. C. Series are constructed in two types, one with blower compartment at side of furnace (OC—100) and other with blower compartment beneath the heating element (OH—100). Both types are compact and easily handled. Conditioner includes quiet centrifugal type blower, spun glass filters, heating element, pre-cast combustion chamber liner, and pan type humidifier. No burner or controls are furnished.



O. C. Series

SPECIFICATIONS

Kaustine Deluxe and C. S. Series*—For Gun Type Oil Burners

Cat. No.	Btu Capacity	Blower Size	Cfm Capacity	Blower Motor	Overall Dimensions L	W	H
0811	80,000 to 110,000	11″	600 to 1200	1/4 Hp	64″	29″	47″
0124	120,000 to 140,000	13″	1000 to 2000	1/4 Hp	32″	36½″	60″
0157	150,000 to 180,000	15″	1800 to 2400	1/3 Hp	32″	36½″	60″
0207	200,000 to 270,000	17″	2400 to 3200	1/2 Hp	38″	42″	66″

NOTE. When requesting information on C. S. Series add C. S. to Deluxe Series Catalog Number.

Kaustine O. C. Series—For Gun Type Oil Burners

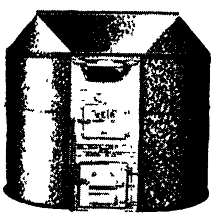
Cat. No.	Btu Capacity	Blower Size	Cfm Capacity	Blower Motor	Overall Dimensions L	W	H
OC-100	80,000 to 100,000	10″	600 to 1200	1/4 Hp	52″	27″	45″
OH-100	80,000 to 100,000	10″	600 to 1200	1/4 Hp	28″	27″	65″

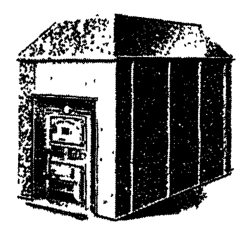
Kaustine Co., Inc., also manufactures Underwriters' Approved Basement and Underground Oil Storage Tanks in any size. Write for literature and prices.

THE MEYER FURNACE COMPANY
PEORIA, ILLINOIS

Manufacturers of Heating and Air Conditioning Equipment for Coal, Gas and Oil Burning

Branches and Distributors
KANSAS CITY, MO.
OMAHA, NEB.
GREEN BAY, WIS
PITTSBURGH, PA.
PHILADELPHIA, PA
ST. LOUIS, MO.
COLUMBUS, O.
MINNEAPOLIS, MINN.
NEW YORK, N. Y.
ATLANTA, GA





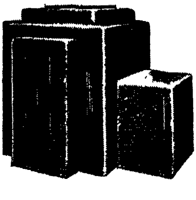
WEIR and MEYER Steel Warm Air Furnaces have a 60-year reputation for efficiency and dependability. Meet exacting standards of U. S. Government; supplied to numerous Camps and Cantonments. Heavy duty designs adaptable to commercial and industrial applications. Available in wide range of sizes for gravity circulation, as well as forced air, in round, oval or rectangular casings, for hand- or stoker-firing and in special designs for gas or oil, as well as coal.

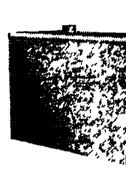
The outputs at the bonnet in the table below are based on the NWAH&ACA Technical Code for forced air furnaces, assuming 12,000 Btu coal.

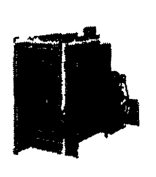
WEIR Forced Air Furnaces

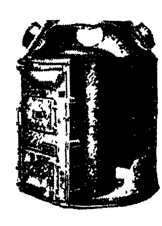
Furnace No.	Grate Area (Sq Ft)	Ratio Htg. to Grate Area	Output at Bonnet (Btu/Hr)	Air Delivery (Cfm)	Furnace Casing Dimen. (In.)	Smoke Outlet Diam. (In.)
20R	1.63	20.00	99,000	1,700	44	9
22-46Q	1.86	20.60	110,000	1,900	46	9
24-48Q	2.23	18.47	126,400	2,100	48	9
27-50Q	2.82	16.64	153,900	2,600	50	9
30-55Q	3.69	13.89	189,500	3,200	55	9
46-22C	1.86	28.50	127,300	2,200	46	9
48-24C	2.23	24.80	143,000	2,400	48	9
50-27C	2.82	21.07	168,400	2,900	50	9
55-30C	3.69	19.40	213,200	3,600	55	9
62-35CS	5.46	16.74	428,000	7,300	66	10
35C	5.46	26.10	477,500	8,100	56 x 82	2-10
540B	6.85	17.60	546,600	9,200	53 x 100	12
540B-2R	6.85	27.66	664,000	11,300	54 x 146	2-12
544B	8.67	16.23	673,000	11,500	57 x 108	12
544B-2R	8.67	26.10	816,000	13,900	58 x 158	2-12

The WEIR-MEYER Line includes, in addition to direct-fired furnaces in a wide range of sizes, a variety of self-contained winter air conditioning units, especially designed for coal, gas or oil, each featuring in highest degree, efficient heat interchange, quiet operation, design for "eye appeal," compactness, ease of installation, accessibility.









Pictured are but a few representative numbers. Ask for complete descriptive literature, including engineering data on any or all types.

ST. LOUIS
MEMPHIS
OMAHA
MINNEAPOLIS
SALT LAKE CITY
CHICAGO

L. J. Mueller Furnace Co.

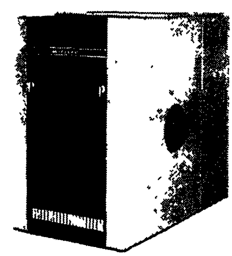
ESTABLISHED 1857

2009 W. Oklahoma Ave., **Milwaukee, Wis.**

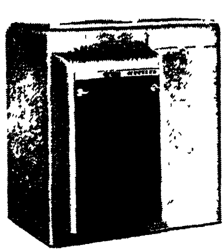
LOS ANGELES
KANSAS CITY
BALTIMORE
PHILADELPHIA
PITTSBURGH
WASHINGTON

SERIES 50 OIL-FIRED WINTER AIR CONDITIONING UNIT

Designed and constructed to meet the needs and purse of the moderate-sized home, this unit automatically heats, filters, humidifies and circulates the air within the home—efficiently and economically. The heating drum and radiator are made of heavy gauge steel, all electric welded, with no joints. Uniform distribution of the conditioned air is secured by the quiet, efficient Mueller fan. Filters furnished are of ample area, and with large dirt-holding capacity. Series 50 unit is available with a Mueller Vaporizing or Pressure Atomizing type oil burner. If desired, any standard burner may be used. Three sizes, from 100,000 to 225,000 Btu per hour.



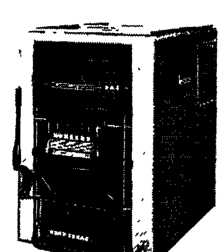
SERIES "EPS" GAS-FIRED WINTER AIR CONDITIONING UNIT



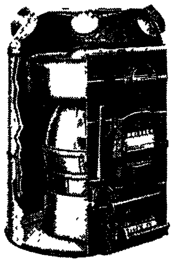
Designed and styled for the modern home, this Mueller unit meets every requirement for an automatic Winter air conditioning unit. Provides balanced distribution of filtered, humidified warm air in ample volume to every room. Heating unit consists of Mueller steel Heatspeeder sections, providing quick heat in desired volume. The fan operates quietly and efficiently, with ample capacity for any requirement. Filters thoroughly clean the air. Humidity is supplied automatically. Available in three sizes with AGA input ratings from 90,000 to 180,000 Btu per hour.

SERIES "FB" COAL-FIRED WINTER AIR CONDITIONING UNIT

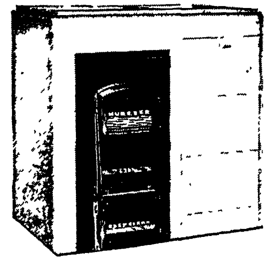
The smart, new, straight-line styling and unified design of this unit provides the same smartness, compactness and trim lines usually identified only with automatic heating equipment. The heating unit is of all-cast-iron construction, assuring a lifetime of dependable, economical heat. The heating unit, blower, and filters are enclosed within the crinkle-lacquered, insulated housing. Filters are replaceable type, and are of ample area. The blower provides uniform distribution of the conditioned air to all rooms. Available in six sizes, with hand-fired ratings from 68,000 to 199,000 Btu at register.



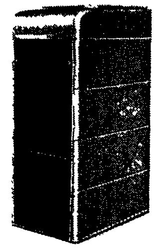
Mueller Heaters For All Fuels
A Complete Line for All Purposes



Return Flue all-cast Furnace. 18 in. to 30 in. firepots, single and double firedoor styles. Available in round, galvanized or square, lacquered casings.

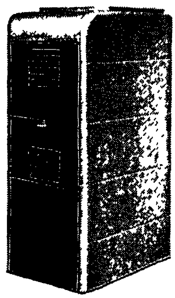


Series P-400 Steel coal-fired fan-filter-furnace unit. Also available with round casing for gravity operation. Four sizes, 20 in. to 27 in. drums.

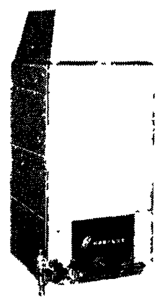


Series OVP oil-fired winter air conditioner—steel construction. Equipped with Mueller vaporizing burner. One size, 80,000 Btu at bonnet.

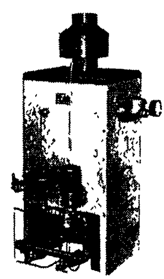
Series CVP gas-fired winter air conditioner. All-cast-iron heating unit. A.G.A. input ratings from 125,000 to 200,000 Btu per hour.



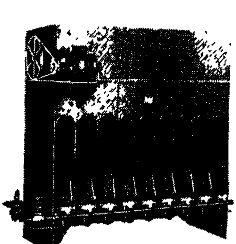
Gas-fired air conditioning furnace. A.G.A. input ratings from 60,000 to 100,000 Btu per hour. Wide range of air deliveries.



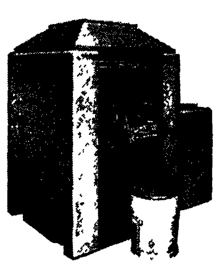
Series G gas-fired gravity furnace. One size, A.G.A. input rating, 90,000 Btu per hour. Available with square or round casing.



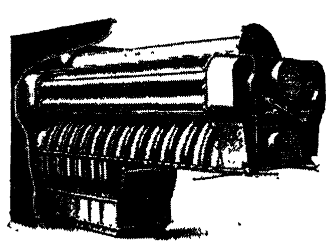
Series "AE" Gas Boiler. *AGA* ratings, 180 to 1,260 sq ft steam; 290 to 2,015 sq ft hot water.



Gas-fired unit heater. Sizes from 4 to 12 sections. A.G.A. ratings, 180,000 to 540,000 Btu per hour.



Series "SA" stoker-fired furnace, with fan-filter unit. Any stoker may be used. Capacities, 110,000 and 175,000 Btu.



Horizontal Tubular Heaters, for schools, churches and other large buildings. Three sizes, with capacity range from 1,188,000 to 1,390,000 Btu per hour.

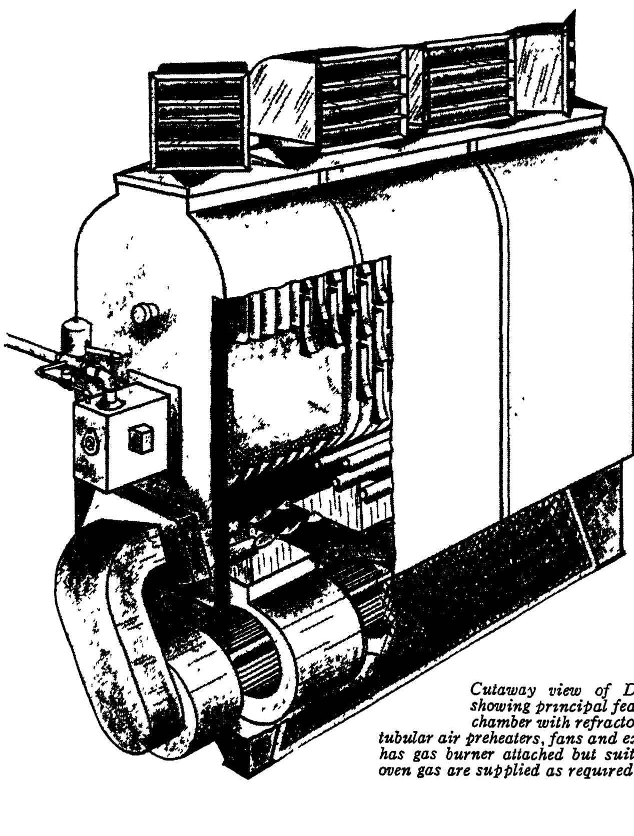
Complete catalogs on each of above units available upon request.

DRAVO CORPORATION

MACHINERY DIVISION

300 Penn Avenue, PITTSBURGH, PA.

DRAVO



Direct Fired Heaters produce efficiencies up to 85%

Cutaway view of Dravo Direct Fired Heater, showing principal features; corrugated combustion chamber with refractory lining, fins and deflectors, tubular air preheaters, fans and exhausters. The heater shown has gas burner attached but suitable burners for oil or coke oven gas are supplied as required.

A study of the rugged construction, the simplicity of operation and speed of installation offered by the Dravo Heater is convincing as to its overall practicability. Each unit is complete and is operated individually. A heating system for any size structure can be devised by a combination of units, often at a cost far below that of a steam boiler plant. Investigate this new heating method!

The exceptionally high heat transfer efficiency is the result of two fundamental design features—first, accurately controlled and scientifically correct combustion of fuel and second, highly effective transfer of heat to air. Both results are accomplished in a small rugged, compact unit.

Built under Anderson patent No. 2715057—other patents pending. Standard stock sizes are obtainable for production of 700,000 to 3,000,000 Btu per hour. Dravo Bulletin No. 502 with detailed description mailed on request.

Dravo Direct Fired Heaters

9 Advantages

1. Give instant heat when the starter button is pushed.
2. Maximum heat utilization is secured from fuel (maximum efficiencies up to 85 per cent.)
3. Fully automatic firing, natural gas, light or heavy, fuel oil, or coke oven gas.
4. Initial cost and installation usually less than other types of heating systems as no boiler plant is required.
5. Lower maintenance cost, eliminating freeze-ups, broken traps, leaking valves, etc.
6. Saving in labor costs—can be operated by maintenance man.
7. Can be used to furnish summer air circulation. Arrangements can easily be made to provide outside fresh air. Likewise, filters can be installed on air intakes.
8. Deliver heat over an entire floor area or to a localized section with minimum temperature differential between floor and ceiling.
9. Can be installed quickly during building construction for temporary heat.

Effective heat distribution is accomplished as far as 200 feet and floor to roof temperature range is less because cold air is taken from floor.

Duct systems may be attached to standard models if required for partitioned spaces or any exceptional condition. Heaters may be suspended if necessary. Dravo Engineering Service will quickly estimate requirements and submit recommendations suited to building and climatic conditions.

ILLUSTRATIONS

One of 4 Heaters at Air Associates, Inc., Bendix, N.J. This is a new airplane parts manufacturing plant of roughly 88,000 sq ft floor area with a 27 ft high factory section.

The U.S. Armor Plate Plant at So. Charleston, W. Va., where 22 of the Dravo 1,000,000 Btu output heaters are in use.

Pittsburgh Rolls Division — Blaw-Knox Company. The "L" shaped machine shop building, the wings of which are 114 ft and 330 ft, totals 23,900 sq ft. A duct system is used to deliver heat right at machine stations.

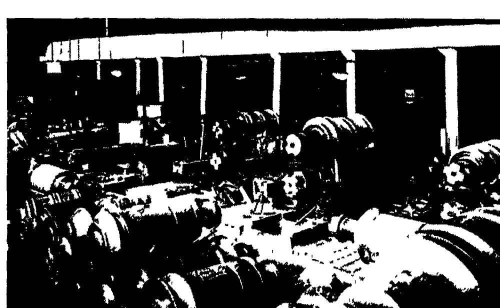
Ideal for temporary heating during construction, expansion or remodeling, Dravo Direct Heaters are used extensively for providing temporary heat. Ease of installation and removing is an outstanding advantage.

Fuels used—Oil, Gas, Coke oven Gas. Dravo also offers a coal-fired all steel shell type heater with stoker firing if desired.

Special Industrial Types of Heaters for process heating. Where high temperatures are required another design of heater is used, in which the products of combustion mix with the air stream. Outlet temperatures go as high as 1000 degrees; for soap drying, defogging, pickling plants, drying strip steel, etc.

Air Associates, Inc., Bendix, N.J.

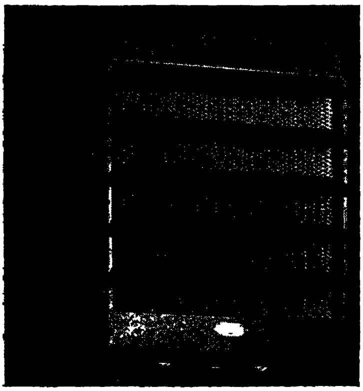
U.S. Armor Plate Plant, Charleston, W. Va.



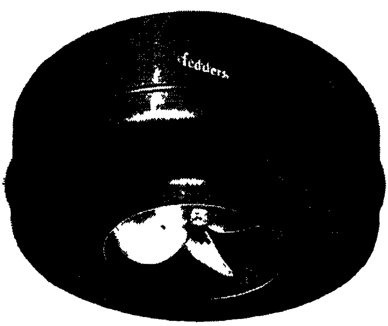
Pittsburgh Rolls Div.—Blaw-Knox Co.

FEDDERS

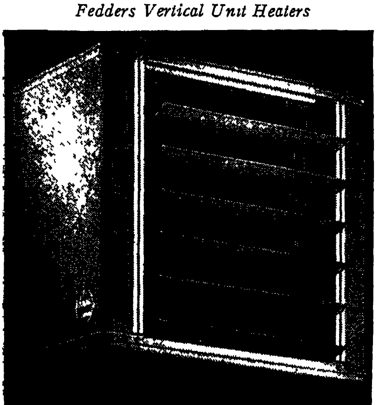
MANUFACTURING COMPANY, INC.
85 TONAWANDA ST.
BUFFALO, NEW YORK



Fedders Horizontal Unit Heaters



Fedders Vertical Unit Heaters



Fedders Unit Coolers

FEDDERS UNIT HEATERS

Horizontal Type

Streamline tubes, individual non-clogging fins, patented full-floating mounting eliminates expansion stresses between heating element and mounting, relief of differential expansion among tubes, rugged monopiece cabinets, quiet operation. Standard horizontal delivery unit heaters built in 24 basic sizes, all available with single, two or multi-speed and standard speed motors. Also available with slow speed motors and motors for odd frequencies.

FEDDERS UNIT HEATERS

Vertical Type

Designed for high ceiling installations where supply and return piping will not interfere with overhead equipment such as craneways, shafting, tall machinery. High velocity fan delivers heated air down to the working zone where draftless diffusion is accomplished by using suitable directional outlet to fit conditions.

FEDDERS UNIT COOLERS

Built in a complete range of single and twin fan sizes for use with refrigerant or cold water. Finished in attractive polar green.

FEDDERS AIR CONDITIONING UNITS

Available in any combination for heating, cooling, humidifying and dehumidifying and air filtering. Seven basic models from 1 to 25 tons cooling capacities designed for floor or ceiling installations.

Fedders Air Conditioning Units

HEAT TRANSFER SPECIALISTS SINCE 1896

FEDDERS

MANUFACTURING COMPANY, INC.
85 TONAWANDA ST.
BUFFALO, NEW YORK

FEDDERS TYPE K HEATING COILS

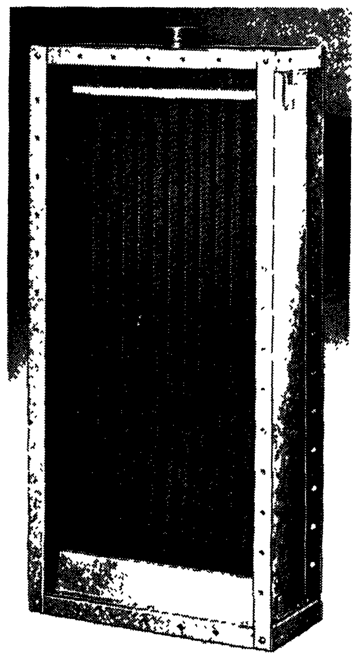
Strong, rigid casings . . . large cylindrical headers . . . full-floating protection against overall expansion . . . top header tri-point supported by center anchorage brackets and drop forged bronze trunnions . . . knee action relief of differential expansion among tubes . . . scale breaker-tube orifices . . . floating type tube supports . . . permanently bonded fins and tubes.

7 Standard Face Widths 12¾ in. to 36⅛ in.
18 Standard Face Lengths 1½ ft. to 10 ft.
3 Fin Spacings.
1 or 2 Row Deep Coils of all 3 Fin Spacings.

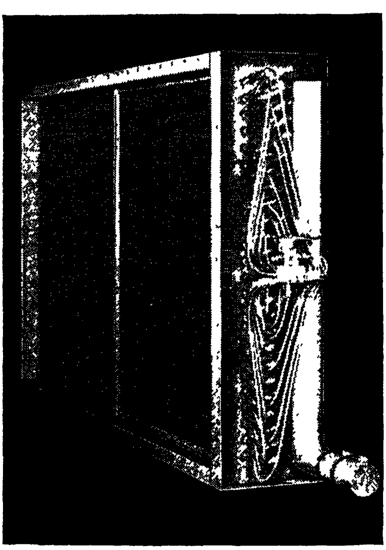
BASIC TEMPERATURE RISES

Coil Type	K15	K16	K18	K25	K26	K28
Air Temp. Rise° F*	36.1	52.3	62.5	67.0	93.0	108.0

*Enter. Air 0°F, 500 FPM, 5 lbs steam



Fedders Type K Heating Coil



Fedders Type R Cooling Coil

FEDDERS TYPE M MODULATING COILS

Non-freeze design with general construction similar to Type K Coils. Wide range of standard sizes.

FEDDERS TYPE B BOOSTER HEATING COILS

Featuring same advanced engineering as Type K Coils. Complete range of sizes.

FEDDERS COOLING COILS

Type R for use with refrigerant—shown at left. Note multiport thermostatic expansion valve with equalizer connection, and balanced distribution system.

Equipped with flat fins for rapid air-side condensate removal.

Built in wide range of standard sizes. Type W for Water.

Hastings Air Conditioning Co., Inc.

Hastings, Nebr.

Manufacturers of

Air Conditioners.
Unit Heaters.
Utility and Package Blowers.

Dealers and Representatives in Principal Cities

A Complete Line of Highly Successful COLD WATER Air Conditioners. Capacities listed depend on entering air and water temperatures.

All equipment available for combination heating and cooling.

FLOOR MODELS

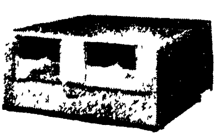
Floormasters—Unusual design and special features permit maximum installation possibilities with minimum floor space and installation costs.

Air Delivery—2240 cfm. Cooling Capacity—3 to 6 tons. Dimensions—Height 93 in., Width 48 in. Depth 25 in. Motor—½ hp. Filters—3 16 in. x 25 in.

Royal—For offices, homes, hospitals, etc.

Air Delivery—590 cfm. Cooling Capacity—1 to 2 tons. Motor—1/6 hp. Filter—1 16 in. x 25 in. Dimensions—Height 40 in., Width 28 in., Depth 20½ in.

CENTRAL PLANTS



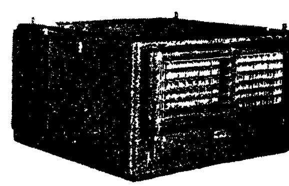
Sectional construction for ease of handling. Motors inside mounted to provide very neat appearing compact units.

SPECIFICATIONS

Size	CFM	Motor Hp	Filters	Capacity Tons
CP 30	3,000	1	5	4-9
CP 40	4,000	1	8	6-12
CP 60	6,000	2	10	9-18
CP 80	8,000	3	12	12-24
CP120	12,000	5	20	18-36

GENERAL UTILITY MODELS

Master—Singly or in multiple are suitable for any business or space size. Large jobs handled without duct work by proper location of units.

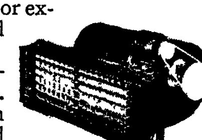


Air Delivery—2,240 cfm. Cooling Capacity—3 to 6 tons. Dimensions—Height 29 in., Width 49 in., Depth 50 in. Motor—½ hp. Filters—4 16 in. x 23 in.

Majestic—Similar to Master except size. Air Delivery—2,240 cfm. Cooling Capacity—1½ to 3 tons. Motor—¼ hp. Filters—2 16 in. x 25 in. Dimensions—Height 26 in., Width 28 in., Depth 40 in.

Zephyr—Same capacity, motor and filter as the Royal. For use where suspended or concealed units are desired. Dimensions—Height 26 in., Width 24 in., Depth 28 in.

UNIT HEATERS

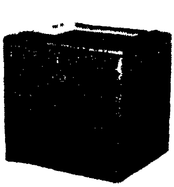


Centrifugal Type for extreme quietness and efficiency.

Steam pressure—to 150 lbs per sq in.

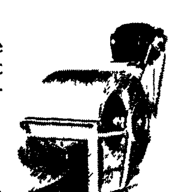
Finish—Brown wrinkle enamel and stainless steel louvers.

PACKAGE AND OPEN TYPE BLOWERS



May be knocked down for narrow doorways. Finished in attractive green wrinkle.

Utility type blowers are available with or without motors and in any discharge desired.

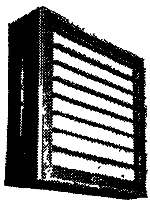


All sizes from 9 in. to twin 21 in.
Air deliveries from 1000 cfm to 16,000 cfm.

Write for Catalogues, Literature, or Information

Kramer Trenton Co.

Manufacturers of
HEATING, COOLING AND REFRIGERATION DEVICES
Trenton, New Jersey

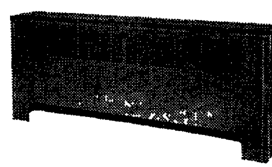


KRAMER UNIT HEATERS

All-copper heating element. Oval-section tubes with hair-pin bends. High-discharge air velocity insures proper heat distribution. For pressures up to 150 lb.

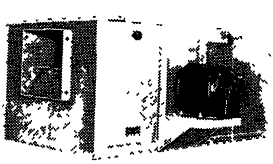
Send for Bulletin H-141

KRAMER COPPER CONVECTORS



All-copper heating element. Oval tubes with fins metallically fused to tubes. Noiseless operation. Guaranteed for operating steam pressures up to 50 lb

Send for Bulletin H-240

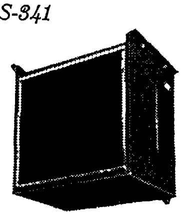


HEATING and AIR CONDITIONING UNITS for Residential Use

Designed for split-system installations. A range of sizes adaptable to residential requirements. Rubber mountings and flexible connections minimize noise.

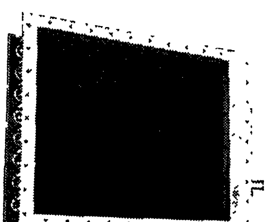
Send for Bulletin SS-341

KRAMER COMFORT COOLERS



Suspended type for small tonnages—1 to 3 tons—and for remote compressor operation. All-copper coils. Specially designed grille for proper diffusion.

Send for Bulletin R-142

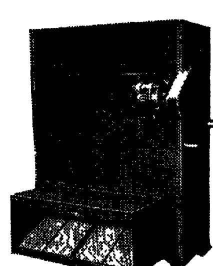


KRAMER TURBO-FIN

For blast heating and cooling. All-copper blast surfaces; fins metallically fused to tubes. Air side flow-disturbers. Coil finished in electro tin plate for permanence.

Send for Bulletin AC-540

KRAMER AIR CONDITIONING UNITS
Ceiling and Floor Type



Wide variety of sizes and capacities—2 to 30 tons in cooling; 65,000 to 1,280,000 Btu per hour in heating. Accurately rated. All-copper Turbo-fin coils; fiber-glas air filters. Complete cabinet types for either floor or ceiling mounting.

Send for Bulletin AC-440

McQuay, Inc.

1602 Broadway, N.E., **Minneapolis, Minn.**

MANUFACTURERS OF AIR CONDITIONING EQUIPMENT

Sales Offices in all Principal Cities

- **Air Conditioners**
- **Air Conditioning Coils**
- **Blast Heating Coils**
- **Refrigeration Coils**
- **Convection Radiation**
- **Unit Heaters**
- **Unit Coolers**

PROVEN McQuay PRODUCTS

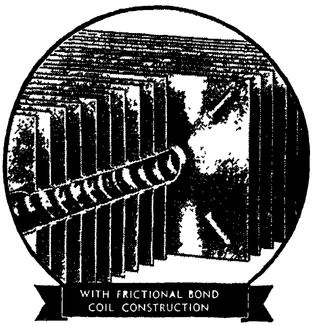
- **Comfort Coolers**
- **Blower Coolers (Suspended & Floor Type)**
- **Room Coolers (Cabinet Type)**
- **Ice Cube Makers**
- **Icy-Flo Accumulators**
- **Zeropak Low Temp. Units**

THE EXCLUSIVE McQUAY FRICTIONAL BOND FIN-AND-TUBE COIL ASSEMBLY

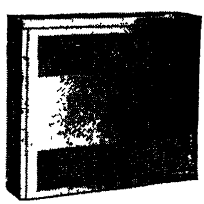
The McQuay Fin and Tube assembly in all McQuay coils and cores is one of the reasons McQuay products are considered "Tops in Over-All Efficiency" by many heating and refrigeration authorities.

Heat transfer efficiency primarily depends on three elements in coil construction. First, "Area of Contact," Second, "Contact Pressure" and finally "Quality of Contact" between collar and tube.

In McQuay coils *all three* necessary elements are found developed to their highest degree. The famous McQuay "Wide Fin Collar" plus Exclusive Hydraulic Expansion together with the polished surface, secured by "spinning" the fin collar, truly provides the last word in Heat Transfer.



McQUAY STANDARD CONVECTORS



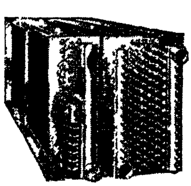
STANDARD CONVECTOR

The Standard all purpose Convector has been designed to meet all heating requirements. They are available for free standing, partially recessed, fully recessed and wall mounting applications.

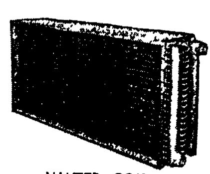
All enclosures are constructed from high grade steel, properly reinforced to make a sturdy cabinet.

The heating element is constructed of a series of round seamless copper tubes to which are attached die formed radiating fins, which are bonded to the tubes by the exclusive McQUAY Hydraulic Expansion Process.

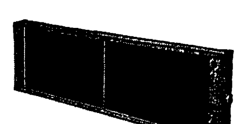
We offer the services of our Engineering and design department to help solve your heating problems.



COMBINATION COOLING COIL



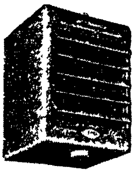
WATER COIL



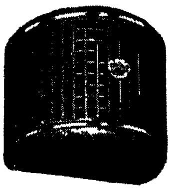
STEAM BLAST COIL

MORE THAN 1,000,000 STANDARD COIL TYPES AND SIZES

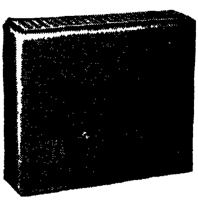
McQuay manufactures the most complete line of Standard Coils in the Industry.
Coils for Heating—1 to 10 rows deep using low or high pressure steam or hot water.
Non-Freeze—(steam distributing tube) type coils 1 and 2 rows deep.
Removable Plugs—(cleanable tube) type coils 1 to 12 rows deep.
Water Coils for Cooling—1 to 12 rows deep.
Direct Expansion Coils—for cooling 1 to 10 rows deep.
Refrigeration Coils—all types and sizes.
Special Coils—of various materials furnished on order for special applications.



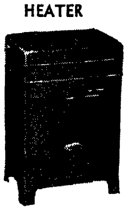
STANDARD UNIT HEATER



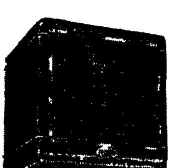
RADIAL UNIT HEATER



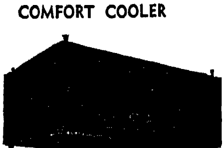
CABINET TYPE UNIT HEATER



COMFORT COOLER



COMFORT COOLER



AIR CONDITIONER (YEAR-ROUND)

UNIT HEATERS

The new and exclusive McQuay Radial Heater joins a distinguished old family of proven heating units—the Down Flow—the Cabinet Unit—the Large Blower Type, and the veteran Standard Unit Heater, making the McQuay line the most complete in the industry.

This newest McQuay development provides wide uniform heat distribution, lower installed cost and fine appearance, combined in one unit. One Radial Heater can now be used in place of two or more Standard Units—effecting important savings in piping expense as well. Actually it is a deluxe unit heater at lower cost than standard units, in most cases.

All McQuay unit heaters feature the exclusive Frictional Bond coil construction. All types of Heaters are furnished in a wide range of sizes with motors to meet all electrical current characteristics, making it convenient to select the proper size heater for every installation.

McQUAY COMFORT COOLERS

Made in two types—one for use with water or brine; another for freon or methyl chloride. Eight sizes in each type—all with 4-speed motors.

AIR CONDITIONERS—COLD WATER AND FREON TYPES

Choice of recirculation of indoor air, entire intake of outside air, or a combination of both. Cold water or brine used in one type; freon or methyl chloride in another. Modern "sound isolated" construction assures quiet operation. Capacities to 6 tons.

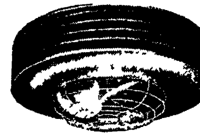
CENTRAL SYSTEM AIR CONDITIONING UNITS

Suspended and floor types, cools, dehumidifies, filters, and circulates air in summer; heats, humidifies, filters and circulates air in winter. Extreme flexibility and accessibility "built-in". Cooling capacities from 5 to 50 tons in both Suspended and Floor Type.

McQUAY ICY-FLO ACCUMULATORS

The new practical "Storage-Battery" for refrigeration effect is now available for handling heavy loads of short duration.

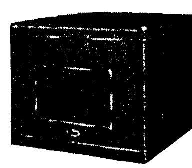
New Descriptive Bulletins are ready on all McQuay Products. Write McQuay, Inc., Minneapolis, Minnesota.



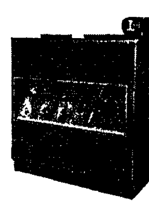
DOWN FLOW UNIT HEATER

BLOWER TYPE UNIT HEATER

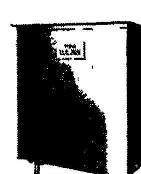
BLOWER TYPE UNIT HEATER



AIR CONDITIONER



AIR CONDITIONER (YEAR-ROUND)



ACCUMULATOR

Modine Manufacturing Company

Heating and Air Conditioning Division

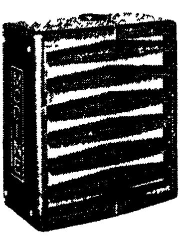
General Offices: 17th and Holburn Sts., **Racine, Wis.**

Factories at Racine, Wis. and La Porte, Ind.

Branches in all Principal Cities

Complete information on the following products including engineering data and prices, can be secured by writing to the Modine General Offices at Racine, Wisconsin—or by communicating with nearest Modine representative.

MODINE UNIT HEATERS



Front View

Back View

The New Modine Unit Heater incorporates in its design, many features which contribute to more satisfactory and economical industrial and commercial heating.

Sound Silenced—Interior surface of casing is coated with acoustical-mastic, deadening noise from within. Venturi fan shroud, integral with casing, quiets air in-rush sound. Velocity generator eliminates air-rush noise peaks. Concentric rings of fan guard act as vibration dissipators.

Safety Fan Guard—Provides staunch, steel safeguard against hazard of unshielded fans. Safety is built in as standard equipment.

Protection Against Rust—Available by Bonderizing—When applied, Bonderizing of casing and sheet metal parts makes them resistant to formation and progress of rust. It holds finish to metal, making it more durable and permanently fine in appearance.

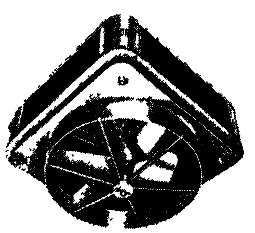
And These Additional Features—(1) *Velocity Generator* gives greater heat throw without increasing power requirements. (2) *Patented Expansion Bend* permits tubes to stand extreme expansion. (3) *Direct Pipe Suspension* permits installation without hangers—saves cost.

Unit Heaters—Capacities and Dimensions (In Inches)

Model No.	Over-all Height	Width	Depth Less Motor	E.D.R.	C.F.M.	Motor R.P.M.
74	11¾″	10″	6″	74	296	1580
104	16¾″	14¼″	9″	104	350	1580
140	16¾″	14¼″	9″	140	540	1580
172	19½″	14¼″	9″	172	661	1580
206	22½″	19″	11″	206	843	1120
252	22½″	19″	11″	252	980	1120
304	22½″	19″	11″	304	1290	1120
362	24″	19″	11″	362	1450	1120
414	22½″	19″	11″	414	1370	1120
514	27½″	23″	11½″	514	2250	1120
606	27½″	23″	11½″	606	2440	1120
711	27½″	23″	11½″	711	2430	1120
808	29½″	26½″	13″	808	2760	1120
904	29½″	26½″	13″	904	3370	1120
1050	33¾″	26½″	13″	1050	3770	1120
1200	33¾″	26½″	13″	1200	4120	1120
1380	33″	34″	11″	1380	4960	1120
1610	30″	50¾″	11″	1610	5920	1120
2030	30″	54¾″	11″	2030	7720	1120

All above models are available with variable speed motors Units for hot water application also available.

MODINE VERTICAL DELIVERY UNIT HEATERS

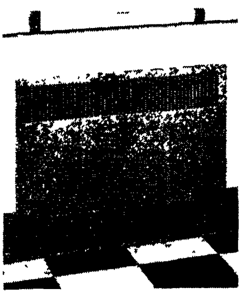


Modine Vertical Delivery Unit Heaters are indicated wherever conditions call for more directly downward delivery of air than is provided by the conventional design of unit heaters. In factories where high clearance is essential, as for craneways, Modine Verticals serve excellently. Over store and office doorways, they fit perfectly into the need for a blast of warmed air to offset the in-pressing winter gales. In a room where only one unit heater is needed, the Modine Vertical gives delivery of heat to the entire perimeter of the room. Or by adjustment of the Modine Cone-Jet Deflectors, this delivered air may be concentrated in limited directions, even checked almost entirely to a single side of the unit.

Similarly, multiple installation of Modine Verticals may be controlled as to heated air deflection so as to give more delivery of heated air to outside walls, thus conforming to normal heating requirements. Modine Verticals are made in thirteen models. Write for catalog.

MODINE CONVECTORS

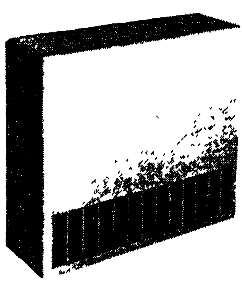
The popular radiators for commercial and public buildings, low cost houses, etc. —wherever the benefits of convector heating are desired. Attractive enclosures with removable fronts. Wide selection of grilles.



Rust protection of enclosures available by Bonderizing. High capacity heating units. Enclosures are made in two types of Recessed and Floor and Wall Cabinets and Concealed (plaster-front) types. **Catalog 241A.**

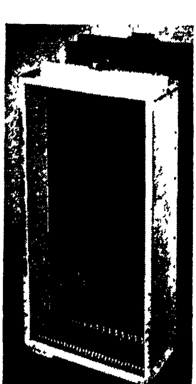
CABINET UNIT HEATERS

Modine Cabinet Unit Heaters are designed for the heating of offices, lobbies, corridors, auditoriums, etc. Used in conjunction with a steam or hot water system, they elimate the need for unsightly obsolete radiators.



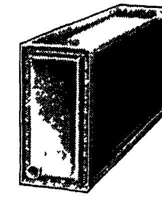
Models include the Floor Cabinet, Wall Cabinet, Ceiling and Recessed types—each available in three capacities: 105, 310 and 450 E.D.R. **Bulletin 841.**

MODINE BLAST HEATERS



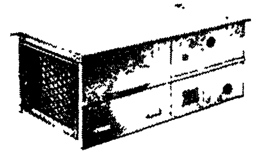
Made in over 250 sizes types and capacities to meet the specific demands for heat transfer service. Outstanding features are: (1) Expansion Bend. (2) All steam carrying passages are cylindrical for greatest possible strength. (3) From inlet to outlet condenser is of copper or copper alloy. (4) Copper fins are bonded metallically to tubes. **Catalog 340.**

MODINE COOLING COILS



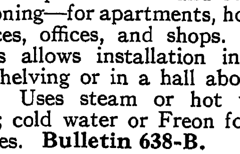
For use in central system cooling and air conditioning plants, Modine Cooling Coils, Cold Water Type, are installed with a blower fan and duct work. Adaptable where cold water or non-corrosive brine is used as the cooling medium. Coils are available in *cleanable* and *continuous Tube* types. **Catalog 540.**

UNIT COOLERS (Blower Type)



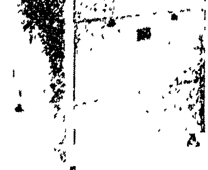
For stores and offices. This unit cools, cleans, dehumidifies and circulates the air. Equipped with powerful, yet quiet blower, extra deep cooling coils and large-area air filters. May be installed with or without duct work. Choice of cold water or Freon cooling coils. **Bulletin 440.**

AIR CONDITIONER
(Apartment House Type)



A compact unit performing every function of complete winter and summer air conditioning—for apartments, hotel suites, residences, offices, and shops. Its compactness allows installation in a closet above shelving or in a hall above a false ceiling. Uses steam or hot water for heating; cold water or Freon for cooling. Two sizes. **Bulletin 638-B.**

AIR CONDITIONER
(Large Central Type)



For residential and commercial year-'round air conditioning—may be used in straight air conditioning or split systems. Uses steam or hot water for heating and cold water or Freon for cooling. **Catalog 639.**

The Unit Heater and Cooler Division

D. J. Murray Mfg. Co.

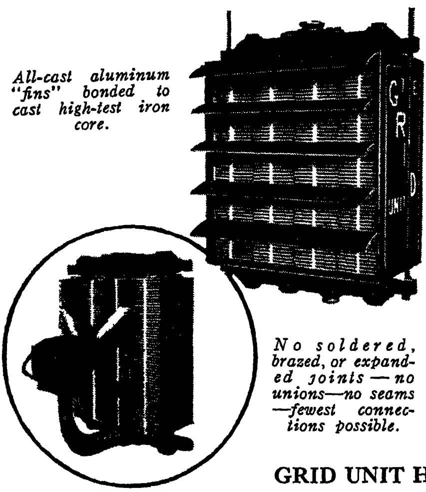
Wausau, Wisconsin

Offices in Principal Cities

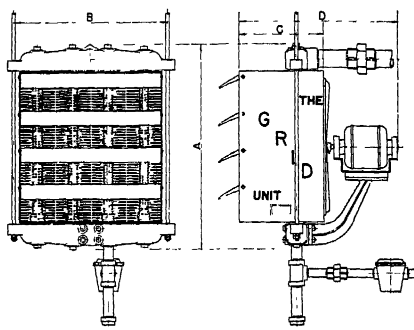
MANUFACTURERS OF THE GRID UNIT

(PATENTED)

All-cast aluminum "fins" bonded to cast high-test iron core.



No soldered, brazed, or expanded joints — no unions—no seams —fewest connections possible.



Typical Installation of Grid Unit Heater

GRID UNIT HEATER DATA

Model No.	Dimensions Inches				Face Area Sq Ft	Motor		Vol. at Fan	Capacities 5 Lb Steam 60° Air		Approx. Shipping Weight	Pipe Sizes	
	A	B	C	D*		Hp	Rpm		Btu	Final Temp.		Supply	Return
1000	16¼	11¾	9⅜	16	0.84	1/20	1700	578	29,400	107	100	1¼"	1¼"
1200	18½	13½	11½	17½	1 04	1/20	1700	711	46,000	119	120	1¼"	1¼"
515	23⅜	17½	11½	20	1 65	1/10	1750	1290	59,600	102	160	1½"	1¼"
1500	23⅜	17½	11½	20	1 67	1/10	1750	1450	77,500	109	210	1½"	1¼"
1520	27⅜	17½	11½	20	2 2	1/10	1750	1700	104,000	113	250	1½"	1¼"
520	28⅜	22⅛	11⅞	21½	2.9	1/6	1150	2500	102,300	97	250	2"	1¼"
2000	28⅜	22⅛	11⅞	21½	2.9	1/6	1150	2500	148,000	114	320	2"	1¼"
2025	33½	22⅛	11⅞	21½	3 6	1/6	1150	2875	177,000	115	370	2"	1¼"
525	35¼	27½	13	28	4 5	1/2	1150	4200	166,400	94	390	2"	1¼"
2504	35¼	27½	13	28	4.5	1/4	1150	3200	210,000	118	420	2"	1¼"
2500	35¼	27½	13	28	4.5	1/2	1150	4200	225,000	108	440	2"	1¼"
2530	37¼	27½	13	28	5 3	1/2	1150	4650	282,000	115	530	2"	1¼"
530	39⅛	32⅝	13	29	6 5	1/2	1150	5300	260,500	105	600	2½"	1¼"
3000	39⅛	32⅝	13	29	6 5	1/2	850	6350	341,000	109	690	2½"	1¼"
3000	39⅛	32⅝	13	29	6.5	1½	1150	8100	394,000	104	725	2½"	1¼"

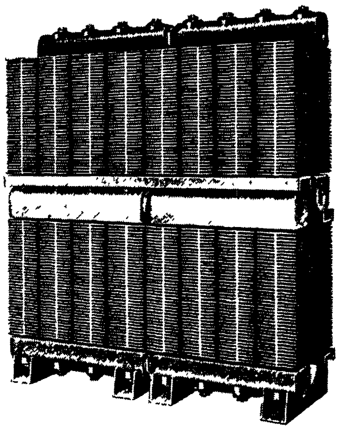
*Varies with type of motor.

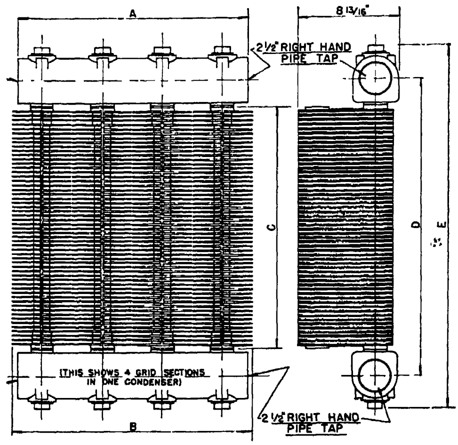
GRID UNIT HEATERS ARE NOT AFFECTED BY ELECTROLYTIC ACTION

No leaks—no breakdowns.
Low maintenance expense.
More air changes per hour.
Positive "directed" heat.
Lower outlet temperatures.
Larger air volume.
Reduced fuel cost.
Applicable to either low or high steam pressure lines.

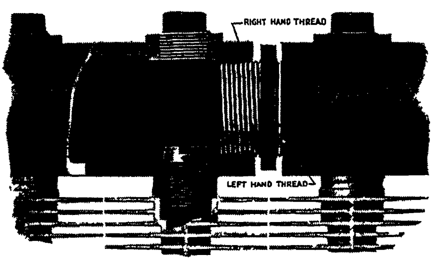
Send for Bulletin on Units not listed above.

GRID BLAST COILS





FACE VIEW SIDE VIEW



No Electrolysis to cause corrosion. All-cast aluminum heating sections. Only one type metal in contact with steam. Therefore, no electrolytic action is developed with its resultant corrosion to cause leaks and breakdowns. Grid Blast Coils are *different.*

Cutaway section above shows why Grid Units are different — and how manifold is connected with steam chamber—a simple and efficient connection. Guaranteed for pressure up to 250 lb steam pressure with high temperatures. Compact and occupies less space than other radiation of equal capacity. No leaks—no breakdowns. Low maintenance expense. No tortuous air passages; hence, easy to clean at all times. Free area is much greater than other types of coils. Send for complete data.

GRID BLAST COILS DATA

Model	Face Area Sq Ft	No. of Sections Wide	Dimensions, Inches					Shipping Weight Lb
			A	B	C	D	E	
B215	1.15	Two	9⅜	10¼	16⅛	20½	26½	127
B315	1.72	Three	14½	15⅜	16⅛	20½	26½	184
B415	2.30	Four	19⅝	20½	16⅛	20½	26½	236
B515	2.88	Five	24¾	25⅝	16⅛	20½	26½	288
B615	3.45	Six	29⅞	30¾	16⅛	20½	26½	346
B220	1.50	Two	9⅜	10¼	21⅛	25½	31½	181
B320	2.25	Three	14½	15⅜	21⅛	25½	31½	223
B420	3.00	Four	19⅝	20½	21⅛	25½	31½	293
B520	3.76	Five	24¾	25⅝	21⅛	25½	31½	358
B620	4.52	Six	29⅞	30¾	21⅛	25½	31½	424
B225	1.86	Two	9⅜	10¼	26⅛	30½	36½	194
B325	2.80	Three	14½	15⅜	26⅛	30½	36½	267
B425	3.72	Four	19⅝	20½	26⅛	30½	36½	343
B525	4.66	Five	24¾	25⅝	26⅛	30½	36½	443
B625	5.58	Six	29⅞	30¾	26⅛	30½	36½	512
B230	2.22	Two	9⅜	10¼	31⅛	35½	41½	207
B330	3.32	Three	14½	15⅜	31⅛	35½	41½	294
B430	4.43	Four	19⅝	20½	31⅛	35½	41½	390
B530	5.54	Five	24¾	25⅝	31⅛	35½	41½	478
B630	6.65	Six	29⅞	30¾	31⅛	35½	41½	566

THE HERMAN NELSON CORPORATION

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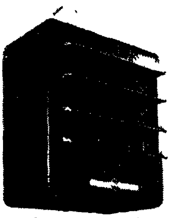
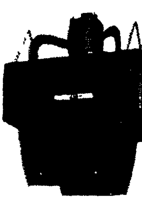


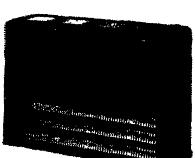
HERMAN HELSON HORIZONTAL SHAFT PROPELLER-FAN TYPE hiJet HEATER

Designed for ceiling suspension, this hiJet Heater projects warm air downward in the desired direction. Incorporates patented stay tube which maintains proper relationship between headers without increasing strain on loops and prolongs life of unit. 48 models, sizes and arrangements.



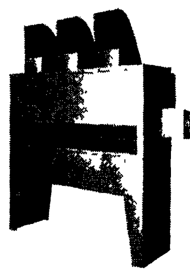
HERMAN NELSON VERTICAL SHAFT PROPELLER-FAN TYPE hiJet HEATER

This hiJet Heater discharges air vertically downward, or at an angle to vertical in various directions. Long life heating element incorporates Herman Nelson's patented stay tube. Unit can be secured with either high or low velocity discharge. 33 models, sizes and arrangements.



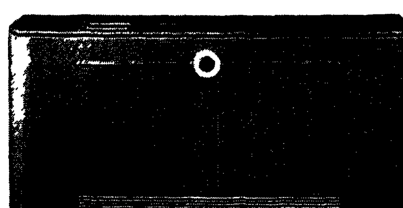
HERMAN NELSON DE LUXE hiJet HEATER

This attractive unit heater is unusually compact and provides excellent distribution and large heating coverage. Incorporates patented stay tube in heating element and light-weight aluminum fan. Her-Nel-Co motor is mounted in end compartment out of air stream. Unit may be placed on floor, wall or suspended from ceiling. 18 models, sizes and arrangements.



HERMAN NELSON BLOWER-FAN TYPE hiJet HEATER

Provides efficient heating of large areas. Can be supplied with by-pass damper if desired. Streamline discharge outlets maintain large air delivery with high velocity. Design of heating element assures durability and contributes to high velocity discharge. For floor, wall, ceiling, or inverted wall mounting. 150 models, sizes and arrangements.



HERMAN NELSON UNIT VENTILATOR

Maintains desired air conditions for school classrooms and rooms in public buildings. Both classroom and auditorium type units have either damper or radiator control. Exclusive "draw-through" design prevents drafts and eliminates overheating. Locating motor in end compartment provides additional space for fan assembly and use of larger fans running at slower tip speeds.

Herman Nelson Unit Heaters and Unit Ventilators are tested and rated in accordance with the Standard Test Code adopted jointly by the Industrial Unit Heater Association and the American Society of Heating and Ventilating Engineers.

THE HERMAN NELSON CORPORATION

Autovent Fan & Blower Division

1809-23 N. Kostner Ave., **Chicago, Illinois**

Member National Association of Fan Manufacturers

AUTOVENT "31 Series" PROPELLER FANS

Exclusive Autovent design—will not overload motor. Ruggedly constructed for economical operation under severe conditions. Available in wheel diameters from 9 to 72 inches; capacities 450 to 45,000 cfm. Write for literature!

ACID-MOISTURE PROOF PROPELLER FANS

For use where corrosive acid fumes or excess moisture exists. Exclusive "31 Series" fan features. Wheels treated with protective coating for average or severe conditions. From 750 to 12,500 cfm in wheel sizes 12 to 36 inches.

VAPOR-EXPLOSION PROOF PROPELLER FANS

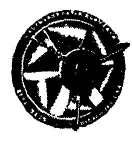
Designed for explosive dust or hazardous fume conditions. Non-ferrous fan wheels. Chemical coatings furnished to requirements. Underwriters Label Class 1, Group D, totally enclosed motors. 12 to 36 in. wheels, 800 to 12,500 cfm. "31 Series" fan construction features.

ALLVENT ALL-PURPOSE FAN

Developed for use in commercial, industrial and public buildings . . . stock motor with "V" belt drive provides maximum efficiency. Non-overloading "31 Series" wheel. Mounted on steel panel for easy installation. Six sizes, 24 to 54 inch wheel diameters with capacities from 5,000 to 23,000 cfm. Commercially quiet operation. All steel construction.

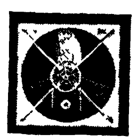
AUTOVENT "V" BELT DRIVE UNIT BLOWERS

Fully self-contained unit including motor, drives and housing; forwardly curved blades; adjustable motor pedestal with vibration dampeners; universal discharge; eight sizes having wheels with diameters of from 12¼ to 30 inches; capacities 1,200 to 8,000 cfm. Compact and sturdy.



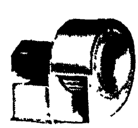
AUTOVENT "BW" PROPELLER FANS

A slow speed operating bucket wheel type fan. Provides efficient ventilation and quiet operation with minimum power consumption. Capacities: 1,000 to 40,000 cfm. Wheel diameters: 16 to 72 inches. Sturdy construction.



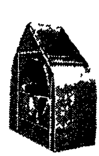
COOLVENT ATTIC FANS

Solve the low cost summer comfort-cooling and ventilating problem for homes with a *quiet* attic fan! Bearings and motor are rubber mounted to insure quiet operation. Sizes 24 to 54 in., 4,070 to 21,100 cfm.



AUTOVENT VOLUME BLOWERS

Compact, direct connected, motor driven unit blowers for general ventilating applications; fume hoods, chemical labs, processing, drying, forced draft, toilet ventilation, etc. Universal discharge. Mount on floor, wall or ceiling. Forwardly curved and backwardly curved blade wheels. Can be furnished with special coatings for acid fume conditions. Available in nine sizes: Wheels 6 to 24½ inches in diameter; capacities from 300 to 6,000 cfm.



AUTOVENT TYPE "H" and TYPE "HB" BLOWERS

For heavy duty ventilating and air conditioning installations. Type "H"—forwardly curved blade wheels; Type "HB"—backwardly curved blade wheels incorporating non-overloading power characteristics. Single or double width, Class 1 or Class II construction; 17 sizes having wheel diameters from 12¼ to 73 inches. Can be furnished to any speed or discharge requirement. Write for new catalog.

The Complete Line of Autovent Propeller Fans and Blowers is tested and rated in accordance with the Standard Test Code adopted jointly by the National Association of Fan Manufacturers and the American Society of Heating and Ventilating Engineers.

John J. Nesbitt, Inc.

Holmesburg, Philadelphia, Pa.

11 Park Place, New York City — 205 W. Wacker Drive, **Chicago, Ill.**

Manufacturers of

THE NESBITT SYNCRETIZER Heating and Ventilating Unit,
sold by John J. Nesbitt, Inc., and American Blower Corporation;
NESBITT HEATING SURFACE with Dual Steam-distributing Tubes,
NESBITT SERIES H HEATING SURFACE, and
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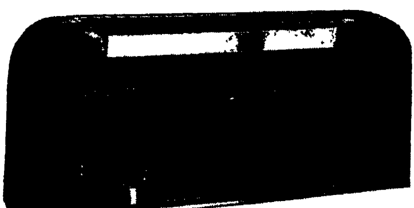
sold by leading manufacturers of fan-system apparatus;
WEBSTER-NESBITT UNIT HEATERS and AIR CONDITIONERS
(See page 1035), distributed in U. S. A. by Warren Webster & Company.



Interior View

The Nesbitt Syncretizer—Series 400

The last word in heating and ventilating units for schoolrooms, offices, etc., where the continuous introduction of outdoor air is desired. For engineering data, get Publication No. 225-1; for "The Story of Syncretized Air," Publication No. 231.

Nesbitt Series B Thermovent

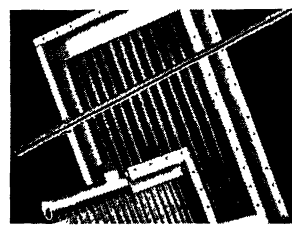
For heating and ventilating auditoriums, gymnasiums, assembly halls, and similar gathering places. Publication No. 227-1.

NESBITT COOLING SURFACE

Series W (Water) Surface with exclusive drain feature

Uncased Surface Showing Drain Header

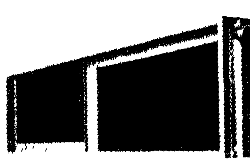
For air cooling and cooling and dehumidifying (with cold water) or air heating (with hot water). Constructed of copper tubes and plate-type aluminum fins. Available in either continuous or cleanable tube type, in single sections having one to eight rows of tubes deep, in three fin spacings, in eleven fin widths, and up to sixteen finned tube lengths. Sturdy galvanized casings. For particulars and engineering data send for Publication No. 233.



NESBITT HEATING SURFACES
With Dual Steam-distributing Tubes

Copper tube-and-fin surface for low-pressure applications. Perfectly adapted to close, continuous automatic control with modulating steam valves. Steam-distributing tubes within the condensing tubes carry the steam equally to the full section assuring UNIFORM discharge temperatures even under a throttled steam supply; eliminating temperature stratification; preventing tube freezing without preheaters; giving ideal system results.

Cased or uncased units of many sizes and capacities. For full particulars and engineering data, send for Publication No. 229-1.

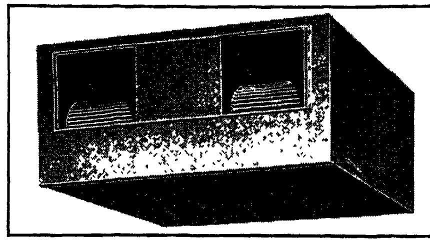


For above advantages plus uniform distribution in extended fin lengths from 80 to 128 ins., specify *Nesbitt Duplex Heating Surface* with Dual Steam-distributing Tubes. Publication No. 236.

Nesbitt Series H Heating Surface

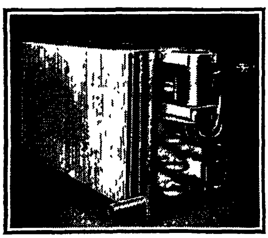
A lightweight, enduring, highly efficient blast-coil heating surface designed for use with steam pressures up to 200 lb gauge. Well suited to high-pressure as well as low-pressure applications. Seven types, each in eight fin widths and up to sixteen finned lengths—a total of 784 sizes from which to select. Send for Publication No. 232 for complete engineering data.

Refrigeration Economics Co., Inc.
Canton, Ohio
RECOY PRODUCTS

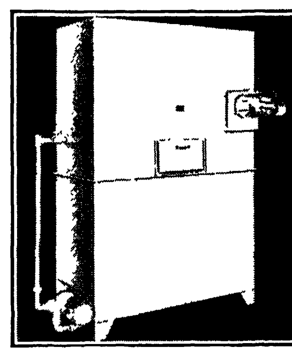


RECOY AIR CONDITIONING UNITS of the suspended type as shown, or vertical floor type, are made for all season purposes, also for summer cooling or winter heating and humidifying.

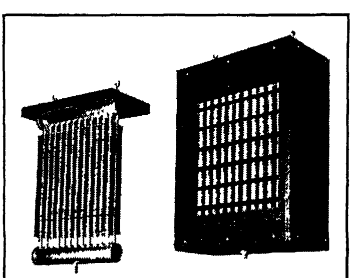
Capacities range from one ton up to any size required. Cooling and heating surface, and filter area are liberally proportioned and blowers are of moderate speed, all to insure the highest efficiency and quiet, satisfactory performance. Bulletin "E".



RECOY CONTINUOUS FIN BLAST COILS for cooling or heating are constructed of copper tubing with aluminum fins, or all steel hot dip galvanized after fabrication and are suitable for use with any cooling or heating medium. Bulletin "F".



RECOY EVAPORATIVE CONDENSERS are cooling towers and condensers combined into one efficient unit for use indoors or out. They reduce the water consumption 95 per cent and are used with no water at all in cold weather. Made in sizes from one to one hundred tons. Ceiling Type 2 to 12½ tons. Bulletin "G".



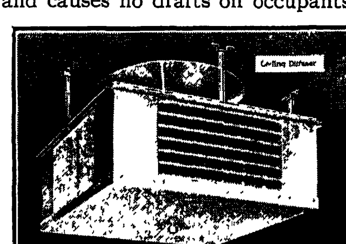
RECOY BLAST HEATERS have all welded coils and headers so will stand any steam pressure and remain tight for years. Coils are copper tubing with aluminum fins or all steel hot dipped galvanized after fabrication. The entire unit is suspended on the top header and coil is free to expand.

Fan motors are oversize to insure continuous service and fans are guarded.

The casings have liberally rounded corners and are beautifully finished in baked crinkle enamel.

Quotations and data on request.

RECOY CEILING DIFFUSERS as illustrated are ideal for air conditioning or refrigeration in that the air is distributed at the ceiling and causes no drafts on occupants.



For heating the construction is reversed and air is diffused radially at bottom.

Fan motors are oversize and ball bearing for continuous, satisfactory service.

Coils are three rows deep, welded and made of copper tubing with aluminum fins or all steel hot dipped galvanized after fabrication, and will stand any pressure and remain tight for years.

Casings are heavy steel attractively finished with baked on crinkle enamel.

Quotations and data on request.

The Trane Company

2021 Cameron Avenue, La Crosse, Wisconsin

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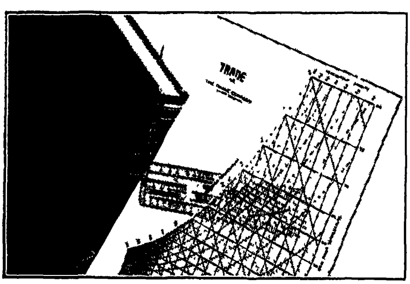
TRANE EDUCATIONAL MATERIALS

Trane Air Conditioning Manual (New, Enlarged Edition)



Trane offers the engineering profession a comprehensive, straightforward and unbiased textbook covering the fundamentals of air conditioning. Trane engineers have gathered all available material, sifted and analyzed it carefully to produce in one volume the essence of air conditioning practice. The Trane Air Conditioning Manual not only shows how to design every type of air conditioning system, but also clarifies underlying principles enabling both the student and the engineer to reason out their own problems rather than to blindly follow complicated formulas. Price —$5.00.

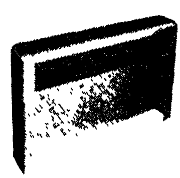
Trane Air Conditioning Ruler and Psychrometric Chart

To solve air conditioning problems with speed and accuracy, The Trane Company has developed the Air Conditioning Ruler and Psychrometric Chart. It eliminates the laborious calculation entailed by outmoded methods—saves two-thirds of your time in figuring air conditioning problems.

TRANE PRODUCTS

Trane Convectors

In using the Trane Convector it costs no more for the smooth, steady flow of clean, even heat obtained. It costs no more for a lighter, yet sturdier, unit which has superior heat transfer ability. You pay no bonus for the harmonious design of this clean, space saving method of heat diffusion for all steam and hot water heating systems. Trane offers a complete line for both visible and concealed installation.



Trane Warm Water Heating

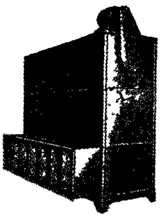
Trane Warm Water Heating Systems bring low cost luxury and comfort to the most modest residence. The Trane Circulator, Trane Flo Valves and Fittings make the hot water heating system function properly and economically. Unique in design and application to achieve lower initial cost of complete heating systems. Ideal for defense housing.

Trane Unit Heaters

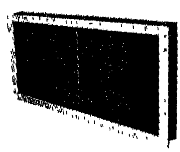
Trane pioneered and introduced the original projection type unit heater—the Trane Projection Unit Heater —which accomplishes a superior diffusion of heat from high or low ceiling installation. The broad range of Trane Projection, Propeller and Blower Type Unit Heaters enables Trane to make unbiased recommendation of the unit best suited to do your job, large or small.

Trane Climate Changers

The Trane Climate Changer line affords a unit selection for practically every known air conditioning purpose. Five major types possess a diversity of application possibilities as well as a capacity of range from 250 to 20,000 cfm. Units may be arranged for complete summer and/or winter air conditioning and may be equipped for either steam or hot water heating, and for cooling with water, brine or direct expansion refrigerant. Literally scores of sizes and styles are available for floor, suspended or concealed installation—comfort or process—in all types of buildings.

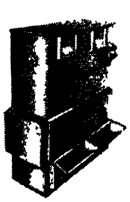


Trane Coils



The integral fin-and-tube construction of the Trane Coil provides complete heat transfer for all heating, cooling, drying, and air conditioning services. Trane has several thousand different styles and types of coils to meet all requirements. Types include: high and low pressure steam coils; hot and cold water coils; blast coils; drying coils; direct expansion coils; coils of special materials for special gases or liquids; easy-to-clean coils for water containing foreign matter; coils for installation in units, in ductwork, or for separate service.

Trane Cooling Equipment



Trane manufactures a complete line of Evaporative Condensers, Evaporative Coolers, Product Coolers, Brine Spray Units, Comfort Coolers, Railroad and Bus Air Conditioners, and Radio Tube Coolers. The Trane Evaporative Condenser (shown) saves up to 90 per cent in water costs for refrigerant condensing. Trane Cooling Equipment is available in the correct size or type to meet any need.

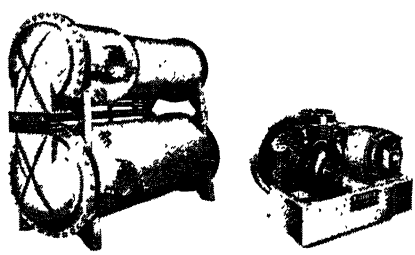
Trane Dry Blast Systems

Typical of many Trane arrangements available for industrial processes is this system which increases the efficiency and achieves uniformity of blast furnace operations.

Trane Heating Specialties

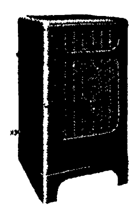
There are over fifty Valves, Traps, Vents, Strainers and allied specialties in the Trane Heating Specialty line. Designed to afford protection and accurate control to the steam, vapor, or vacuum heating system, Trane Specialties are expertly fabricated of select materials to insure top performance and create definite fuel savings. Trane Heating Specialties meet the rigid specifications of the United States Navy.

Trane Refrigeration Units



The Trane Turbo-Vacuum Compressor is a hermetically sealed, centrifugal type water chiller which operates under low pressure. It is a complete "equipment room" in one compact package. Only two moving parts and 25 per cent lighter than comparable types. Sizes 50, 70, 100, 200 T. Trane also manufactures a complete line of Reciprocating Compressors available in sizes from 3 to 50 tons capacity. Built for long term performance.

Trane Gas-Fired Equipment



Trane manufacturers a complete line of gas-fired equipment, including Gas Space Heaters, Unit Heaters, and Horizontal and Vertical Gas-Winter Air Conditioners. These units embody a light-weight heat generator and heat exchanger—speedy heat maker—safe and automatic.

Other Trane Equipment

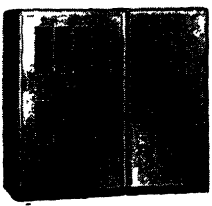
Handsome, sturdy Self-Contained Air Conditioners in 3, 5, 7½, 10 and 15-ton capacities. Also, Condensation, Circulating and Booster Pumps, Temperature Control Valves, Air Washers, Spray Nozzles, Fans, Unit Ventilators and Dehumidifiers. Bulletins available on all Trane Products. Countless national defense applications.

Utility Fan Corporation

4851 S. Alameda St., **Los Angeles, Calif.**

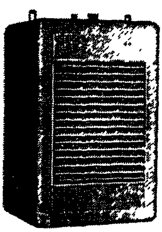
Utility Gas-Fired Heating Equipment, Evaporative Coolers, Blowers and Fans

FORCED AIR FURNACES



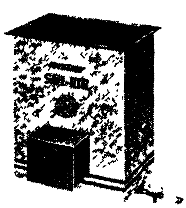
Basement and closet types . . . Compact design . . . Multiple-fin element with air-cooled hollow baffle . . . Element guaranteed against burnout forever . . . Automatic controls . . . Filters . . . Built-in motor overload protection.

UNIT HEATERS



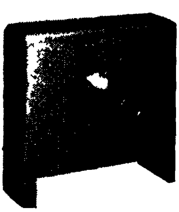
Compact heat exchanger with no inside baffles—Individual burners for each element section—Silent, disc-type fan—All-welded cabinet—Directional grilles—Built-in draft diverter—Temperature limit control. Four sizes.

FLOOR FURNACES



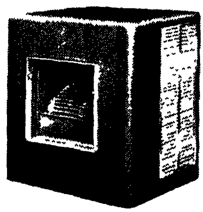
All-welded construction . . . Die-stamped grilles . . . heavy cast-iron burner . . . removable inside jacket . . . interlocking gas valve. Floor or dual registers. 25,000, 37,000 and 50,000 Btu input.

CIRCULATING HEATERS



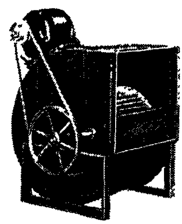
Fan sends stream of air through nozzle-shaped outlet to hold warm air in living zone. Built of heavy furniture steel . . . all die-formed and electric welded. Vented and unvented models.

EVAPORATIVE COOLERS



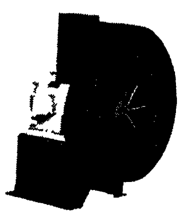
Comfort cooling—residential, commercial, industrial. Exlcusive feature—Uni-flowmeter (Pat. Pend.) for uniform water distribution. Patented No-Sag cooling pads. 14 Models.

BLOWERS



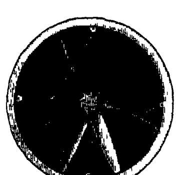
Complete line. Dynamically balanced, multiple-vane centrifugal blowers—standard and heavy duty. Wheel diameters 6 to 66 in.; widths 6 to 66 in.

EXHAUSTERS



High flow and high pressure designs in four drive arrangements, for exhausting many materials. High efficiency . . . decreased sound. Available with acid-proof plastic surfacing.

PROPELLER FANS



Airplane propeller type for high efficiency, economy and maximum air delivery. 2, 4 and 8 blade models—12 in. to 30 in. diameter; 300 to 250,000 cfm.

Utility Blowers are tested in accordance with the A.S.H.V.E. Code.

Write for complete information, catalogs and prices.

Factory:
NEWARK, N. J.

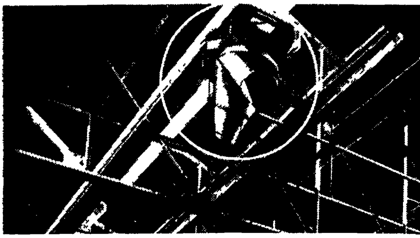
L. J. Wing Mfg. Co.

59 Seventh Avenue, **New York, N. Y.**

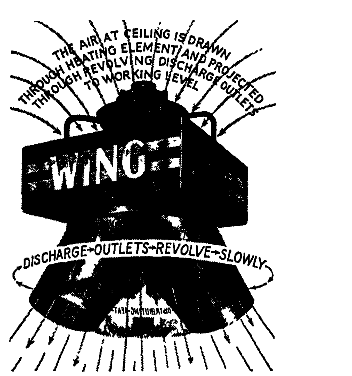
Canadian Factory:
MONTREAL

Branch Offices in Principal Cities

WING REVOLVING UNIT HEATERS




89 *WING Revolving Unit Heaters are installed in the Curtiss-Wright Plant, Buffalo, N. Y. shown here*



This innovation in the method of distributing heat produces a sensation in heating comfort never before attained—a sensation of fresh, live, invigorating air.

The fact that the outlets revolve assures uniform and thorough distribution of comfortably warmed air throughout the entire working area, without drafts, hot spots or cold spots.

Such an unprecedented high efficiency in distributing heat is the result of nearly 20 years of constant study by Wing engineers to improve on the Floodlight System of heating originated by WING in 1920. This method projects the heated air vertically downward by means of light-weight, ceiling-suspended unit heaters.

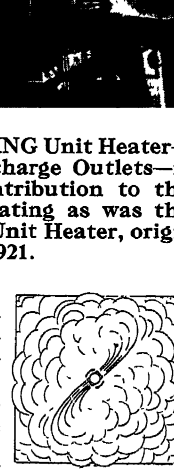
It has needed only this latest refinement of slowly revolving discharge outlets to bring that method to perfection.

The WING Revolving Discharge type supplements the WING line of standard fixed discharge outlets, illustrated and described on the following page.

Bulletin HR-1.

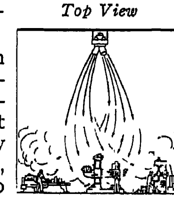
The latest type of WING Unit Heater—with Revolving Discharge Outlets—is just as great a contribution to the art of industrial heating as was the Ceiling-Suspended Unit Heater, originated by WING in 1921.

The area covered by a WING Revolving Unit Heater is slowly swept by the heated air discharged by the outlets which move through an arc of 360 deg. covering every direction of the compass successively.



Top View

By maintaining an active, constant circulation of air throughout an industrial plant at all times, a new sensation of refreshing, invigorating comfort to workers is produced.

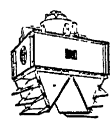


Elevation

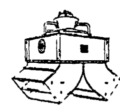
WING FEATHERWEIGHT UNIT HEATERS

Type "HC" Fixed Discharge

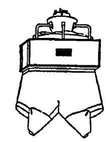
The first light-weight, ceiling-suspended, unit heater. Eight different designs of outlets meet the requirements of every type, size and height of building or occupancy. Located near ceiling or roof, the accumulation of hot air in the upper spaces, with the accompanying costly waste of heat, is prevented. They project the air, comfortably warmed, downward to the working area. *Bulletin H-9.*



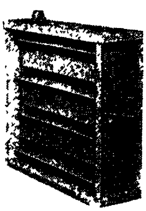
Design No. 3



Design No. 4

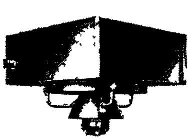


Design No. 8



VARIABLE TEMPERATURE SECTIONS

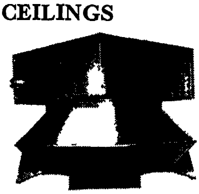
Invaluable in supplying fresh air for space heating or process work. Close control of the delivered air temperature is obtained without danger of freezing. Manual or automatic control. *Bulletin HS-2.*



DOOR HEATERS GARAGE HEATERS

WING originated the vertical cone-discharge heater in 1921 and today it is still applicable for heating the inrush of cold air at large doorways and for garage heating. Often cuts heating costs in half. *Bulletins D-1 and G-1.*

FOR LOW CEILINGS



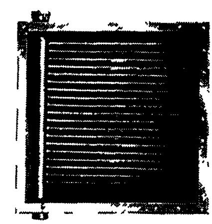
Type "LC"

In this type of WING Unit Heater the position of fan and motor are reversed to meet conditions of ceiling or roof height, form and shape of building, coverage, etc. *Bulletins HR-1 and H-9.*

WING UTILITY UNIT HEATERS

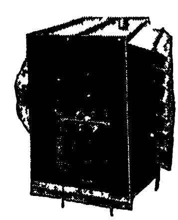
A lightweight suspended unit heater for delivering heated air in one general direction. Has the same powerful fan and rugged heating element as WING Featherweight Unit Heaters. This is the latest refinement of the original horizontal lightweight heater which was developed by WING. *Bulletin U-5.*

FEATHERFIN HEATER SECTIONS



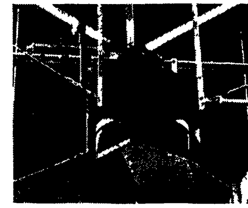
For heating or cooling air for any purpose by steam, hot or cold water or refrigerant. The heating element is extremely light and, for equal heat transfer, offers little resistance to air flow. Available for any desired final air temperature. *Bulletin HS-2.*

WING INDUSTRIAL FOG ELIMINATORS



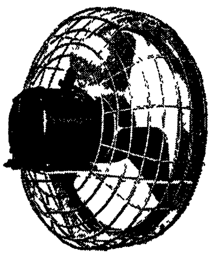
Eliminate fog, odor and fumes in dyeing, bleaching and finishing plants, creameries, pasteurizing, bottling, canning and packing plants, chemical works, paper mills, steel pickling plants, etc. No ducts are required. *Bulletin FE-12.*

TURBINE-DRIVEN HEATERS



Any WING Unit Heater can be furnished with steam turbine-driven fan for locations where high-pressure steam is available. Photo shows turbine-driven revolving unit heater. Can also be supplied for fixed discharge or utility type heater. *Bulletin HR-1 and H-9.*

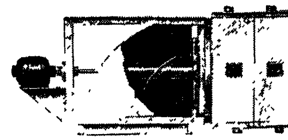
WING-SCRUPLEX SAFETY VENTILATING FANS



A propeller type fan that will deliver air against static pressure, quietly and efficiently.

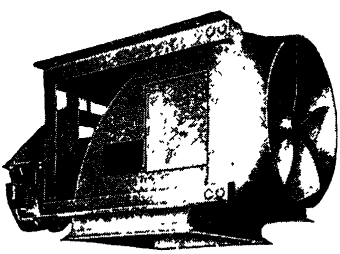
Moves the air forward in straight lines with minimum eddy. Capacities to 100,000 cfm. *Bulletin F-8.*

WING FEATHERFIN PROCESS HEATING UNITS



For manufacturing processes such as drying, aging, etc., requiring the recirculation of the heated air Motor or turbine located outside air current. *Bulletin P-2.*

WING-SCRUPLEX EXHAUSTERS



For economically moving air wherever ducts are used. It combines the efficient WING-Scruplex Propeller Fan with a housing which places the motor entirely outside the air duct. Motor and drive remain cool and clean and are easily accessible.

The powerful WING-Scruplex Fan delivers high air volume with low power consumption against any pressures for which duct systems should be designed. V-belt or direct drive.

Light, compact and easy to install. *Bulletin 78-A.*

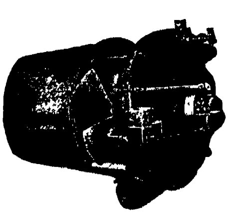
WING SYSTEM OF CONTROLLED COMBUSTION

For low pressure heating boilers and small power boilers. Increases capacity and permits use of lowest cost fuel. Includes Type EM Blower equipped with fully enclosed dustproof motor with speed regulating rheostat and automatic control. Eliminates necessity of frequent firing, allowing intervals as great as 24 hours even in zero weather. *Bulletin M-96.*

Installation of Wing System of Controlled Combustion in a large school

WING TURBINE-DRIVEN BLOWERS

Applied to hand, stoker, oil or pulverized fuel fired boilers, increase boiler capacity, maintain constant steam pressure and permit complete combustion of low-cost fuels. The exhaust steam, free from oil, can be used for heating or processes. *Bulletin T-98.*



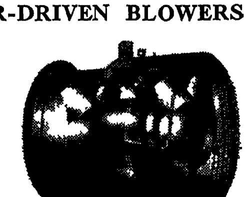
WING DRAFT INDUCERS

Installed in breeching or flue, or on chimney top; provide positive, exact draft regardless of weather conditions or inadequate chimney or breeching construction. Suitable for coal, oil, or gas-fired boilers; industrial furnaces and kilns. *Bulletin I-10.*

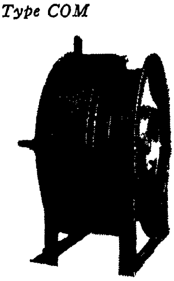
Chimney-Top Installation

WING MOTOR-DRIVEN BLOWERS

Type COM for static pressures up to 10 in. W. G. and volumes up to 50,000 cfm. Type EMD for moderate static pressures up to 2½ in. Both blowers have fully-enclosed dustproof constant speed motor and built-in adjustable control vanes. Type COM has double-staged axial flow fan; Type EMD, single stage fan. Extremely compact; discharge can be vertical, horizontal or inclined. *Bulletin CO-4.*



Type COM



Type EMD

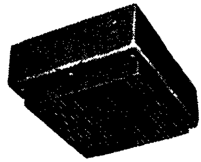
Young Radiator Co.

Plant and Executive Offices

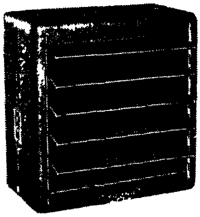
709 Marquette St., **Racine, Wis.**

Sales and Engineering Offices in Principal Cities

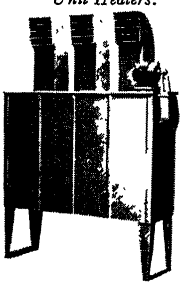
REG. U S PAT OFF



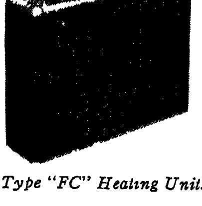
Vertiflow Unit Heaters.



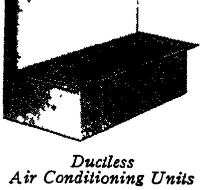
Horizontal Discharge Unit Heaters.



Blower Fan Type Unit Heaters.



Type "FC" Heating Units



Ductless Air Conditioning Units

YOUNG *High Efficiency* UNITS

A complete line of heating, cooling and air-conditioning units. Catalogs with complete information and engineering data will be mailed on request.

VERTIFLOW UNIT HEATERS. Propeller Fan Vertical Discharge Type units for ceiling suspension. Heating element split and pitched for proper draining. Eight types of diffusers. 12 models, capacities 30,000 to 480,000 Btu per hour.

HORIZONTAL DISCHARGE UNIT HEATERS—32 sizes of single and twin propeller fan units. Capacities from 18,000 to 658,000 Btu per hour.

BLOWER FAN UNIT HEATERS—Ten models, capacities 120,000 to 800,000 Btu per hour. Also for ceiling suspension.

TYPE "FC" HEATING UNITS—Floor model unit heater in attractive cabinet for offices, corridors, etc. Two fans—operates on steam or hot water.

DUCTLESS AIR CONDITIONING UNITS—Recessed room unit for use with steam or forced hot water systems. Heats, filters, humidifies and circulates air.

"STREAMAIRE" AIR CONDITIONING UNITS—For complete year-round air conditioning. Also available for winter or summer conditioning only. Eight horizontal and eight vertical models. Capacities from 400 to 16,600 CFM.

"STREAMAIRE" CONVECTORS. Six types of cabinets, high efficiency heating element. Designed for any type of steam or hot water system.

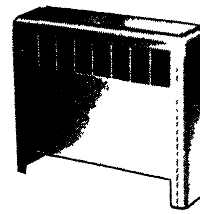
COMMERCIAL HEAT TRANSFER UNITS—Extended heating or cooling surface for air conditioning units, dryers, etc.

WATER COILS—Continuous tube coils for heating or cooling with water. Cleanable and Drainable types available.

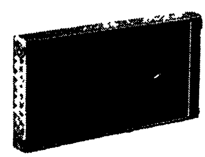
BLAST UNITS—An encased surface for forced air heating or cooling systems.

EVAPORATOR COILS—For mechanical refrigeration systems using Freon or methyl chloride.

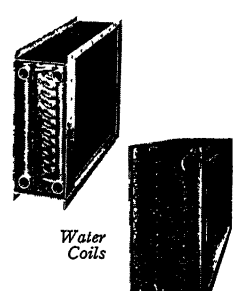
"Streamaire" Air Conditioning Units.



"Streamaire" Convectors.

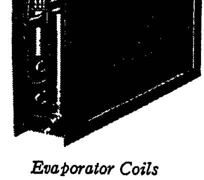


Commercial Heat Transfer Units.



Water Coils

Blast Units



Evaporator Coils

AIR SYSTEM EQUIPMENT

Air systems for heating, cooling and ventilating services are produced by grouping various machines and accessories, each performing a function in the complete cycle of the desired operation. The essential parts and accessories described by the manufacturers are contained in the following groups:

AIR FILTERS AND CLEANERS (p. 918-933)

Mechanical and electrical methods of filtering, also air washing and purifying apparatus and their applications.
Technical data on this subject will be found in Chapter 29.

HUMIDIFYING UNITS (p. 934-937)

For supplying moisture to air and controlling its volume as desired for industrial and commercial uses, or for comfort requirements.
Technical data is contained in Chapter 24.

COOLING TOWERS AND SPRAY EQUIPMENT (p. 935-938)

For cooling and reclaiming water used in industrial processes and air conditioning.
Technical data will be found in Chapter 27.

HEAT TRANSFER SURFACES (p. 938-942)

As parts of heating and cooling units, and for separate use in industrial and commercial heating and cooling systems.
Technical data is contained in Chapter 26.

CONDENSING UNITS AND REFRIGERATING MACHINERY (p. 943-951)

For refrigerating processes and for cooling purposes in industrial, commercial and comfort air conditioning service.
Technical data will be found in Chapter 24.

FANS AND BLOWERS (p. 952-964)

For use as separate air circulating equipment, or as parts of heating and air conditioning units.
Technical data is contained in Chapters 23 and 30.

MOTORS (p. 965-967)

Used in conjunction with blowers, fans, stokers, oil burners and other heating, cooling and air conditioning apparatus.
Technical data on motors will be found in Chapter 36.

REGISTERS AND GRILLES (p. 968-981)

Air diffusion equipment for use with heating, ventilating and air conditioning systems.
Technical data relating to this equipment is contained in Chapters 31 and 32.

SHEET METAL AND TUBULAR PRODUCTS (p. 982-985)

Sheets for air ducts and enclosures; pipes for gas, refrigerants, steam, water, etc.
Technical data on pipe and piping is contained in Chapters 15 and 18.

Manufacturer's products shown in this division are designed for specific applications. Consult the Index to Modern Equipment for additional products of these manufacturers.

The Air-Maze Corporation

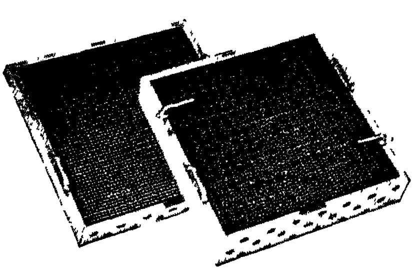
5202 Harvard Avenue, **Cleveland, Ohio**

ENGINEERS AND MANUFACTURERS OF AIR FILTERS EXCLUSIVELY

DIRECT FACTORY REPRESENTATIVES IN ALL INDUSTRIAL AREAS.

DISTRIBUTORS IN PRINCIPAL CITIES AND TOWNS THROUGHOUT THE UNITED STATES.

During more than a decade devoted exclusively to air filter engineering and manufacturing, a great deal about the control and elimination of dust, pollens and grit has been learned by AIR-MAZE engineers. Their design and development of a unique type of filter element construction, embodying distinctive advantages, has been considered a worthy contribution to the air filtering science and has resulted in wide acceptance of AIR-MAZE air filters in all fields of application.



2 in. Thick Panel *4 in. Thick Panel*

AIR-MAZE Permanent Cleanable Panel Filters

Note Advantages Made Possible by Air-Maze Scientific Construction:

Costs Little to Clean—The separating layers and exact spacing of baffles permit free washing action between and around all baffles. Thus, cleaning and charging operations may be easily and economically performed.

Great Dust Capacity—Unique design of the AIR-MAZE screen wire element provides a vast area of baffles on which collected material can become impinged; thus great capacity is assured.

Vibration Proof—Vibrations in service cannot shake filter media out of position—the uniform density remains *permanently* perfect; **no replacements are necessary!**

AIR-MAZE Are Listed by Underwriters' Laboratories—When serviced according to the methods approved by the Underwriters' Laboratories, AIR-MAZE panel filters are approved as fire resistant air filters.

Efficiency—Tests under varying conditions, both in laboratories and field operations, show air filtering efficiency of from 98.00 to 99.83 per cent with practical dust.

No Clogging — Because AIR-MAZE panel filters are easy to *completely* clean and since the exact density enables uniform deposit of dust, no clogging can occur.

Adaptibility—In addition to air conditioning and power equipment installations AIR-MAZE panel filters are effectively used in humidifiers, water eliminator units, paint spray-booths, oil separators, range canopies in kitchens, and other applications where specific problems and unusual requirements are easily handled by adaptations of the panels. AIR-MAZE panels will be made to fit frames of existing installations and can be furnished with locking handles and latches, snap catches, or with flanged edges and lift handles.

Magnified Section of "Loaded" AIR-MAZE Air Filter Element. Note that dust has been quite evenly impinged on the wires. No obstructed spaces can be seen. This feature accounts for the Low Pressure Drop and Non-clogging characteristics of AIR-MAZE

TECHNICAL INFORMATION

Sizes—All sizes and thicknesses are available; two and four inch thick panels are the accepted standard. Installations using large sizes of these permanent panels are surprisingly low in cost.

Capacity—Recommended air capacity is 1½ to 2½ cfm per square inch. Thus, the capacity of a 20 x 20 in. panel is 600 to 1000 cfm. Normally, 2 cfm per square inch should be used.

The Air-Maze Corporation

5202 Harvard Avenue, **Cleveland, Ohio**

Resistance—For 2 in. thick panels the resistance varies from 0.08 in. to 0.10 in. H_2O when handling 2 cfm per square inch of filter area (288 fpm velocity); and for 4 in. thick filters the resistance varies from 0.121 in. to 0.140 in. H_2O at 2 cfm per square inch (288 fpm velocity); the variation being in accordance with the different types of filter media construction available. To obtain specific restriction data write for graphs.

Construction—AIR-MAZE filters are of patented construction consisting of a maze of alternately placed and exactly spaced crimped galvanized wire screens of selected meshes; these are arranged with precision so as to create graduated and progressive density, and to positively embody the baffle impingement principle. The filter element is enclosed in a heavy gauge metalescent enameled steel frame having an open end to simplify servicing.

EASY TO CLEAN AND CHARGE

Wash out filtered matter in a pan of hot water or under a stream of hot water.

Cut-away View

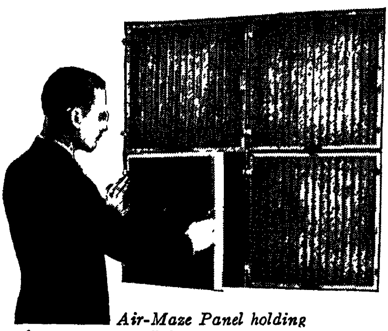
From a flat surface raise one end and let it drop sharply several times. This facilitates drainage.

After cleaning and also after charging, set panel on edge, with open end down, to drain.

Cleaning—Simply tap panel a few times on a hard surface to remove heavy accumulations and then wash under a stream of hot water or in a pan of hot water. Steam also cleans the panels quickly and effectively. Be sure filter is dry before charging.

Charging—(For general applications) Spray both front and back of panel with just enough oil to coat the wires. Any inexpensive oil of S.A.E. 40 or 50 viscosity is suitable. An ordinary hand spray gun will do the work splendidly. Or, if desired, panel may be immersed in oil and then thoroughly drained.

AIR-MAZE INSTALLATION FRAMES



Air-Maze Panel holding frames assure efficient, attractive installations.

AIR-MAZE panel holding frames are constructed of metalescent enameled heavy gage steel having ¾ inch flanged back edge. A thick felt lining on inside of flange insures against air leakage when panels are in place. One frame may be used alone in single panel installations, or a group of frames may be supplied, fixed together; thus a large bank of filter panels may be provided. Every 2 in. frame section is fitted with snap catches as standard equipment; a lift handle is installed on each panel; 4 in. panels and frames have *locking handles*.

In determining frame sizes, ⅝ inch is allowed over the EXACT width, and ⅝ inch over the EXACT height dimensions of the panels. These dimensions include frame edge, clearance and felt edge seals.

Specify AIR-MAZE—for all air filter installations and you will be assured of *efficient, economical* **performance.** Write for specification bulletin CCC-69.

Engineering Service Available—The Air-Maze Engineering Department will gladly offer installation suggestions for special air filter applications.

Other AIR-MAZE Products—In addition to the panel types, Air-Maze Corporation also manufactures a complete line of circular shaped air filters for use in various Railroad, Industrial and Automotive applications.

Literature Available—Catalog GPC-740 describing industrial types "A," "B," Greastop, and Kleenflo panel filters. Catalog describing Air-Maze Oil Bath type, Multimaze and Unimaze filters for internal combustion engine, air compressor and blower applications.

AMERICAN AIR FILTER COMPANY INC.

INCORPORATED

673 Central Avenue, **Louisville, Ky.**

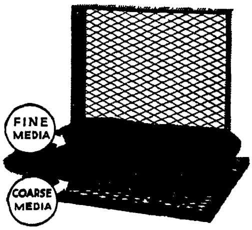
Representatives in Principal Cities

Dust Engineering—Dust Engineering is that branch of applied science which deals with the origin, nature and characteristics of the small solid air-borne particles called "dust," and the development of methods, processes and apparatus for its control or elimination.

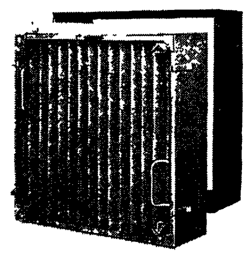
The American Air Filter Company, Inc., has had an important part in advancing the science of Dust Engineering. The efforts of its Research and Engineering Staff for the past twelve years have been devoted exclusively to the study of dust problems and the development of a complete line of air cleaning equipment for modern air conditioning, building ventilation and the control of process dust in industry.

American Air Filter products, therefore, not only embody the knowledge accumulated from years of constant research and the experience gained from designing, building and applying thousands of air filters, but are backed by ample technical and financial resources to insure their outstanding position in the Dust Engineering field.

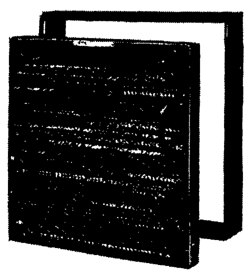
Products—American Air Filters are available for every condition, with operating characteristics and efficiencies to suit specific problems. In general, there are two distinct types based upon the "viscous film" and "dry mat" principles. Each type is made in several styles which differ in method of operation, servicing, space required and initial cost to meet the various conditions encountered in air cleaning problems. A discussion of various filter types will be found in the Technical Data Section under "Air Cleaners."



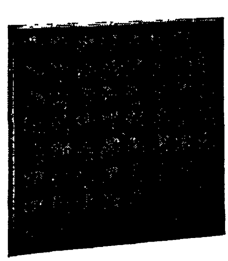
Renu-Vent Filter



Airmat Type PL-24 Filter



M/W 2 Filter



Throway Air Filter

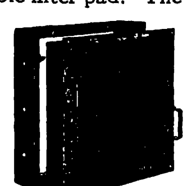
Air filters are generally used for the removal of dust, dirt, bacteria and other foreign matter from the air and are applied to general ventilation, modern air conditioning, process dust control; for air compressors and Diesel Engines; mill motors, turbo-generators and other electrical applications; and for air or gas under pressure to remove entrained oil, moisture and dirt.

Air Filters In Air Conditioning—Filtered air is today recognized as essential in modern air conditioning. There are other important factors which contribute to our comfort such as temperature, air movement and humidity, but science today emphasizes the prime necessity of pure air for health and efficiency.

Air cleaners have, of course, always been considered an integral part of large central systems. These are usually of the fully automatic type such as the American Automatic Self-cleaning filter or the self-cleaning Electro-Matic.

There are now available to manufacturers of unit air conditioners moderate priced unit filters such as the Renu filter, the Throway filter, and other types of filters illustrated on this page.

The Renu filter is an entirely new departure in air filter construction. It consists of a permanent metal frame provided with a removable cover and renewable filter pad. The cover



Standard Viscous Unit Filter

is easily removed without the use of tools, and filter pad can be lifted out and replaced with a new one at very small expense.

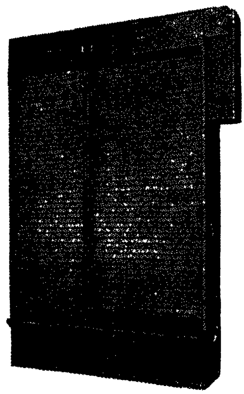
The Throway filter, as the name implies, is designed to be discarded after it has served its maximum period of usefulness and replaced with a new filter unit. The Filter pad is enclosed in a perforated cardboard container which makes it possible to readily dispose of the dirty filter by burning it.

There is probably no single item which costs as little and may mean as much in the design of an air conditioner as air filtration. These units are furnished in any dimensions or shapes desired—usually in units handling 400 cfm and from 2 in. to 4 in. thick. They are usually made in the following sizes—20 x 20 in., 16 x 25 in. and 16 x 20 in. High cleaning efficiencies can be secured, with a resistance to air flow ranging from 1/16 in. to 3/8 in. water gauge.

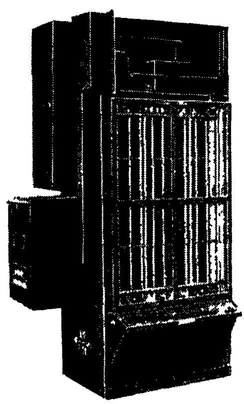
Automatic Self-Cleaning Air Filters—The American line of automatic air filters is among the most complete ever offered. Proved in principle and performance by years of actual service.

The more general use of thermoplastic finishes for refrigerators, stoves, automobiles, and other metal products has created the need for clean air in finishing rooms. This type of finish is hardened by baking, so the product on which it is used must be protected from contamination by dust and dirt from the time it is sprayed until it leaves the oven.

Spray booths exhaust large quantities of air, and if this air is drawn from other parts of the plant, it will contain considerable dust and dirt. If dirt and dust particles are permitted to settle upon freshly sprayed surfaces, they will be trapped in the semi-tacky coating and cause blemishes in the finished product.



American Automatic Self-Cleaning Filter



Electro-Matic Air Filter

This trouble can be eliminated only by enclosing the finishing room and installing a filtered air supply system with sufficient capacity to provide a constant supply of clean air in excess of the volume exhausted by the spray booths.

High efficiency air filters are needed for this service to minimize rejects and do-overs. The automatic self-cleaning filter has proved the most practical type and is widely used for this application because of its ability to maintain a constant, uniform air volume with the minimum of attention.

American Automatic Filters can be furnished with either Multi-Panel, Type "MS," or Double Duty Type "DD" panels, depending upon the service which they are to perform. Complete engineering data is available.

Electro-Matic Air Filter—Incorporates electrical precipitation as an integral function of an automatic self-cleaning viscous filter to obtain a higher over-all efficiency in dust removal. Its higher efficiency as an air cleaning unit, is due principally to the collection of the finer dust particles and smoke, by electrical precipitation. In combination, these two methods of cleaning air not only give the highest efficiency in dust removal but offer operating advantages found only in the automatic self-cleaning filter.

Standard Viscous Unit—The American Unit Air Filter incorporates the time tested unit principle of construction. Each unit consists of a standard steel frame and interchangeable cell equipped with automatic latches to facilitate removal for cleaning and recharging.

Airmat Filter Dry Type—The filtering media in this type is the Airmat sheet, a dry filter mat composed of thin sheets of gauzy, cellulose tissue. The Airmat sheets are supported in screen pockets mounted in a unit frame of box-like construction. These unit frames can be set up to meet any capacity requirement or space condition. Airmat sheets are renewable—their life depending on dust conditions and hours of service.

Airmat filters are used both for comfort and industrial air conditioning. In the latter field they are particularly well adapted for the recovery of valuable dusts and for abating the dust nuisance prevalent in so many industrial plants.

Coppus Engineering Corporation

339 Park Avenue, **Worcester Mass.**

MANUFACTURERS OF AIR FILTERS, STEAM TURBINES, GAS BURNERS, FORCED DRAFT BLOWERS, COOLING FANS

"COPPUS AIR FILTERS PASS CLEAN AIR"

The Coppus Unit Air Filter (patent No. 2050508 and other patents pending) is of the dry type using as filter material all-wool felt. It consists of a distender frame (*C*, Fig. 2), a filter "glove" (*E*, Figs. 1 and 2) and a retainer grid (*B*, Fig. 1). The edges of the retainer grid form a reenforced sheet metal box (*A*, Fig. 1) for protection of the filter element.

The edges of the filter glove are reenforced on all four sides assuring an air tight seal against by-passing of dirty air. By tightening the wing studs which hold the distender frame and the retainer grid together, the filter glove is stretched and held tautly inside of the filter box, giving the pockets a tapered shape so essential for an even air flow.

This design has the advantage of providing an effective filter area entirely unobstructed by wire or screen supports. Cut, Fig. 3 shows the tapered filter pockets on the clean air side. The filter glove can be readily replaced without removing the unit filter from the installation. No auxiliary frames for insertion of the filter cells are required as the completely assembled unit filters can be bolted together to a filter bank of any desired size.

All metallic parts are rust-proofed and Duco Painted.

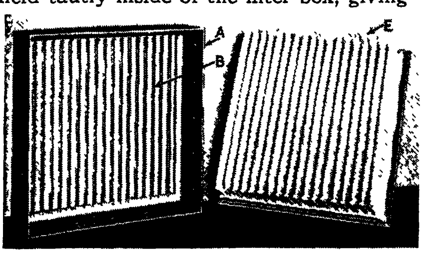


Fig. 1

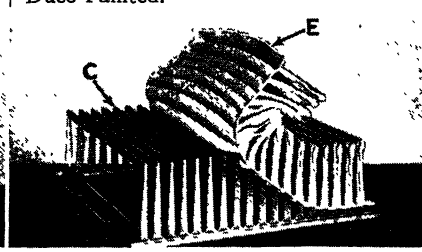


Fig. 2

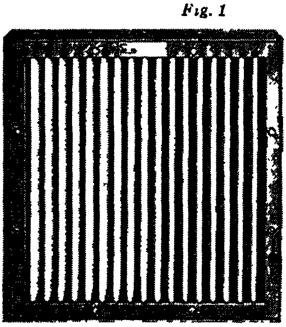


Fig. 3

Specifications

Normal Rating: 800 cfm.
Resistance when clean: 0.2 in. W.G.
Dust Arrestance (cleaning efficiency): 99.61 per cent (Tested in accordance with A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work).
Dimensions: 20 by 20 in. by 5¾ in.
Weight per unit: 25 lb.

ANOTHER COPPUS BLUE RIBBON PRODUCT

Outstanding Advantages

1. It has an exceptionally high dust arrestance.
2. It maintains a high dust arrestance even under diverse conditions of neglect.
3. Its operation is not impaired by atmospheric conditions.
4. It is a Medium Air Resistance Type (Class C) according to the A.S.H.V.E. Code for Air Cleaning Devices.
5. It is easily and quickly cleaned without removing the filter element.
6. Its cost of upkeep is very low because the *permanent* filter element is reconditioned periodically with a vacuum cleaner.
7. It combines scientific knowledge and practical engineering methods with highest quality of material and workmanship.

Cleaning Filter Elements with Portable Vacuum Cleaner

Write for Complete Bulletins

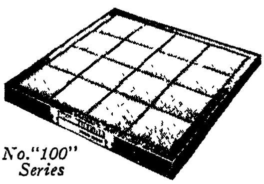
Research Products Corporation

Madison, Wisconsin

RESEARCH AIR FILTERS FOR HEATING AND AIR CONDITIONING

U. S. Patent 2070073

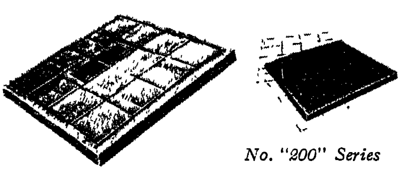
Research Filter With Cardboard Frame



No. "100" Series

Disposable filter, which when dirty, is replaced by entire new unit. For warm air furnaces, air conditioning units, filter banks.

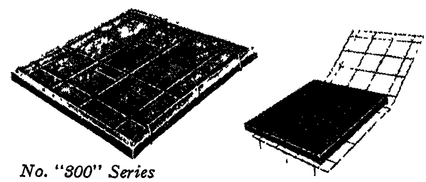
Research Self-Seal Re-Fil-Able Filter With Hooked Wire Grids



No. "200" Series

Filter pad, sandwiched between wire grids which hook together; pad is easily replaced. Used in furnaces, air conditioning units and filter banks.

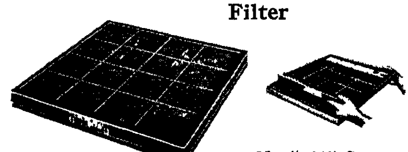
Research Self-Seal Re-Fil-Able Filter With Hinged Wire Grids



No. "300" Series

Easy-to-change Re-Fil-Able filter. Replaceable pad held snugly between hinged wire grids. Used extensively for filter banks.

Research Steel Frame Re-Fil-Able Filter



No. "400" Series

Re-Fil-Able filter with permanent steel frame, complete with wire grids to hold the removable pad in rigid position. New pad easily inserted.

TECHNICAL DATA ON RESEARCH AIR FILTERS

Capacity Rating

Research Air Filters are rated at 2 cfm of air per sq in. of gross nominal area. Recommended maximum air velocity is 400 fpm.

Resistance, Inches of Water	Air Velocity. FPM
.018	100
.065	200
.130	300
.200	400

When Filters are used in tandem the restriction is additive.

A Research Air Filter 20 x 20 x 2 in., when tested according to the test code of the American Society of Heating and Ventilating Engineers has an efficiency of 93 per cent, tested with standard code dust. The dust holding capacity with standard code dust is 150 grams per sq ft of filter area, the restriction at this dust load being .2 in. of water.

RESEARCH "100" AND "400" SERIES 2-INCH AIR FILTERS

Dimensions, Ratings and Manufacturing Tolerances

Nominal Sizes	Ratings	Actual Dimensions			
These dimensions are used by the trade to order filters and refer primarily to the size of holder into which the filter fits.	Volume of Air Cleaned at Velocity 300 F.P.M. C.F.M.	A Width	B Height	C Thickness	D Border
		Tolerances			
		Plus 0.00 In. Minus 1/8 In.		Plus 1/16 In. Minus 1/16 In.	
20 x 25 x 2	1000	19 5/8	24 9/16	1 13/16	3/4
20 x 20 x 2	800	19 5/8	19 5/8	1 13/16	3/4
16 x 25 x 2	800	15 13/16	24 13/16	1 13/16	3/4
16 x 20 x 2	640	15 13/16	19 5/8	1 13/16	3/4
15 x 20 x 2	600	14 15/16	19 15/16	1 13/16	3/4

W. B. CONNOR ENGINEERING CORP.

114 East 32nd Street
New York, N. Y.

DOREX Division

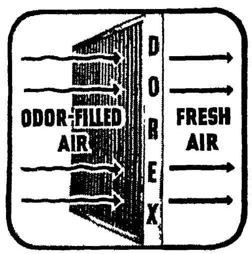
Representatives in All Principal Cities

Canadian Representative: Arthur S. Leitch Co., Ltd., Toronto, Ont.

Manufacturers of a Complete Line of Odor Removal Equipment

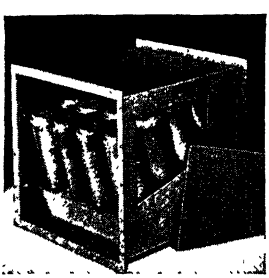
Wherever Odor or Air Contamination is an Annoyance, Nuisance or Hazard, a Dorex Solution is Available

Dorex is the only positive method of extracting gaseous impurities from air. It is the practical application of the fool-proof military gas mask to commercial air purification. Dorex is daily eliminating odors in stores, restaurants, theatres, hotels, schools, hospitals, laboratories, public buildings, plants, factories, etc.



In Air Conditioning, Dorex enables the reduction of outside air requirements to a minimum, making possible substantial savings in equipment costs and operating expense. Because Dorex imposes little resistance and can be inexpensively reactivated upon exhaustion, maintenance is low.

Type "H"—For Contaminated Air Intakes, Noxious Exhausts, Odorous Recirculation, Etc.

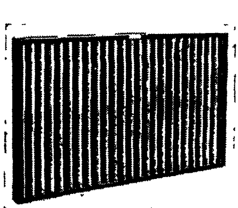


C.F.M.	No. of Canisters	Height, In.	Width, In.	Length, In.	Weight, Lbs.
1000	35	20	41	18	70
5000	175	40	100	18	
		or 60	50	22	350
10000	350	60	100	22	
		or 80	80	22	700
20000	700	80	160	22	
		or 100	82	44	1400

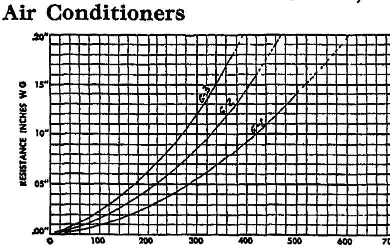
Note: Dimensions given only indicate volume of space required and are arbitrary. If width is increased, height or length may be reduced accordingly and vice versa.

Multiple, removable, perforated metal activated carbon-filled adsorption canisters—compactly arranged on manifold plates in ducts or casings, so that all air must pass uniformly through the granular carbon media. These units have a life of six months to over two years, depending on service. When exhausted, the used canisters are returnable for substantial credit against the purchase of new canisters.

Type "G"—For Insertion in Ducts, Attachment to Air Filters or Grilles, Incorporation in Unit Air Conditioners



Designed especially to remove accumulated odors from air recirculated in air conditioning systems, the Type "G" is extremely compact and is easily installed in existing as well as new systems. Available in sturdy, standard size panels listed, having one, two or three staggered rows of perforated metal carbon-filled, adsorption tubes, designated as G-1, G-2 and G-3 respectively.

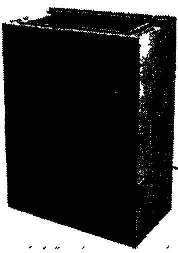


20 x 16½	20 x 20	24 x 24
25 x 16½	Nominal Dimensions, In.	25 x 20

All Dorex Adsorbers Employ Gas Mask Grade, Highest Activity, Specially Processed, Coconut Shell Carbon

Model	Capacity, C.F.M.	Current, Watts	Av. Cu. Ft., Room Volume Served
W-150-FS*	150	90	750
W-300-FS*	300	120	1500

**Note: When intended for Public Lavatory, substitute "FH" for "FS" When extremely quiet operation is required, substitute "B" for "FS."*



Type "W"—For Individual Rooms

Arranged for wall mounting. Completely self-contained, including Type "H" canisters, air filter, circulating fan and motor, all assembled in attractive enameled casing, equipped with chrome finish grilles. Also supplied with casters, floor or bed height.

Model	Capacity, C.F.M.	Motor Hp.	Av.Cu.Ft., Room Volume Served
CL-500-F*	500	1/12	2500
CL-1000-F*	1000	1/8	5000
CL-1500-F*	1500	1/4	7500

Type "CL"—For Larger Individual Rooms

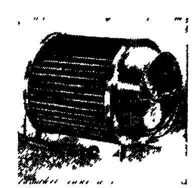
Arranged for ceiling mounting. Completely self-contained, including Type "H" canisters, air filter, circulating fan and motor. Three sizes. Ideal for larger offices, meeting and dining rooms, restaurants, lounges, etc. Height 20 in., length 4 ft. 10 in , and widths from 20 in. to 60 in. Not available for portable use.

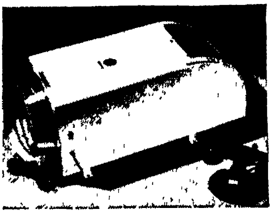
**Note: When extremely quiet operation is required, substitute "B" for "F."*

Type "SQ"

For intermittent light duty. Portable or stationary. Compact, cylindrical cage of perforated metal, carbon-filled, adsorption tubes, with built-in circulating fan and motor. Streamlined, chromium finish.

Unit	Av. Cu. Ft., Room Volume Served	Current, Watts	Size, In. Diameter	Size, In. Length	Weight Lbs.
SQ-8	1200	30	11	17	14
SQ-10	2000	40	13	17	20
SQ-12	3500	75	15	22	29
SO-14	7000	120	17½	30	48





Type "A-100-F"

For general utility duty. Delivers 100 cfm of pure, filtered, odor-free air. Contains 4 carbon filled canisters, dust filter, 40 watt motor and circulating fan—all encased in enameled cabinet. Has a host of practical uses—in homes, offices, doctor's suites, stores, walk-in refrigerators, locker plants, etc.

Among Thousands of Dorex Users

American Cyanamid Co., Anheuser-Busch, Inc., Bonwit, Teller & Co., Consolidated Edison Co. of New York, Inc., Coty, Inc., DuPont Film Mfg. Co., Huyler's Restaurants, Linde Air Products Co., Merck & Co., Inc., Metropolitan Life Ins. Co., The Procter & Gamble Co., Standard Oil Companies of N. Y. and N. J., Union Carbide and Carbon Corp., Western Union Telegraph Co., Westinghouse Electric & Mfg., F. W. Woolworth Co.

There are Dorex Units for compressed air lines and many industrial processes. Write for complete information.

Owens-Corning Fiberglas Corporation

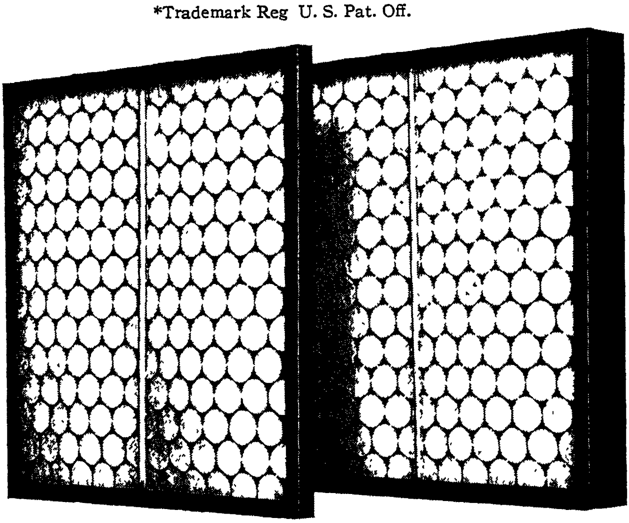
Toledo, Ohio

AIR FILTERS

FOR USE IN RESIDENTIAL, COMMERCIAL *and* INDUSTRIAL HEATING, VENTILATING *and* AIR-CONDITIONING SYSTEMS

FIBERGLAS* DUSTOP* AIR FILTERS

*Trademark Reg U. S. Pat. Off.



Dust-Stop No. 1 Filter, 1 in. thick, at left. No. 2 Filter, 2 in. thick, shown at right. No. 2 Filters are designed for application where their greater dust-holding capacity permits longer intervals between replacements.

The Fiberglas Dust-Stop Air Filter consists of a series of Fiberglas Mats, progressively packed—coarse glass fibers of lesser density at the intake and fine glass fibers of greater density at the discharge face—between stamped metal grilles bound with a fiberboard frame.

Mats are coated with a non-evaporating, adhesive having extraordinary wetting power, will retain viscosity under operating temperatures ranging from 15 F below to 300 F above zero, will not flow or charge the air with adhesive.

Engineered to Provide High Efficiency, Fiberglas Dust-Stop Air Filters also provide low cost of installation and maintenance.

Efficiency—97 per cent (Tested according to A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work).

Available in Two Standard Types—Fiberglas Dust-Stop No. 1 (1 in. thick) is designed for greatest operating economy in commercial and industrial applications. No. 2 (2 in. thick) is recommended for use in unsupervised installations. It permits longer intervals between replacements. Both may be used in domestic applications.

Engineering Service—Owens-Corning Fiberglas Corporation maintains offices in several metropolitan centers where representatives, qualified to assist in the planning of filter installations, are available for consultation.

Literature—Data sheets on all standard Fiberglas products and applications will be furnished to engineers and manufacturers on request.

Standard Sizes for Equipment*

Standard Sizes (Nominal)	Ratings Cfm	Ratings Fpm	Average Resistance Inches Water Gauge Clean
20″ x 25″ x 1″	1000	300	.065
20″ x 20″ x 1″	800	300	.065
16″ x 25″ x 1″	800	300	.065
16″ x 20″ x 1″	640	300	065
20″ x 25″ x 2″	1000	300	.125
20″ x 20″ x 2″	800	300	.125
16″ x 25″ x 2″	800	300	.125
16″ x 20″ x 2″	640	300	.125

*Other standard and any special sizes available.

Owens-Corning Fiberglas Corporation

Toledo, Ohio

AIR FILTER FRAMES FOR HEATING, VENTILATING and AIR CONDITIONING SYSTEMS

FIBERGLAS* DUSTOP* FILTER FRAMES

*Trademark Reg. U. S. Pat. Off.

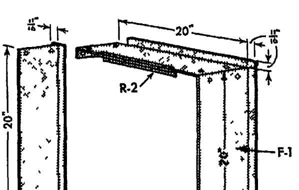
Fiberglas Dust-Stop "L" and "V" Filter Frame Assemblies are installed by engineers of commercial and industrial heating, ventilating and air conditioning. Frame members of heavy steel are assembled vertically in combinations to satisfy any CFM and space requirement.

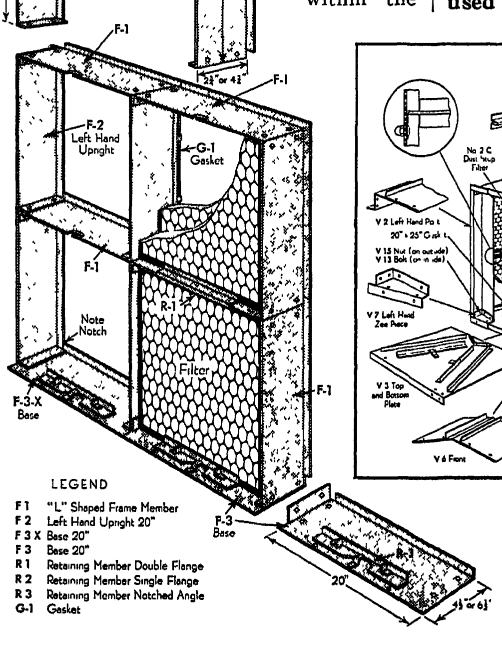
Both types of frames are designed for the convenient and correct handling of Dust-Stop filters. They meet all Fire Underwriters' and local Fire Ordinance requirements, as well as the requirements of Federal Specifications for filter frames.

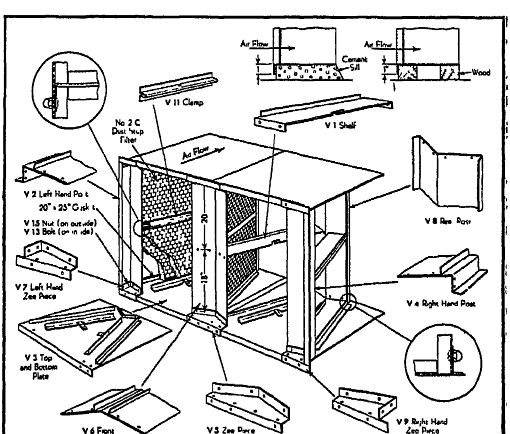
The choice between the "L" type and "V" type frames is determined wholly by the space available for the filter frames. The "L" type filter frame takes less depth within the duct or plenum chamber but requires a larger face area for the same CFM capacity. The "V" type frame requires a face area approximately the same as the cross-sectional area of a duct which will handle the volume of air for which the filters are rated.

Two Depths of "L" Frames—The "L" frame, two filters deep, is designed to hold two Dust-Stop No. 1 filters in each cell. The "L" frame, four filters deep, holds four Dust-Stop No. 1 filters in each cell. The frame that is four filters deep is identical in every way to the frame two filters deep except that the depth of all parts is 2 in. more. When specifying "L" type frames indicate two-filter or four-filter depth. "V" frame is available, four 1-inch filters deep per cell, only.

The "L" frame uses 20 x 20 in. filters only. The "V" frame uses 20 x 25 in. filters only. Filters are always used two or more in series in each cell.







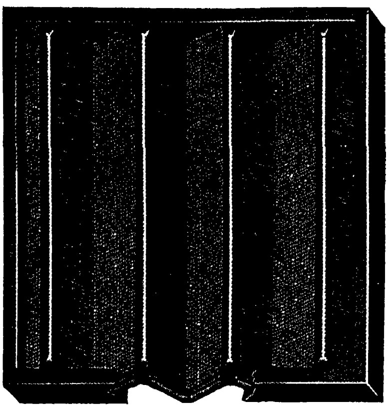
LEFT—Dust-Stop "L" Type Filter Frame.

ABOVE—Dust-Stop "V" Type Filter Frame

H. J. Somers, Incorporated

Factory and General Office

6063 Wabash Avenue **Detroit, Mich.**



All Welded Vee Type

Somers Washable Air Filter

Somers Hair Glass Filters provide everything required in an efficient air-cleaning system. Consider these features: High rating for dust, soot and bacteria separation. Require no adhesive, coating or impregnation. Indestructible in normal service. Minimum Low Pressure Drop. Odorless and non-absorptive. Fireproof; Washable; Do not rot nor disintegrate; Permanent.

Somers Hair Glass Filters consist of a hot galvanized frame holding galvanized wire cloth packed with hair-spun glass strands. The glass strands are flexible, do not break up and cannot be drawn into air stream.

Hair-Glass, being chemically inert, has no facility of absorption; it cannot rust and lasts indefinitely in service. Water either hot or cold may be used to clean it, without impairing its efficiency. These filters eliminate the necessity, the expense and the inconvenience of periodic replacement.

Somers Washable Air Filter—All Welded Vee Type—Stock Sizes (Partial List)

Frame Size Height and Length In.	Frame Depth In.	Filter Surface Sq In.	For Average Dry Filter Installations CFM	Wet Application where water sprays are applied against filter for humidifying CFM
15½ x 24½	3 5/16	1023	1023	511
15⅝ x 24⅝	3¼	1110	1110	555
16 x 21½	3 1/16	816	816	408
16 x 25	3⅛	1056	1056	528
16 x 25	3¼	1632	1632	816
16 x 25	3¼	1344	1344	672
16 x 25	3¼	1440	1440	720
16 x 25	3 5/16	864	864	432
16½ x 24½	3⅛	800	800	400
18 x 18	3 5/16	864	864	432
19 x 20	3 5/16	1482	1482	741
19¼ x 19⅝	3¼	1039	1039	519
19¼ x 20	3	1039	1039	519
19⅜ x 19½	3	936	936	468
19½ x 19½	3¼	1053	1053	526
19½ x 19½	2	480	480	240
19½ x 19½	3	936	936	468
19½ x 19½	3¼	1170	1170	585
20 x 25	3 5/16	1800	1800	900
20 x 30	3 5/16	1800	1800	900
20 x 20	3 1/16	1040	1040	520
20 x 30	3	1560	1560	780
20 x 20	3¼	1200	1200	600
20 x 20	2	480	480	240
20 x 20	3	840	840	420
20 x 20	3	960	960	480
20 x 20	3¼	1320	1320	660
20 x 25	3¼	1560	1560	780
20⅜ x 20¼	3	550	550	275

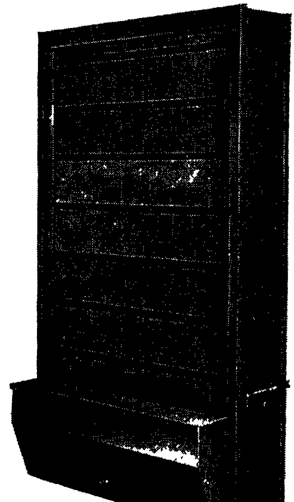
Other sizes from 9½ x 30 to and inclusive of 31 in. x 23¼ also available. Send for complete stock size list.
Frames zinc plated for 100 hour salt water spray test. Refill may be inserted if necessary.
Quotations and further engineering data, including master holding frame drawings will be sent on request.

Staynew Filter Corporation

Air Filters for Every Purpose

6 Leighton Ave. Rochester, N. Y.

PROTECTOMOTOR AIR FILTERS



Automatic Filter

AUTOMATIC FILTER

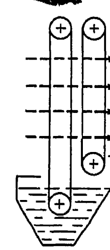
(For Efficiently Filtering Large Volumes of Air at Low Cost)

This latest model Automatic Filter removes dust from the air stream by the impingement principle. Two endless, oil-moistened Filter Curtains of special copper mesh provide four separate stages of filtration. No other filter has the *double Filter Curtain* feature—*double assurance of clean air delivery.*

Both Endless Filter Curtains move intermittently at predetermined intervals. One Curtain moves through an oil reservoir. This Curtain removes most dust particles from the air stream. The second Curtain, running dry except for traces of oil from the first Curtain, removes whatever finer dust particles may still be in the air stream.

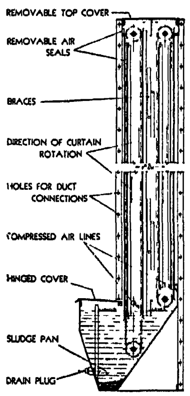
All excess oil (in which dust particles are trapped), is completely removed from both Curtains by a double series of low pressure compressed air jets. *The air jet cleaner feature is another exclusive Staynew development.*

Direction of Curtain travel is an important, exclusive feature. Both Endless Curtains travel counter-clockwise (air flow from the left). This means that air passes last through the cleaned side of each Curtain—that is, the final stage of filtration in each Curtain has been cleaned either by oil bath, air jets, or both.



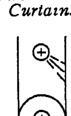
Two Endless Curtains

Specifications:



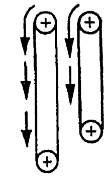
Section View of Automatic Filter

Two standard widths are built—2 ft 9 in., and 4 ft 3 in., in 41 heights from a 4 ft minimum to 14 ft. Combinations of standard sizes will fit almost any required capacity or installation space. Special sizes are built to order.



Air Jet Cleaners

Motive power for the Endless Curtains is supplied by a 1/6 hp motor, Telechron-controlled. Speed reduction is accomplished through a standard reducer. Curtain travel is from 7 to 30 seconds each ½ hour, depending on Curtain height. Each Curtain makes one complete revolution every 24 hours.



Curtains Travel Counter-clockwise—Air Flow is L to R in each Diagram

A ½ in. standard fitting is provided at the rear of each section for connection to user's source of air supply. Flow of air to Air Jet Cleaners occurs simultaneously with Curtain movement and is controlled by a Solenoid valve. A pressure reducing orifice lowers any conventional air line pressure to the required pressure of approx. 1 lb.

Staynew Filter Corporation

Air Filters for Every Purpose

6 Leighton Ave. **Rochester, N. Y.**

PROTECTOMOTOR AIR FILTERS

PROTECTOMOTOR DRY-TYPE FILTERS

(For removing foreign matter from the air with types for building ventilation, dust recovery, and all air-cleaning purposes.)

The Fin or V-type construction is used in all Protectomotor Dry Type Filters. This basic principle permits: (1) the largest possible active filter area in a minimum space and (2) air flow parallel to the filtering surface at low velocity.

Protectomotor Dry-type Filters require no coating of viscous liquid to catch dust—*odorless*, oil-free air is thus assured.

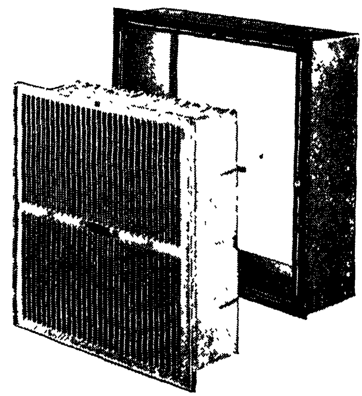
Authorities agree that the Dry-type Filter is most efficient in removing the smaller air-borne particles. Protectomotor Dry-type Filters actually prevent the passage of certain bacteria.

Easy to Clean—Long Lasting

An outstanding advantage of the Dry-type Filter is its ease of cleaning. This is accomplished with the panels in place by means of a vacuum cleaner or jet of compressed air. No cleaning or charging tanks are required.

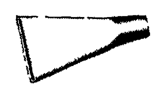
Due to exceptionally large filter areas and low air velocities, these filters operate for long periods without cleaning. Even if neglected, their effectiveness in the prevention of the passage of dust is not lessened—the only effect is increased pressure drop.

Staynew Panel and Wire-Klad Filters last several years at least without service other than cleaning—Multi-V-Type Filters slightly less.



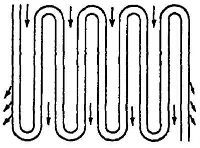
Outlet Side of Panel Unit

Cleaning Nozzle



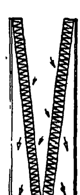
Panel Filters

Consist of Panel Insert and Frame. The Insert is composed of two rows of 30 fins each, 6 in. deep, formed of rust-resisting embossed wire mesh, covered by a single piece of Feltex Filtering Medium, a felt-like material specially made for this application. Fins are supported by steel retaining grates. Specifications below.



Cross Section Showing Panel Unit Construction

Cross Section of 2 Multi-V-Type Cells in V Formation →



Overall Dimensions (Depth less locking keys)	20 x 20 x 7 in.
Size of Insert	19½ x 19½ x 6⅝ in.
Capacity	up to 800 cfm
Area of Filtering Medium	42 sq ft
Linear velocity of air	19 fpm
Resistance of clean filter to air flow	0.2 in. wg
Total Weight	33 lb

Staynew Filter Corporation

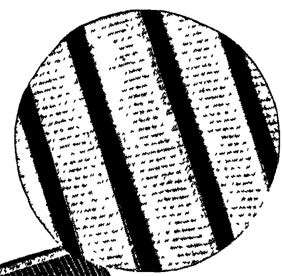
Air Filters for Every Purpose

6 Leighton Ave. **Rochester, N. Y.**

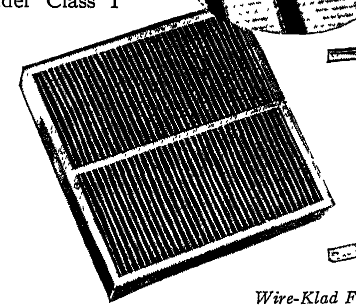
PROTECTOMOTOR AIR FILTERS

Wire-Klad Filters

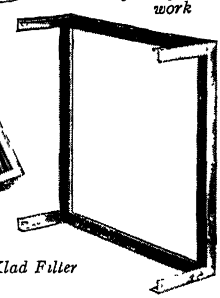
Unique methods of construction permit a high efficiency filter at low cost. Fins are reinforced on both sides with rust-resisting screen cloth, producing a rigid, long-wearing, flame-resistant filter that withstands repeated cleaning exceptionally well. Not only are standard units highly flame resistant, but when furnished with Bonded Glass medium, Wire-Klad Filters are *approved* by the National Board of Fire Underwriters under Class 1 rating.



Enlarged Section showing Wire-Klad Construction



Wire-Klad Filter



Light steel frame can be bolted or welded to others to form framework

The Wire-Kladding feature is not found in any other filter. It is an *exclusive* Staynew development.

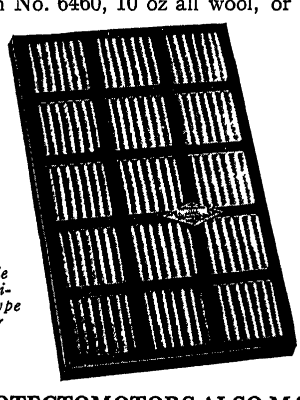
Replacement cost is low. Finned element *only* need be replaced, and is easily removed from its retaining case.

Specifications Given Below. Note Data Regarding various Filter media.

Overall Dimensions Inches	Filtering Area Square Feet	Overall Dimensions Inches	Filtering Area Square Feet
2 x 15 x 20	13.8	4 x 15 x 20	20.1
2 x 16 x 20	14.8	4 x 16 x 20	21.4
2 x 20 x 20	18.5	4 x 20 x 20	26.5
2 x 25 x 20	23.0	4 x 25 x 20	33.5

The 2 in. deep unit is supplied with our No. 6460 woven material or our B-G (bonded glass) media. The 4 in. deep unit with No. 6460, 10 oz all wool, or bonded glass media. Other filter media can be supplied when conditions warrant. Initial resistance depends on medium used. Performance Curves on application.

Multi-V-Type Filters



Single Multi-V-Type Filter

Filtering medium (closely pressed cotton fibres between two sheets of cotton gauze) is arranged in patented V-shaped pockets in a fibre-board and pressed metal frame. These patented cells can be quickly and inexpensively replaced when worn out. Their arrangement makes possible an active filtering surface of 27 times face area. This type filter permits a highly efficient filter installation at an extremely low cost. It is made in 1 and 2 in. depths and in a wide variety of heights and widths. (Protectovent Window Ventilator, which supplies clean, fresh air to home or office, employs Multi-V-Type inserts.) *Complete specifications mailed promptly on request.*

Write for Catalog Mentioning Special Interests

PROTECTOMOTORS ALSO MADE FOR INTERNAL COMBUSTION ENGINES, COMPRESSORS, TURBO-GENERATORS, AIR TRANSMISSION LINES, ETC.

Westinghouse Electric & Manufacturing Co.

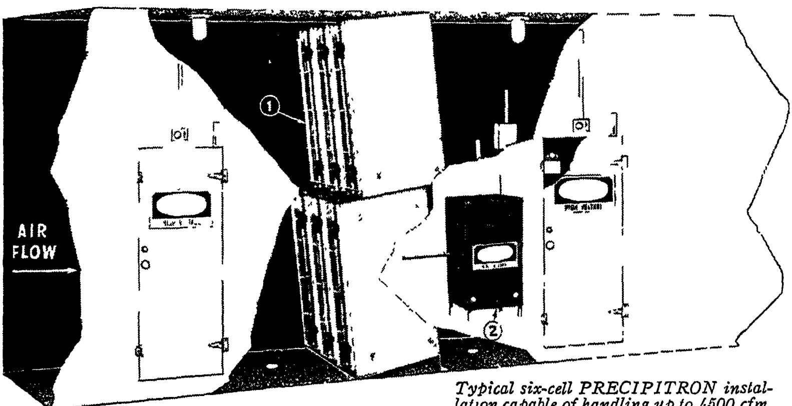
Edgewater Park Precipitron Department **Cleveland, Ohio**

THE PRECIPITRON*

First Commercially Practical Electrostatic Air Cleaner

The Westinghouse PRECIPITRON is the first commercially practical electrostatic method of removing dirt, dust and other air-borne impurities in ventilating and air conditioning systems. The PRECIPITRON—being more efficient than mechanical filters—removes microscopic foreign matter as small as 1/250,000 of an inch in diameter—even freeing the air of tobacco smoke.

*Trade-mark Registered in U.S.A.



Typical six-cell PRECIPITRON installation capable of handling up to 4500 cfm (1) cells; (2) Power Pack.

The PRECIPITRON provides a complete answer to mass air cleaning jobs in all Commercial, Industrial, and Public Buildings using forced ventilation or air conditioning duct systems.

Applications—Used in industry and production work. PRECIPITRON is serving many manufacturing processes in blackout, air-conditioned plants. Removal of smoke and welding fumes makes possible a considerable reduction in fresh air requirements with consequent savings in cooling and heating costs.

In steel mills and power plants PRECIPITRON is cleaning the ventilating air for rotating electrical machinery. Precision tools, dies and gauges are stored in spaces supplied with PRECIPITRON cleaned air to protect them from abrasive and corrosive dust or dirt. Optical instruments such as bomb sights, binoculars and telescopes are being assembled and maintained to a higher degree of precision with electrostatically cleaned air. PRECIPITRON cleaned air is being supplied to paint spray booths, air cooled radio transmitters, food and drug processing and packaging areas and for the processing and molding of plastic materials. Other fields in which the PRECIPITRON is a proved component of the ventilating or air conditioning system are textile mills, telephone exchanges, hospital operating rooms and commercial and public buildings.

Sizes—The PRECIPITRON is available, complete for installation, to accommodate from 300 cfm (for a single 18 in. cell) to any desired volume through multiple cell arrangements. Cells come in two sizes—18 in. x 8⅛ in. x 23⅝ in. and 36 in. x 8⅛ in. x 23⅝ in. For a 90 per cent efficiency, the 18 in. and 36 in. cells are rated at 300 and 600 cfm respectively. For 85 per cent efficiency, ratings are 375 and 750 cfm. Two sizes of Power Packs: Type S for installation up to 12 36-in. cells and Type L for 12 to 50 36-in. cells.

Advantages—More efficient than mechanical filters. Safe. Easily installed. Non-clogging, and non-varying resistance. Easily cleaned. Passed by Underwriters' Laboratories on standard flame tests for fire hazard on duct installation. Once each month accumulated dirt is washed away.

Information—Westinghouse will gladly provide complete information about the PRECIPITRON. Address your requests to Section G, Precipitron Department, Westinghouse Electric & Manufacturing Company, Edgewater Park, Cleveland, Ohio.

Oakite Products, Inc.

General Offices: 36 E. Thames St., **New York, N. Y.**

OAKITE

MATERIALS . . . METHODS . . . SERVICE

CLEANING

Established 1909

Representatives in all Principal Cities of the United States and Canada

Specialized OAKITE Materials for:

**Slime Control in
Re-circulating Systems.
Cleaning Air Filters,
Lube Oil Coolers,
Heat Exchange Equipment.**

**De-Scaling
Condensers, Compressors,
Jacket Water Coolers,
Diesel and Gas Engine
Cooling Systems.**

Controlling Slime Growths

Control of bacteria and slime growths in re-circulating water supplies is inexpensively established with Oakite Airefiner. A dry, non-volatile, white powder, completely soluble, it prevents formation of slime accumulations and their unpleasant odors. Economical to use . . . one lb to each 300 gallons of water usually recommended.

Prevents Equipment Corrosion

Oakite Airefiner prevents corrosion of eliminators, air washing chambers, spray heads, etc., because it maintains water at a point sufficiently alkaline to counteract the tendency of the water to become acidified. In addition, it gives wash waters greater wetting-out action, thus making dirt removal more complete. Water lines are also kept free of scale.

Free Booklet Gives Details

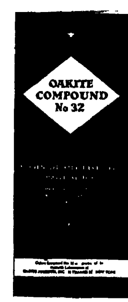
Cleaning Air Filters

For cleaning viscous type filters, hot or cold solutions of recommended Oakite material may be used. Short immersion of filter, followed by rinse, thoroughly removes dust, dirt, soot, lint and pollen without injuring filtering medium or frame metal. Steam cleaning methods also available. Full filtering capacity is economically restored.

FREE 16-page booklet gives details.

De-Scaling Equipment Safely, at Low Cost

When water scale and rust deposits form in jacket water coolers, ammonia condensers, air or gas compressors or other water-cooled equipment...heat transfer is reduced, operating efficiency impaired. These insulating deposits may be safely, effectively removed simply by soaking with or circulating recommended solution of Oakite Compound No. 32. Does not harm base metal. Method is economical, thorough.



Free Booklet

New Free Booklet Gives Details

New 20-page booklet gives successful formulas and directions for a wide range of other work such as safely de-scaling and cleaning Diesel and gasoline engine cooling systems, lube oil coolers and heat exchange equipment of all types. Write for YOUR copy today!

Nation-Wide Service

Because Oakite cleaning materials are backed by a binding GUARANTEE and supplied through a Nation-Wide organization of Service Representatives throughly experienced in their application, users are assured of obtaining the economies and advantages they provide to effectively promote maximum performance of air conditioning, mechanical refrigerating or other equipment.

American Moistening Company

ESTABLISHED 1888

ATLANTA, GA.
BOSTON, MASS.

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CHARLOTTE, N. C.
GREENVILLE, S. C.

AMCO

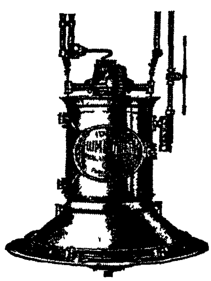
UNIT HUMIDIFYING AND AIR CONDITIONING EQUIPMENT

A few of many AMCO products with a Long Record of Dependable Performance

Sectional Humidifiers.
Amtex Humidifiers.
Hand Sprayers.
Mine Sprays.
Fabric and Paper Dampeners.

Mechanical Psychrometers.
Electro Psychrometers.
Sling Psychrometers.
Hygrometers.

The Amco line of devices for the supply, maintenance and control of humidity is complete in its ability to meet any presented problem of applied humidification. Used independently or as an adjunct to Central Station equipment, these devices automatically maintain any required humidity condition in a capable uniform performance.

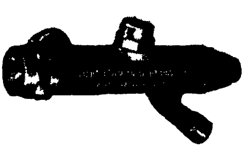


IDEAL HUMIDIFIERS—Senior Type

A high capacity unit for use where conditions require a great amount and good distribution of moisture. Motor driven fan gives wide distribution of atomized spray. Amco heads serve the triple purpose of humidifying, air washing and cooling.

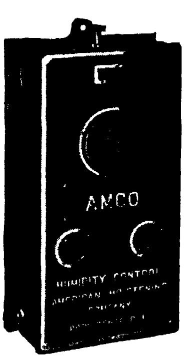
IDEAL HUMIDIFIERS—Junior Type

Similar in construction to Senior Type. Used where medium capacities are required.



AMCO ATOMIZER—No. 4

Quality and quantity of spray are maintained even under adverse conditions because this atomizer is automatically self-cleaning. When the compressed air supply is shut off, either manually or in response to a humidity control, *both* air and water nozzles are thoroughly cleaned.



AMCO HUMIDITY CONTROLS

Compressed Air Operated

An extremely accurate and active device operated by compressed air which assures a regulation of humidity within exceedingly close ranges.

AMCO HUMIDITY CONTROL

Electrically Operated

Similar in principle to the Compressed Air Type except that the hydroscopic element operates electrical contacts which control the units.

April Showers Company

4126 Eighth Street, N. W. **Washington, D. C.**

(Trade Mark Reg. U. S. Pat. Office)

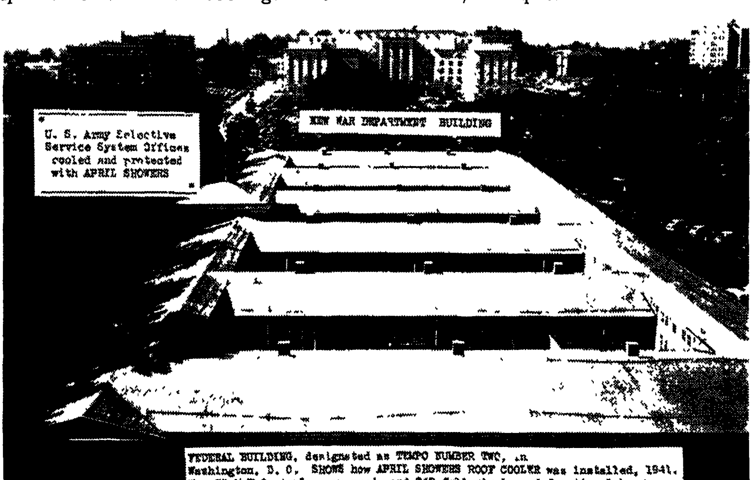
AUTOMATIC EVAPORATIVE ROOF COOLING

FIRE PREVENTION (from external sources) SYSTEM

AN EFFECTIVE WATER INSULATOR for all KINDS of ROOFS

Distributors and Dealers in Principal Cities

- **Solar radiation** is converted to cooling effect, reducing normal heat transmission 70 per cent upward. Entire roof surface temperature is normally held at wet bulb temperature when evaporative factors are favorable. Transmission of solar heat through glass skylights is reduced as much as 85 per cent.
- **Fire Prevention** from external sources is obtained through installation of manual emergency switch. Roof is quickly and completely sprinkled at will, putting out fire-brands, sparks, embers, and maintaining a water covered roof.
- **For Cooling Automatically** upper level, floors, lofts, rooms of buildings. For giant stores, theatres, amusement palaces, stadiums. APRIL SHOWERS is self operating by use of an electric thermal control placed upon the surface of the roof in the SUN. Evaporative cooling effect of liquid applied turns system off. Operating cycles repeat as roof temperature calls for cooling. Water may be used from city mains, wells, or waste water from condenser units.
- **High Temperature** in lofts, attics, space below roof, or roofing materials is abolished. Roofs of built-up composition, waterproofed with pitch, will remain firm and intact. LIFE of roof is lengthened; disintegration lessened; expansion and contraction which destroys is completely eliminated.
- **Literature Available.** Lists of installations in groups, residences, factories, theatres and amusement places, stores, apartments and hotels. Also circulars: General Description 1940, Residence circular 1941, Industrial Circular 1942. Literature giving engineering data, flow charts, testimonial letters, water consumption figures may be had without charge.
- **Water** consumption can be adjusted to approximately twenty gallons per day for 1,000 sq ft.



Hundreds of installations, from Boston to Los Angeles have been made. Write for information and address of nearest distributor.

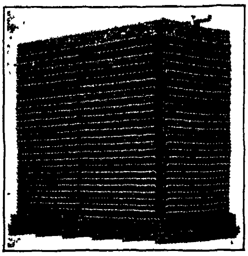
Inquiries will be promptly answered. Estimates free upon request.

The Marley Company

(Fairfax and Marley Roads,) **Kansas City, Kansas**

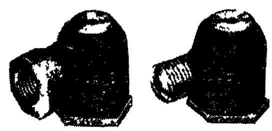
Branches or Agents in Principal Cities

Spray Nozzles and a Complete Line of Water Cooling Equipment

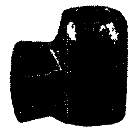


MARLEY NATURAL DRAFT TOWERS

Practically unlimited range of closely graduated sizes, entirely shop fabricated. Minimum initial, maintenance and operating costs. Many exclusive MARLEY advantages. **Bulletins 201 and 202.**



MARLEY Small 2-Piece Nozzles for Brine Spraying, Air Washing and Similar uses.



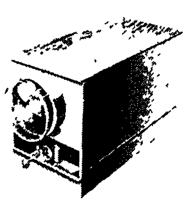
MARLEY Ice-Melting Nozzle for cooling systems using ice.

MARLEY Humidifying Nozzle adds moisture to air in open rooms or duct system.

Also Water Cooling Nozzles for Cooling Towers, Spray Ponds, etc.

MARLEY PATENTED NON-CLOG SPRAY NOZZLES

Made in scores of types and sizes. Practically any metal or alloy the purpose may demand. **Bulletins 101 and 102.**



SMALL INDUCED DRAFT TOWERS

Small, self-contained, steel units for 2 to 170 ton service, to go indoors or out. Smaller sizes (horizontal air flow) shipped all assembled, larger ones (vertical air flow) all shop fabricated for fast, easy assembly at location. **Bulletins 503 and 505.**

LARGE MARLEY MECHANICAL DRAFT TOWERS

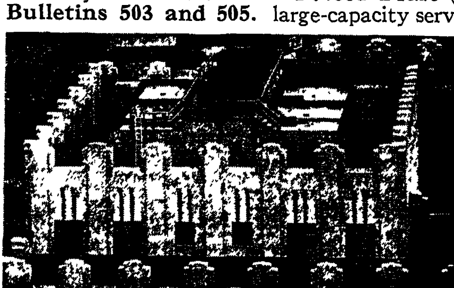
Both Forced and Induced Draft Towers, for heavy duty water cooling services of all kinds. Any capacity, with one fan or many, individually engineered to the exact requirements of each installation. MARLEY patents cover a variety of important features for extreme operating flexibility, high efficiency and economy.

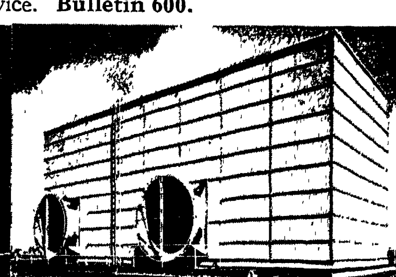
Redwood or Steel are standard materials, Transite and other materials on special order.

"Double-Flow" Induced Draft (below left) for largest capacities. **Bulletin 602.**

"Standard" Induced Draft (above) for usual large-capacity service. **Bulletin 601.**

Forced Draft (below right) for suitable applications in large-capacity service. **Bulletin 600.**



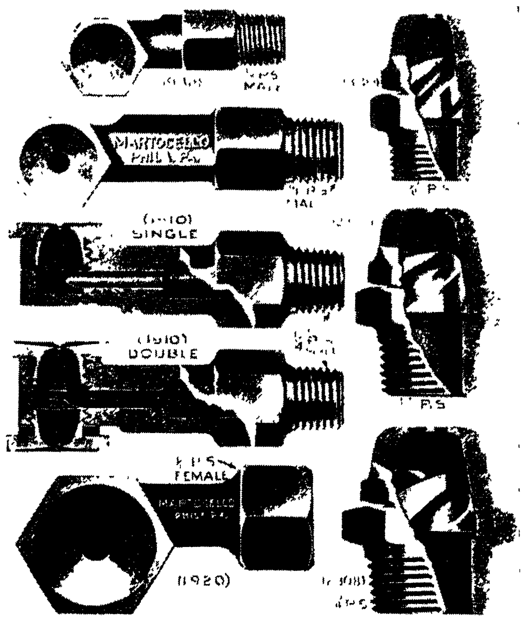


Also Many Other Types of Towers, Spray Ponds and Related Equipment

Jos. A. Martocello & Company

229-31 North 13th Street, **Philadelphia, Pa.**

ATOMIZING SPRAY NOZZLES



Types of Martocello Spray Nozzles

Martocello Spray Nozzles are broadly used for all types of installations. Manufactured with precision and of a design which has been thoroughly tested for results and durability, they are guaranteed to give you satisfaction.

Successful-Efficient results depend largely upon selecting the proper number and type of Nozzle suitable for your installation. Consult with us.

Martocello Spray Nozzles produce a uniform fine wide spray with minimum friction and at lowest pressure requirements.

We appreciate your inquiries and offer our cooperation in assisting you to select a proper Nozzle for best results.

For maximum efficiency we recommend Nozzle orifices as indicated in table below. Any reasonable range of capacity for various pressures can be provided.

Sizes and Capacities

Pipe Size Inches	Part No.	Diam. Orifice Inches	Capacity, Gallons per Minute							
			5 lb	10 lb	15 lb	20 lb	25 lb	30 lb	35 lb	40 lb
1/8	1930	7/64	.22	.29	.34	.39	.44	.49	.54	.59
1/4	1910	13/64	.54	.77	.96	1.13	1.29	1.44	1.58	1.71
1/4	1910 Double	5/32	.86	1.18	1.48	1.76	2.02	2.24	2.44	2.63
3/8	1920	17/64	1.48	1.96	2.38	2.75	3.08	3.36	3.60	3.82
3/8	2300	7/32	1.98	2.63	3.15	3.62	4.05	4.44	4.80	5.13
1/2	2304	5/16	2.66	3.77	4.71	5.52	6.24	6.87	7.47	8.04
3/4	2308	11/32	3.59	4.87	5.92	6.83	7.62	8.33	8.98	9.60

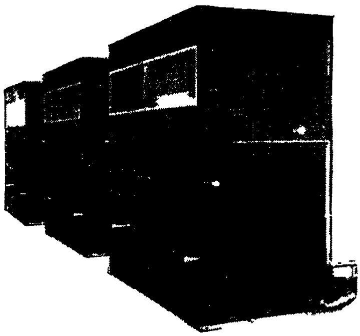
Nozzles illustrated above are made in Brass Forging and machined brass bar stock. Cast Red Brass Nozzles in 1 in. to 2 in. pipe sizes also available. All sizes carried in stock for prompt shipment.

Satisfaction Guaranteed

Acme Industries

Jackson **Michigan**

Offices in 30 Principal Cities



EVAPORATIVE CONDENSERS

Combine desirable features of cooling tower or spray pond and the water-cooled condenser. Heavy galvanized iron casing with bitumastic rust resisting paint coating inside. Rests on heavy sheet steel base. 11 Models—capacities up to 100 tons.

Ask for Catalog No 27

FLOODED SHELL AND TUBE COOLER

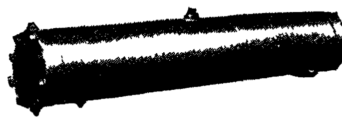
Constructed so that the water passes through the tubes surrounded by the refrigerant contained in the shell. Plain or insulated types. Cleanable tubes and heads. Designed for cooling drinking water, and processing or ingredient waters.

Ask for Catalog No. 29

DRY-EX WATER CHILLERS

Water chillers that are especially designed for recirculating systems—air conditioning of indirect type, processing, etc. Refrigerant in tubes—no bends; controlled water velocity; small refrigerant charge.

Ask for Catalog No. 30

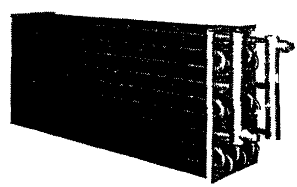


AMMONIA CONDENSERS
FREON CONDENSERS

For use with Ammonia, Freon, Sulphur Dioxide or Methyl Chloride. Tailor-made to order to meet varying water temperatures available and condensing temperature desired. Sizes range from fractional tons up.

For Ammonia—Ask for Catalog No. 21

For Freon—Ask for Catalogs Nos. 23 and 24



AIR CONDITIONING COILS

Air Conditioning Coils. Direct-expansion and water-cooled types. Aluminum plate fins on copper tubing.

Ask for Catalog No 34

ACME HEAT INTERCHANGERS

For use with Freon, Methyl Chloride and other refrigerants. Sizes suitable for use with varying imposed loads, refrigerant liquids, and gas temperatures.

Ask for Catalog No. 12

ACME ALSO MANUFACTURES: Oil Separators, Pipe Coils, Liquid Receivers, Accumulators, Specialties. Ask for Catalogs.

You don't have to choose between quality and price—BUY ACME

AEROFIN CORPORATION

410 So. Geddes Street

Syracuse, N. Y.

AEROFIN

Standardized Light-weight Heat Exchange Surface

Branch Offices

CLEVELAND, CHICAGO, NEW YORK, PHILADELPHIA, DETROIT, DALLAS, TORONTO

Aerofin is the modern Standardized Light-Weight Encased Fan System Heating and Cooling Surface originated by *Fan Engineers* to meet the present and future requirements of this highly specialized field. All Standard AEROFIN Units are furnished as completely encased Units, ready for pipe and duct connections. The patented casings are built of pressed steel and are exceptionally strong and rigid, protecting the Unit from all the strains of pipe connections and expansion or contraction in service. The casings are flanged on both faces, top and bottom, and template punched for bolting together adjacent Units, or for duct connection.

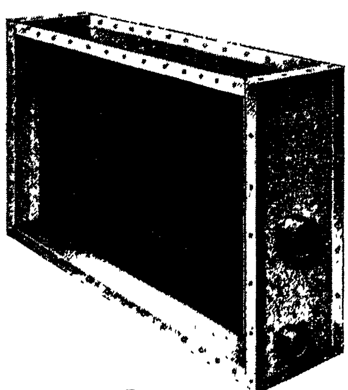


Fig 1

Aerofin Non-freeze heater (Fig. 1) is non-freeze, non-stratifying spiral fin coil built into casing for air conditioning units or for installing in ducts. May be installed horizontally or vertically. Used on any two-pipe steam system for preheating or reheating. Modulating control on preheaters.

Available in 13 lengths and 3 widths, from net face area of 2.76 sq ft to 26.28 sq ft.

Tubing 1 in. O.D. Innertube ⅝ in. O.D.

Headers—Cast Brass.

Fins—spiral turned copper.

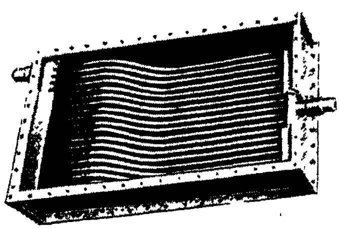


Fig. 2

Flexitube Aerofin (Fig. 2) is distinguished from all other developments by its off-set tubes, so arranged as to absorb all expansion and contraction strains.

Headers—Cast bronze or aluminum.

Tubing—⅝ in. O.D. copper, admiralty or aluminum.

Joints—Where admiralty or copper tubes are used together with bronze headers tubes are brazed to headers using Mueller patented joint. Where both aluminum tubes and headers are used tubing is welded to headers.

Casings—Copper, aluminum or galvanized iron.

Design—Constructed with headers on opposite ends making possible installation of units with tubes horizontal or vertical.

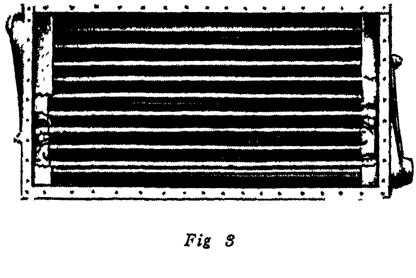


Fig 3

Universal Aerofin (Fig. 3) is distinguished by its "S" bend construction of

tubing, units designed with steel headers on opposite ends, the ends of the "S" bends being connected thereto by compression nuts, the bends taking care of the expansion and contraction of the tubing.

Recommended where close control is desired.

Headers—Pressed steel.

Tubing—1 in. O.D. Copper, admiralty or aluminum.

Casings—Copper, aluminum or galvanized iron.

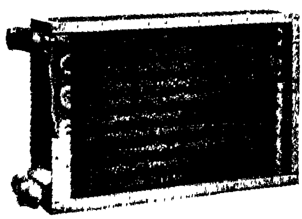


Fig. 4

High Pressure Aerofin (Fig. 4) is of continuous tube design, being recommended where extremely high pressures of steam are used.

Headers—Pressed steel.

Tubing—1 in. O.D. Copper, aluminum or admiralty.

Casings—Copper, aluminum or galvanized iron.

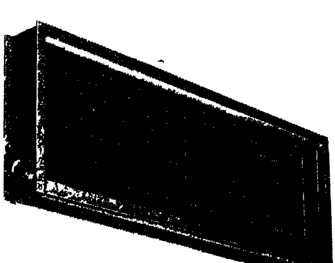


Fig. 5

Booster Aerofin—straight tube type, single pass construction for pressures from 1 to 200 lb gauge.

Headers—cast bronze.

Tubing—⅝ in. O.D. copper.

Casings—copper, aluminum or galvanized iron. Recommended where small coils are needed or to raise the air temperatures in branch ducts.

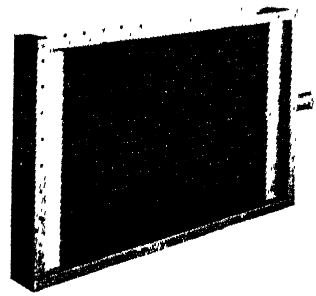


Fig. 6

Narrow Width Aerofin: (Fig. 6) recommended for water cooling or for flooded Freon systems. Made in straight tubes only with headers on opposite ends, joints between headers and tubing being brazed. Construction similar to Flexitube Aerofin.

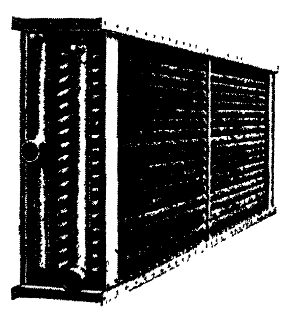


Fig 7

Aerofin Continuous Tube Water Coils (Fig. 7) are designed for air cooling by circulating cold water through the Aerofin and air over extended fin surface. Made for either horizontal or vertical air flow.

Tubes and fins are copper, completely tinned with permanent metallic bond

between fin and tubes. Headers are made of one-piece cast bronze and casings of heavy galvanized iron or copper.

Units tested to 1000 lb hydrostatic pressure.

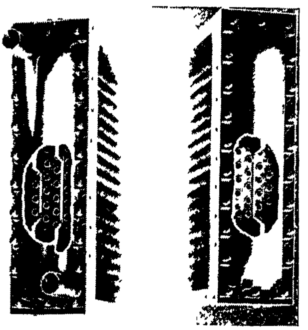


Fig 8

Aerofin Cleanable Tube Units (Fig. 8) for cooling only and all made with headers removable to permit cleaning out tubes. Recommended for use where sediment or scale forming chemicals are present in the cooling water.

Headers—Cast iron.

Tubing—Copper or admiralty.

Casings—Copper or galvanized iron.

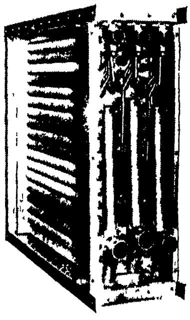


Fig. 9
End plate removed showing distributing and suction headers.

Aerofin Direct Expansion Units: (Fig. 9) Row Control Type—Recommended for use where cutting on or off rows of tubes in direction of air flow is desired. Suitable for use with Freon or Methyl-Chloride.

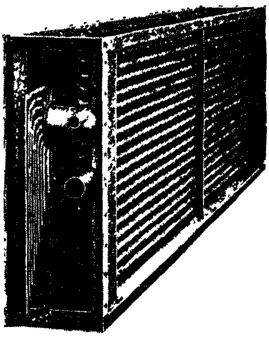


Fig. 10

Aerofin Direct Expansion Units: (Fig. 10) Centrifugal Header Type—Recommended where control of rows in direction of air flow is not required.

Advantages: Weighs but 9 to 16 per cent of same equivalent cast iron surface and occupies one-third of the space. Eliminates expensive foundations and building re-inforcement. Can be suspended from roof beams or trusses if necessary.

AEROFIN Sizes

Flexitube: 13 standard lengths, three widths, one and two rows deep.

Narrow: same as Flexitube.

Universal: 17 standard lengths, two widths, one and two rows deep.

Continuous Tube: 13 standard lengths, three widths, 2-3-4-5 and 6 rows deep.

Cleanable Tube: 17 standard lengths, one width, 2 and 4 rows deep.

Direct Expansion: Row Control—11 standard lengths, 3 widths, 1-2-3 rows deep. Face Control—11 standard lengths, 3 widths, 2-3-4-5-6 rows deep. Centrifugal Header—11 standard lengths, three widths, 2-3-4-5-6 rows deep.

Steel Supporting Legs: 18 in. and 24 in. high. Punched same bolt hole centers as standard casings. Quickly attached. No other foundation required.

Sale: AEROFIN is sold only by manufacturers of nationally advertised Fan System Apparatus. List upon request.

Write Syracuse for Heating Bulletin G-32; Direct Expansion Bulletin DE-34 on refrigeration type units; Continuous Tube Bulletin C. T. 34 for Water Cooling Coils; or phamplet on Cleanable Type AEROFIN for cooling.

The G & O Manufacturing Company

138 Winchester Avenue **New Haven, Connecticut**

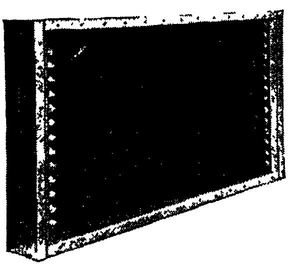
G&O

SQUARE FIN TUBING

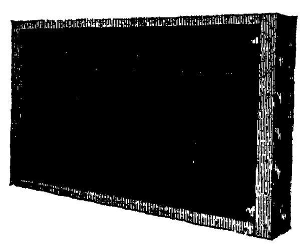
STRAIGHT LENGTHS—U-BENDS—CONTINUOUS COILS

RADIATING ELEMENTS FOR ALL HEAT TRANSFER PURPOSES

G&O Finned Radiation Coils for industrial applications are available in a wide range of sizes.



Universal U-102



Standard No 10

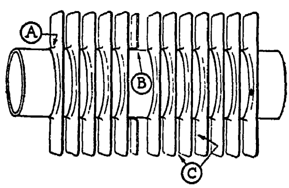
Send for Catalog and Price List

THE use of INDIVIDUAL fins results in high efficiency in heat transfer from primary tube surface to secondary fin surface.

Fins of any size or shape may be obtained giving any desired proportion of primary and secondary surface.

A square fin has about 30 per cent greater surface than a round fin of a diameter equal to one side of the square.

Individual fins permit of any fin spacing; also, of using fins in groups at intervals along tubes.



A—Generous Fin Collar provides large contact area between Tube and Fin.

B—Tube expanded against Fin Collar; insures mechanically tight joint, made permanent by bond of high temperature alloy—complete thermal contact.

C—Free air-flow passages; non-clogging.

STANDARD SIZES

O.D. of Tube	Fin Size	Fin Spacing per Inch	Surface per Linear Foot
3/8″	7/8″ sq.	6	0.80 sq. ft.
3/8″	7/8″ r'd.	6	0.60 sq. ft.
5/8″	1 1/8″ r'd.	6	0.87 sq. ft.
3/4″	1 1/2″ r'd.	6	1.55 sq. ft
3/4″	1 5/8″ sq.	6	2.40 sq. ft.
1″	2 1/8″ sq.	6	4.00 sq. ft.
1 3/8″	2 3/8″ r'd.	4	2.33 sq. ft.

Baker Ice Machine Co., Inc.

Omaha, Nebr.

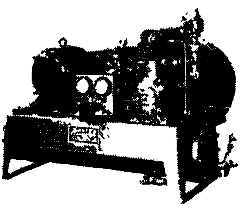
MANUFACTURERS OF INDUSTRIAL AND COMMERCIAL REFRIGERATION AND AIR CONDITIONING

Sales and Service in Principal Cities **Cable Address: BAKERICE**

AUTHORITY ON MECHANICAL COOLING FOR 35 YEARS

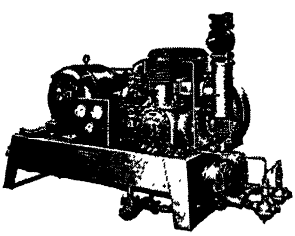
Precision manufactured and designed for maximum service and economy, Baker equipment is world-famous for its high quality and dependable performance. Important compressor features include: full force feed lubrication, honed cylinders, double trunk type pistons, Timken roller bearings. Write for specifications and descriptive literature.

BAKER "FREON-12" AIR CONDITIONING UNITS



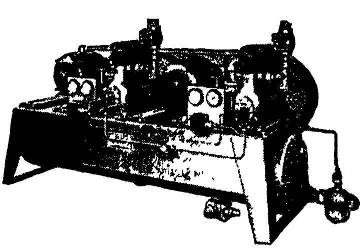
Small Condensing Units

Complete line of self-contained, automatic units. From ¼ hp to 15 hp capacity. 2-and 4-cylinder types. Air- or water-cooled.



Large Condensing Units

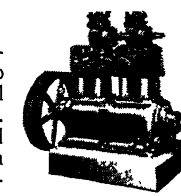
Available in 6 models, 20 to 60 hp inclusive, 4-cylinder, self-contained, automatic units. Shell and tube condenser-receiver. Pressure lubrication from gear type pump.



Dual Condensing Units

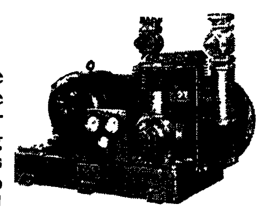
Designed especially for variable load requirements. Dual 4-cylinder type water-cooled unit. Automatic capacity control. Shell and tube type condenser.

Compressors

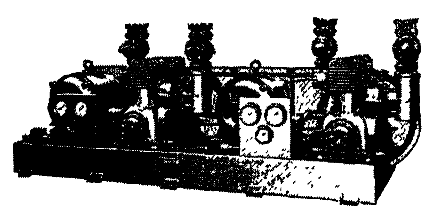


4-cylinder type, available in sizes from 10 hp to 60 hp. Semi-steel cylinders and pistons. Counter-balanced crankshaft, precision ground. Nickelite connecting rod bearings.

Compressor Units



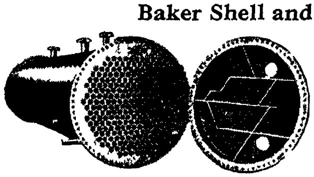
Arranged for use with evaporative type condenser or water cooling tower. Sizes range from 2 to 60 hp. 2- and 4-cylinder types. Automatic controls.



Dual Compressor Units for Capacity Control

19 different models, 10 hp to 120 hp inclusive. Two separate 4-cylinder compressors, automatic controls, independent motors and pressure lubrication. For use with separately mounted shell and tube or evaporative condenser.

Baker Shell and Tube Condensers and Liquid Coolers



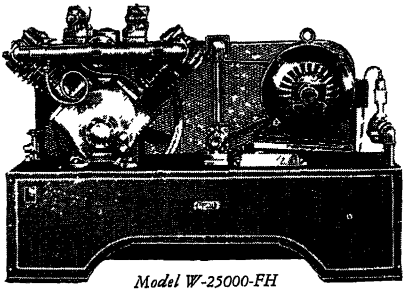
(1 to 250 tons capacity)

Horizontal multipass or vertical shell and tube construction. Complete range of sizes up to 2500 sq ft of cooling surface in single shells. Code welded, seamless steel or hard copper tubes. Easily cleaned.

BRUNNER MANUFACTURING COMPANY

UTICA, N. Y., U. S. A.

BRUNNER



Model W-25000-FH

COMMERCIAL REFRIGERATION

The Brunner Line of Refrigeration Equipment includes Air Conditioning models up to and including 25 hp for all types of high temperature applications within their capacity, using either "Freon-12" or Methyl Chloride as refrigerant.

BRUNNER DEPENDABILITY is based on time-proven features of design and manufacture . . . all parts are precision machined with extremely close tolerances . . . bronze bearings throughout . . . extra large fin surface on cylinders and heads . . . bellows seal . . . silent eccentric drive (except on 20 hp and 25 hp models, which employ crankshaft) . . . suction and discharge valves in "all-in-one" plate assembly . . . heavy-duty motor with high starting torque . . . adjustable motor base . . . multiple V-belt drive. Throughout, Brunner Refrigeration Units are geared to the demands of *heavy-duty* service.

SPECIFICATIONS					CAPACITIES Air Conditioning Units Based on 75° F Water Temperature "Freon-12" Refrigerant	DIMENSIONS
Model No.	H. P.	Cyls.	Bore and Stroke	R.P.M.	B.T.U. per Hr. 40° Evap. Temp.	L. W. H.
W 300-FH	3	4	3¼ x 2¼	260	38547	50″ 24″ 28¾″
W 500-FH	5	4	3¼ x 2¼	420	62270	50″ 24″ 28¾″
W 750-FH	7½	4	4¼ x 3	260	91526	71″ 29½″ 38½″
W 1000-FH	10	4	4¼ x 3	350	123211	71″ 29½″ 38½″
W 1500-FH	15	4	4¼ x 3	525	184815	71″ 29½″ 38½″
W20000-FH	20	4	4¼ x 5	435	255046	73″ 33¼″ 48¾″
W25000-FH	25	4	4¼ x 5	540	316652	73″ 33¼″ 48¾″

Additional air and water cooled models from ¼ h. p. for commercial and industrial applications.

The Brunner Field Sales Organization is available in all parts of the country, backed by outstanding achievements in engineering, and adoption of modern methods and design of air conditioning equipment.

Installation of Brunner refrigerating units is insurance of the finest quality materials and workmanship—plus the highest efficiency possible in modern design and manufacture.

FREE . . . COMPLETE ILLUSTRATED CATALOG

with large section devoted to ways of selecting the proper units for any application.

Curtis Refrigerating Machine Division

of Curtis Manufacturing Company

1959 Kienlen Ave., **St. Louis, Mo., U. S. A.**

ESTABLISHED 1854

93 Condensing Units from 1/6 to 50 hp

CURTIS REFRIGERATION AIR CONDITIONING AND COMMERCIAL

Unit Coolers and Evaporator Coils

PRODUCTS: Refrigerating Machinery; Forced Draft Cooling Units; Cooling Coils, Condensers, Shell and Tube Coolers, Valves, Fittings and Accessories, Complete Refrigerating Equipment for Dairies, Creameries, Ice Cream Cabinets, Ice Cream Making Plants, Cold Storage Locker Systems, Walk-in Coolers, Drinking Water Systems, Commercial and Low Temperature Cooling, Processing and Air Conditioning Installation, Packaged and Remote Types.

Commercial Refrigeration

45 air cooled condensing units from ⅙ to 5 hp, inclusive, and 48 water cooled units from ⅛ to 50 hp, inclusive. All models available for either Freon (F-12) or Methyl Chloride. Mechanical advantages include Timken Bearings, Centro-Ring Positive Pressure lubrication.

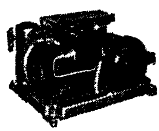
Special models are available for ice cream, frozen food cabinets and for the dairy industry.

Saturated Air Condenser

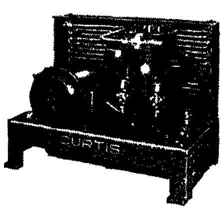
For condensing refrigerant vapors economically and efficiently. Saves approximately 95 per cent water cost. Used for air conditioning or commercial refrigeration installations up to 5 ton Capacity.

Air Conditioning

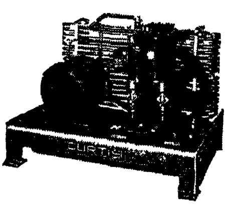
For stores, offices and all types of commercial establishments Curtis offers complete packaged, refrigerated air conditioning units, requiring only water and electrical connections to install. Cools, dehumidifies, circulates and filters the air. Eliminates costly installation expense. Adaptable for heating.



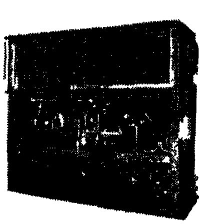
1/6 to ½ hp Self-Contained Condensing Unit.



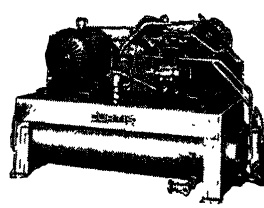
1½ hp Air Cooled Condensing Unit. Other sizes from ¼ to 5 hp.



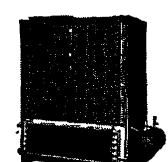
5 hp Water Cooled (Counterflow) Condensing Unit. Other sizes from ⅛ to 5 hp.

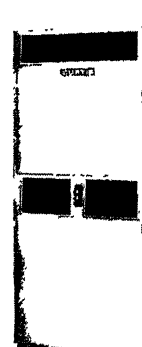


7½-10-15 ton Remote or Central Type Air Conditioner.



15 hp Cleanable Shell and Tube Condensing Unit. Other sizes from 3 to 30 hp.





3 and 5 ton Packaged Type Air Conditioner

ALBANY
ATLANTA
BALTIMORE
BOSTON
BUFFALO
CHARLOTTE
CHICAGO
CINCINNATI
DALLAS
DETROIT

Frick Company

(Incorporated)

Air Conditioning, Refrigerating and Ice-Making Equipment

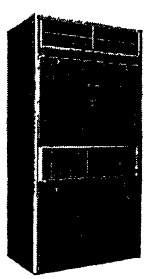
Waynesboro, Penna.

KANSAS CITY
LOS ANGELES
MEMPHIS
NEW ORLEANS
NEW YORK
OKLAHOMA CITY
PALATKA
PHILADELPHIA
PITTSBURGH
ST. LOUIS
SEATTLE

Distributors in 150 Principal Cities

AIR CONDITIONING

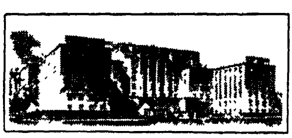
We furnish complete air conditioning systems as well as refrigerating machinery for use with equipment supplied by others. More than a thousand installations attest the value of the various Frick systems of air conditioning, some of which are patented, and of those made under the patents of the Auditorium Conditioning Corp. Ask for Bulletin 505, describing the five principal kinds of systems; also Bulletin 504, illustrating typical jobs. Estimates cheerfully furnished by Frick Branch Offices and Distributors throughout the world.



Frick Unit Air Conditioners, built in two sizes, are part of the complete line of Frick Air Conditioning Equipment

FRICK FREON-12 REFRIGERATION

Includes a complete line of enclosed type Freon-12 compressors. Large capacity, ample gas passages, pressure lubrication from internal pump, patented FLEXO-SEAL at shaft. Coils, coolers, condensers and controls for Freon-12 systems. Bulletin 508.



Frick Freon-12 Compressors of 750 tons capacity air condition the new War Dept. Building in Washington.

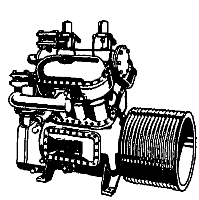
AMMONIA REFRIGERATION

Machines in all capacities from ½ ton up. Combined units and vertical enclosed type compressors; complete high and low sides. Widely used for air conditioning, with material savings in power. The Philcade and Philtower buildings in Tulsa, Okla., using 1000 tons of refrigeration, are typical of the many air conditioned with Frick ammonia systems—with entire satisfaction. Bulletins No. 104 to 700.

The Fondren Library, at Dallas, is air conditioned with Frick Ammonia Refrigeration.

LOW PRESSURE REFRIGERATION

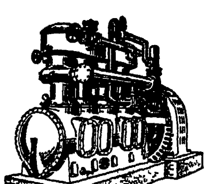
Commercial units in more than 50 sizes and types, with motors of ¼ to 60 hp. Charged with Freon-12 or methyl chloride. Air and water cooled condensers. Finned coils, fan and blower units, air conditioners. Bulletins 97 to 100.



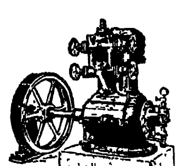
"Eclipse" Compressor: 3, 4, or 6 Cylinders.

SERVICES

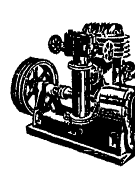
Offered to the industry include surveys, recommendations, literature, estimates, sales, manufacturing, installation, test and maintenance.



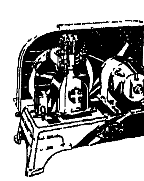
Large 4-Cylinder Compressor



Enclosed Type Ammonia Compressor



Frick Enclosed Freon-12 Compressor



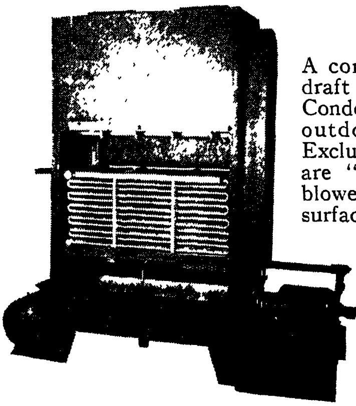
Low Pressure Refrigerating Units

Marlo Coil Co.

6135 Manchester Ave., **St. Louis, Mo.**

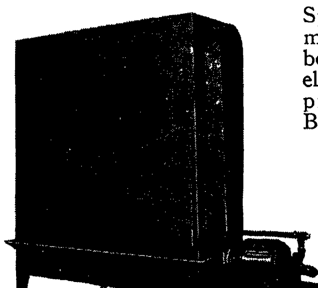
Manufacturers of Heat Transfer Equipment

Brine Spray Units—Unit Coolers—Evaporative Condensers—Low Temperature Units—Air Conditioning Units—Heating and Cooling Coils.



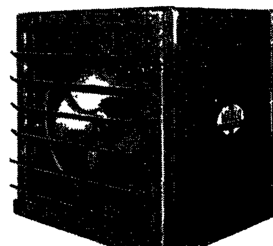
Evaporative Condenser

A combination forced-draft Cooling Tower and Condenser for indoor or outdoor installations. Exclusive Marlo features are "Unidrive". pump-blower motor; all prime surface coils; internal surface covered with corrosion resistant mastic; frame electric welded and galvanized after fabrication. See Bulletin No. 404.



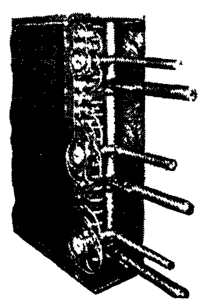
Brine Spray Units

Specially designed to maintain temperature below freezing, and yet eliminate all defrosting problems. Write for Bulletin No. 403.



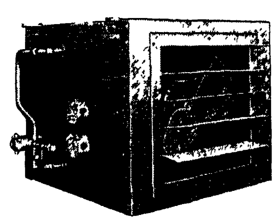
Unit Coolers

In this new model, air is pulled instead of forced through coils, thus utilizing complete coil surface and obtaining greater efficiency. Available in eight sizes, for all common refrigerants. Request Bulletin No. 402.



Air Conditioning Coils—Blast Coils

Durably built; conservatively rated; available in materials suitable for any cooling or heating medium. All coils thoroughly dehydrated and tested at 1,000-pound pressure under water. Ask for Bulletin No. 396.



Low Temperature Unit

Designed for sub-zero temperature application. Equipped with the original Marlo electric-heating element for manual or automatic defrosting. Available for any refrigerant. Full details in Bulletin No. 407.

Air Conditioning Units

Air Conditioning Units in either ceiling suspended or floor type. Capacities from 900 cu ft to 12,000 cu ft. Sturdily built on angle welded iron frames of sectional design for easy installation. Bulletin No. 409 gives complete details.

The Vilter Manufacturing Company

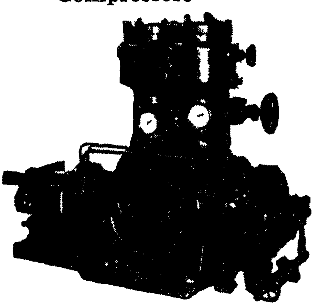
Since 1867

Milwaukee, Wisconsin

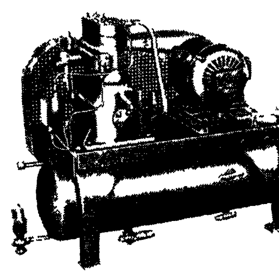
AIR CONDITIONING EQUIPMENT FOR INDUSTRIAL OR COMFORT COOLING

COMPRESSORS OF MODERN DESIGN

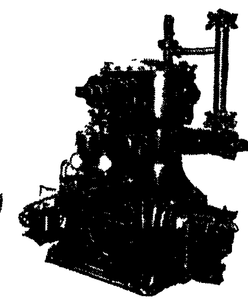
Ammonia Compressors



Freon or Methyl Chloride Condensing Unit



Freon Compressors for Large Installations



Ammonia Compressors—The result of over seventy years of research development and experience gained through thousands of installations of all types, in all industries. Famous for high tonnage capacity at low hp and low operating costs. Built in a wide range of capacities from 2 to 100 tons standard A.S.R.E. rating in Vertical Types; up to 750 tons in Horizontal Type.

Freon Compressors—Embody many outstanding new features that prevent leakage and minimize friction—resulting in extremely low relative hp per ton. Made in capacities up to 150 tons. Capacitrols available at slight additional cost provide flexibility of operation.

Freon Condensing and Methyl Chloride Units—Self-contained units made in sizes from ¼ hp to 30 tons capacity. Embody latest engineering features.

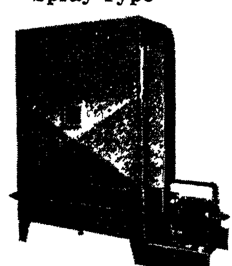
Unit Air Coolers—Available in a wide range of sizes and types for any air conditioning requirement—product coolers, dry coil coolers, spray type coolers, low temperature electric defrosting coolers, and floor or ceiling central system air conditioners.

Water Coolers and Condensers—A complete line of shell and tube water coolers, brine coolers and condensers for Freon or ammonia.

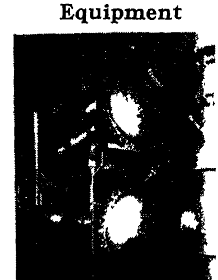
UNIT AIR CONDITIONERS

Dry Coil Type

Spray Type



Shell and Tube Equipment



Vilter also builds a complete line of air conditioning coils, evaporative condensers and air washers—and special units for central station comfort cooling systems.

Worthington Pump and Machinery Corporation

WORTHINGTON — Carbondale Division — CARBONDALE

General Offices: HARRISON, NEW JERSEY

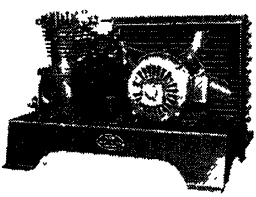
ALBANY	CHICAGO	EL PASO	NEW HAVEN	PROVIDENCE	SEATTLE
ATLANTA	CINCINNATI	FORT WORTH	NEW ORLEANS	ROCHESTER, N.Y.	SPRINGFIELD, MASS.
BALTIMORE	CLEVELAND	GALVESTON	NEW YORK	ST. LOUIS	SYRACUSE
BIRMINGHAM	DALLAS	HOUSTON	PHILADELPHIA	ST. PAUL	TULSA
BOSTON	DENVER	KANSAS CITY	PITTSBURGH	SAN FRANCISCO	WASHINGTON, D.C.
BUFFALO	DETROIT	LOS ANGELES	PORTLAND, ORE.	SALT LAKE CITY	WILMINGTON, DEL.

Representatives in Principal Cities of Foreign Countries CA2-1

REFRIGERATION SYSTEMS FOR AIR CONDITIONING IN COMFORT COOLING OR INDUSTRIAL PROCESS

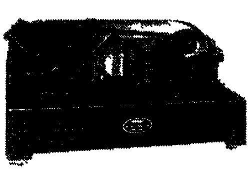
Complete refrigerating systems for use with Freon-11, Freon-12, Methyl Chloride, Ammonia, or Carbon Dioxide, either direct-expansion or water cooling applications. A complete line of refrigeration compressors, permitting impartial recommendations. A nation-wide organization of Distributors in major cities to provide sales and engineering service and plan complete air conditioning systems of the central or unit type. Architects, Engineers, and Contractors are invited to consult with us. Write to Harrison, N. J., or any branch office, for bulletins on these products.

Small Self-contained Units



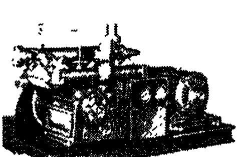
Freon-12 or methyl chloride condensing units; motors ¼ to 2 hp. with air or water-cooled condensers. Used in small air conditioning systems, and in commercial refrigeration. Capacities up to 2 tons.

Medium Self-contained Units



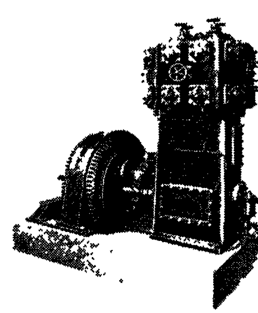
Freon-12 or methyl chloride compressor units for use with "shower" condensers or water-cooled condensers. Features: FEATHER (Pat'd.) Valves; automatic capacity control. Capacities 3 to 30 tons.

Large Self-contained Units



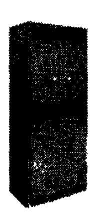
Freon-12 or methyl chloride compressor units for use with "shower" condensers or water-cooled condensers. Features: Worthington FEATHER (Pat'd.) Valves; automatic capacity control. Capacities 25 to 60 tons.

Vertical Two-cylinder Double-acting Compressors



Freon-12 or ammonia; large tonnage compressors; force-feed lubrication; roller main bearings. Crankcase sealed from cylinders, preventing contamination of oil by refrigerant. Equipped with patented Feather Valves; automatic capacity control features. Crosshead incorporated in enclosed crankcase. Capacities up to 250 tons; duplex to 500 tons.

Combined Unit Air Conditioners



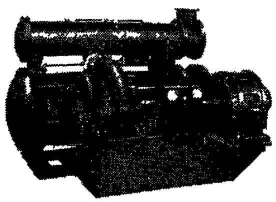
Complete air conditioning system in one attractive, compact unit. Contains compressor, condenser, motor, coils for cooling and heating, fans, filters and accessories. Capacities from 2 to 15 tons.

Miscellaneous

High and low side equipment for every purpose.

Worthington Pump and Machinery Corporation, Carbondale Division

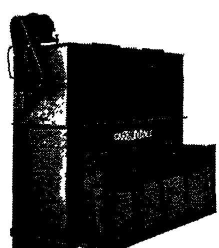
Centrifugal Refrigeration Water Cooling Systems



Freon-11 centrifugal compressor, water cooler and water-cooled condenser in compact unit assembly. Electric motor or steam turbine drive. 56 unit sizes . . . 150 to 1200 tons.

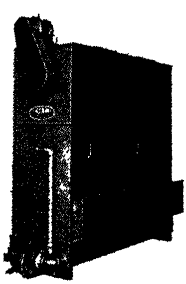
Air Conditioning Units

For Direct Expansion Freon-12 or Chilled Water Circulation



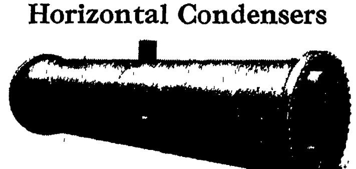
Vertical and horizontal; 500 to 12,000 cfm; large air passages; slow speed, quiet rugged fans; separable sections; readily accessible. The design permits flexibility in installation arrangements.

Shower Condensers



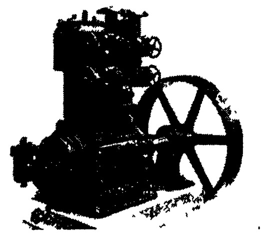
A combined condenser, receiver, and modified cooling tower, in one assembly, for Freon-12 or methyl chloride systems; 2 to 130 tons refrigeration; built in separable sections; all parts easily accessible. Saves 90 to 95 per cent in cost of water.

Horizontal Condensers



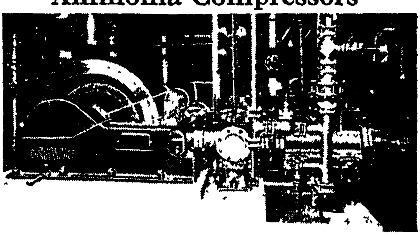
Atmospheric drip type, for warm corrosive waters. Double-pipe for closed systems, can be retubed without shutting down. Multi-pass, as illustrated above, for closed systems and space saving.

Vertical Ammonia Compressors



Pressure-lubricated; roller main bearings; safety heads; patented Feather Valves; belt drive, or direct-connected to electric motor, Diesel or gas engine; ratings from 2 to 160 tons in one unit.

Horizontal Ammonia Compressors



Single and duplex; single-stage and two-stage; belt drive, or direct-connected to electric motor, Diesel, gas or steam engine; patented Feather Valves; ratings from 60 to 750 tons. Automatic capacity control features are easily applied. Space requirements vary depending upon type and drive.

Carbon Dioxide Compressors

A series of convenient types and sizes for every requirement is available.

Liquid Cooling Equipment

Various designs of horizontal single and multi-pass types, for a wide range of services; also vertical types. Chillers for oil dewaxing. Single and double-pipe for milk, wort, chemicals, etc. Cold liquid circulating systems.

American Coolair Corporation

Jacksonville, Florida

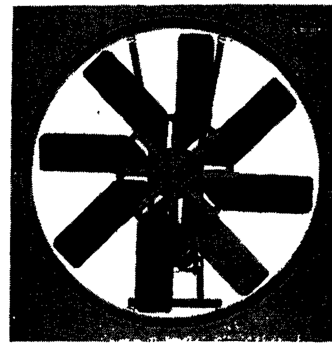
Cooling and Ventilating Fan Systems
For Homes, Offices, Stores, Factories, Etc.
A Pioneer Manufacturer of Attic Fans for Home Cooling
Charter Member—Propeller Fan Manufacturers Association

COOLING BY AIR MOVEMENT

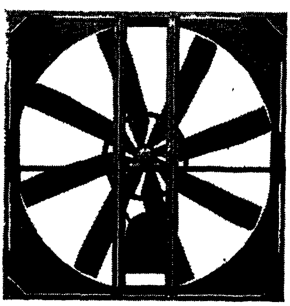
Engineers and architects know well that people working or living within a building can be kept cool, healthfully and economically, by sufficient movement of fresh air. It is not so well known, perhaps, that such cooling requires four to eight times the volume of moving air needed for simple ventilation. Satisfactory cooling requires a complete air change in working or living quarters at least once a minute.

The American Coolair Corporation pioneered in the manufacture of attic fans for home cooling and during the past 12 years, Coolair engineers have been directly responsible for many of the developments in this field.

In planning a Coolair installation, determine cubic content of space to be cooled or ventilated and select fan of ample capacity. For your convenience a table of Recommended Air Changes is included in Coolair's FREE catalog containing detailed suggestions for home and commercial jobs. Write for it today.



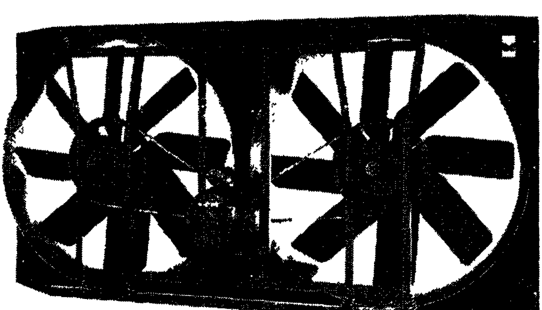
Coolair Type O, showing new built-in springs. Ideal for attic, window and wall installations in homes, offices, restaurants, stores, etc.



Coolair Type S, diameters 6 to 9 ft, capacities up to 150,000 cfm. Used in hotels, factories, auditoriums, etc.

COOLAIR'S QUALITY FEATURES

1. **New Built-In Springs**—completely insulate moving parts from frame, eliminating vibration noise. Simplify window, wall and attic installations. 2. **Light, Compact Fabricated Steel Frame**—fits into many places where fans with bulky metal housings cannot be used. Easy to handle and install. 3. **Reversible**—when equipped with reversible motor, fan will blow in or exhaust at will. 4. **Ball Bearings in Fan Hub**—Eliminate sleeve bearing chatter and end thrust knock. Permit operation in any position (specify ball bearing motor for vertical or angle discharge). Uses grease instead of oil, requiring attention only once a year (once each three years for residential service). 5. **Eight Large, Slowly-Moving Steel Blades**—instead of four or less usually found on cheaper fans. Up to 12 blades on Type S models. Low tip speeds for quiet performance, steady flow of air. 6. **Efficient V-Belt Drive**—and small motor for economical operating speed. 7. **Efficient Long Hour Service Motors**—rubber mounted on adjustable support. Nationally known make approved for this application. 8. **Certified Air Ratings**—in accordance with Standard Test Code of NAFM and ASHVE. 9. **Full Streamlined Orifice**—avoids "spill-off" at end of blades, reduces power consumption.



Coolair Type OT, space-saving, efficient twin unit widely used where headroom is limited. Shown with new built-in springs. U. S. Patent No. 2,108,738

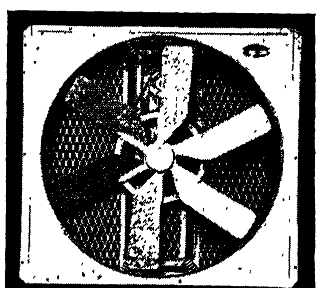
Performance Data—Coolair Belt Drive Fans

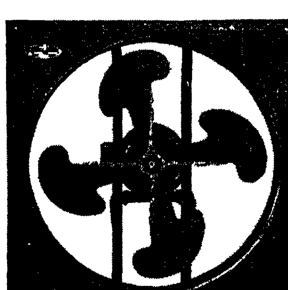
Fan Size		Horse Power	Fan R. P. M.	Cubic Ft. Air Per Min.
28-W	L	1/4	515	6000
2-O	L	1/4	570	6200
	D	1/3	630	6800
2½-O	B	1/4	411	8000
	C	1/3	454	8800
	D	1/2	522	10100
	D	3/4	600	11700
3-O	B	1/4	312	10000
	C	1/3	345	11000
	C	1/2	398	12700
	D	3/4	450	14400
	D	1	500	16000
3½-O	B	1/3	261	13000
	C	1/2	300	15000
	C	3/4	345	17000
	D	1	380	19000
	D	1½	440	22000
4-O	L	1/2	258	19000
	C	3/4	317	22000
	D	1	353	25000
	D	1½	405	28000
4½-O	B	1/2	224	22000
	C	3/4	255	25000
	C	1	276	27000
	D	1½	319	32000
	D	2	355	35000
5-O	B	1/2	200	27000
	C	3/4	225	30000
	C	1	245	33000
	C	1½	282	38000
	D	2	310	42000
	D	3	355	48000
6-S	B	1	155	35000
	C	1½	177	40000
	C	2	195	45000
	D	3	224	50000
	D	5	270	60000
7-S	B	2	163	49500
	C	3	176	53500
	D	5	227	69000
	D	7½	262	80000
8-S	B	3	147	66500
	C	5	176	80000
	D	7½	220	100000
	D	10	259	117000
9-S	B	5	144	93000
	C	7½	165	106000
	D	10	182	117000
	D	15	220	142000
2-OT Twin	B	1/4	455	9800
	C	1/3	500	10800
	C	1/2	570	12400
2½-OT Twin	B	1/3	359	14000
	C	1/2	411	16000
	C	3/4	470	18200
3-OT Twin	B	1/2	312	20000
	C	3/4	360	23000
3½-OT Twin	B	3/4	272	27800
	C	1	300	30700
4-OT Twin	B	3/4	235	36000
	C	1	258	38000
	C	1½	317	44000

B—Very Quiet (Homes, Theatres, Hospitals, etc.).
C—Quiet (Stores, Offices, Restaurants, Barber Shops, etc.).
D—Industrial (Laundries, Factories, Canneries, Bakeries, Pressing Clubs, Garages, etc.).
L—Has adjustable diameter motor pulley for Very Quiet and Quiet performance.

Dimensions in Inches

Fan Size	Overall Height	Overall Width	Overall Depth (Approx)
28-W	31	33	15
2-O	30⅝	30⅝	18
2½-O	36⅝	36⅝	18
3-O	42⅝	42⅝	18
3½-O	49	49	19
4-O	55⅛	55⅛	19
4½-O	61⅛	61⅛	19
5-O	67⅝	67⅝	19
6-S	75¼	75¼	28
7-S	86⅞	86⅞	34
8-S	99	99	38
9-S	111⅛	111⅛	38
2-OT	30⅝	61¼	18
2½-OT	36⅝	73¼	18
3-OT	42⅝	85¼	19
3½-OT	49	98	19
4-OT	55⅛	110¼	22





28-W WINDOW FAN

Equipped with new sound-absorbing springs, safety guard, reversing plug, easily mounted in any standard window, no interference with window operation. See preceding page and Coolair Home-Cooling Fan Bulletin for further data.

DIRECT DRIVE FANS

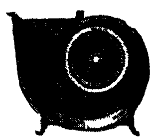
Commercial exhaust fans scientifically constructed for quietness in restaurants, stores, factories, etc. Four sizes 16 to 24 inches in diameter, capacities 1360 to 6900 cubic feet of air per minute. See Coolair Price Sheet for full data.

Bayley Blower Company

1817 S. Sixty-Sixth Street **Branches in Principal Cities** Milwaukee, Wis.

Builders of Heating, Ventilating, Cooling, Purifying, Humidifying and Air Washing Equipment; Exhaust and Drying Apparatus, Mechanical Draft and Blast, Fans and Blowers of all Types

Bayley Plexiform Fan:



Is a multi-blade fan for supplying air for heating and ventilating systems, manufacturing processes, drying systems, forced and induced draft systems. It is suitable for handling high or low temperature gases at medium or low pressure. Will deliver maximum quantities requiring minimum space with great economy.

This is a distinct Bayley product, high class material and workmanship, properly designed to avoid excessive vibration and overstressing of parts. Inlets and outlets are properly sized for maximum delivery and maximum efficiency. Fans are furnished in single or double width of any required arrangement and with sleeve or anti-friction bearings.

Aeroplex Fan:

Is of high speed design with self limiting power characteristics. Application parallel to the Plexiform Fan. Highly efficient and quiet in operation.

Bayley Exhausters and Pressure Blowers:

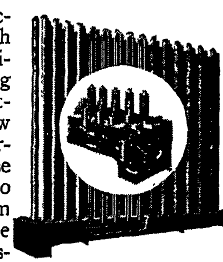
Type "B" exhaust fan is for heavy duty, handling refuse from industrial and textile plants. Type "EX" is used in handling smoke, fumes and dust-laden gases. Type "H" for high-pressure work. These units are highly efficient and of high class design and workmanship.

Bayley Turbo Air Washers, Humidifiers and De-Humidifiers:

The Turbo Atomizer used in the Bayley Washer produces a steady, fine spray. Water at low pressure is delivered to the center of a rapidly revolving cone-shaped rotor provided with atomizing pins set in its periphery. This atomizer requires very little attention, and will operate successfully under low water pressure. The orifices are large and this atomizer, unlike high pressure nozzles, cannot clog.

The Bayley Turbo Air Washer Showing Turbo Atomizer and Eliminator

Bayley Chinook Heating Sections:



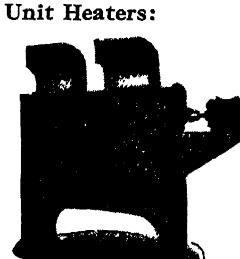
The Chinook section is used with blast heating, ventilating and drying systems, and is suitable for high or low pressure steam circulation. The base is divided into two chambers. Steam enters (see cut) the lower chamber, rising through $\frac{3}{8}$-in. pipes located within the $1\frac{1}{4}$-in. pipes leading from the upper chamber. Condensation takes place in the larger pipes, the water falling into the upper chamber and draining away through the return outlet. The Chinook can be repaired in the middle of the bank without breaking steam connections or taking down a section.

Shipped assembled in smaller sizes, and knocked down in the larger units. May be installed in horizontal or vertical position.

Bayley Chinookfin Heating Sections:

Are the same design as the Chinook Heaters, using heavy gauge copper fin tubes. As compared with Chinook it is much lighter and occupies less space.

Bayley Plexfin Unit Heaters:



This unit incorporates Chinookfin radiation and Plexiform or Aeroplex fans. The fan assembly including top plate and motor is removable as a unit for maintenance and inspection. The heating element is a removable unit. Casing all welded extra heavy gauge. This is an exceptionally high grade unit at a moderate price.

Buffalo Forge Company

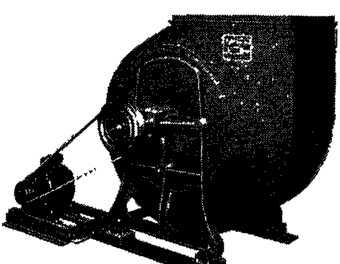
450 Broadway, **Buffalo, N. Y.**

Branch Offices

Albany, N. Y. .... 1305 Standard Bldg.
Atlanta, Ga. .... 305 Techwood Drive
Baltimore, Md. .... 508 St. Paul St.
Boston, Mass., Melrose Sta. .... 486 Main St.
Chicago, Ill. .... 20 North Wacker Drive
Cincinnati, Ohio .... 626 Broadway
Cleveland, Ohio .... 418 Rockefeller Bldg.
Dallas, Texas .... 1801 Tower Petroleum Bldg.
Davenport, Iowa—D. C. Murphy Co., 305 Security Bldg.
Denver, Colo.—Hendrie & Bolthoff Mfg. & Supply Co., 1635 Seventeenth St.
Des Moines, Iowa—D. C Murphy Co., 214 Old Colony Bldg.
Detroit, Mich.—Coon DeVisser Co., 2051 W. Lafayette Blvd.
Greenville, S. C .... 21 Blue Bldg.
Houston, Texas .... 505 Rusk Bldg.
Kitchener, Ont., Canada—Canadian Blower & Forge Co., Ltd.
Knoxville, Tenn.—C. F. Sexton .... 702 Empire Bldg.
Los Angeles, Calif. .... 708 Pershing Square Bldg.
Miami, Fla.—Southern Air Conditioning Corp., 149 N. E. 20th Terrace
Minneapolis, Minn. .... 2102 Foshay Tower
Nashville, Tenn.—Southern Sales Co., 117 Fifth Ave., North
Newark .... 27 Washington St., Room 203
New Orleans, La.—Devlin Bros. .... 1003 Maritime Bldg.
New York, N. Y. .... 39 Cortlandt Bldg., Room 1110
Omaha, Neb. .... 106 North 18th St.
Philadelphia, Pa. .... 703 Cunard Bldg.
Pittsburgh, Pa. .... 431 Fulton Bldg.
Richmond, Va.—Williamson & Wilmer, Inc., Mutual Bldg.
San Francisco, Calif.—Moore Machinery Co., 1699 Van Ness Ave.
St. Louis, Mo. .... 1598 Arcade Bldg.
Seattle, Wash. .... 500 First Ave., South
Toledo, Ohio .... 1922 Linwood Ave.
Washington, D. C.—640 Woodward Bldg., 15th and H Sts., N. W

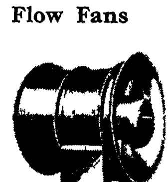
PRODUCTS: Heating and Ventilating Equipment including: Unit Heaters, Multiblade Fans, Pipe Coil Heaters, Buffalo Air Washers, Buffalo Unit Air Washers, Buffalo Unit Coolers, Drying Equipment, Mechanical Draft Fans, Air Preheaters, Exhaust Fans, Blowers, Dust Collectors, Disc Fans, Spray Nozzles.

Buffalo Limit-Load Fans



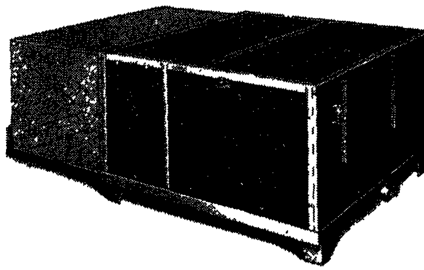
Buffalo Limit-Load Fans for ventilating embody several improvements to deliver stepped-up efficiency under practical conditions. Durably built for years of service. Dynamically balanced. Quiet, economical to operate. Non-overloading characteristic prevents motor from overloading and burning out, regardless of fan load.

Buf-flow Axial Flow Fans



This specially designed high pressure fan—with directional guide vanes—propels the air stream in a true axial direction. Energy losses, are reduced to a minimum with a resulting increase in fan efficiency and marked power savings. What's more, this fan cannot overload and burn out the motor.

Buffalo Conditioning Cabinets



Buffalo "PC" Conditioning Cabinets are flexible in application, available in combinations for simple cooling or complete air-conditioning, including summer cooling and dehumidifying, winter heating and humidifying and year 'round cleaning. Any or all functions may be automatically controlled. These units are neat and compact, exceptionally quiet in operation. Easy to install and easy to service. Cooling capacities, 3 tons up.

Fans for Every Ventilating Need

Buffalo Fans represent over 60 years of specialization in the design and construction of fans for practically every ventilating and air-handling application from small kitchen fans to rugged fans for boiler draft. For complete information state the type of fan you are interested in and a catalog will be sent.

(*See also Page 1129*)

Champion Blower & Forge Co.

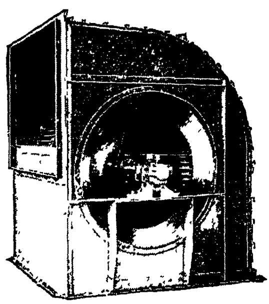
Manufacturers and Engineers

Plant and Offices: Lancaster, Pa.

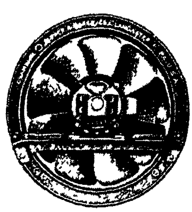
Address Correspondence to Div. 9

Manufacturers of Blowers, Ventilating Fans and Exhaust Fans for Air and Material; and Blast Gates

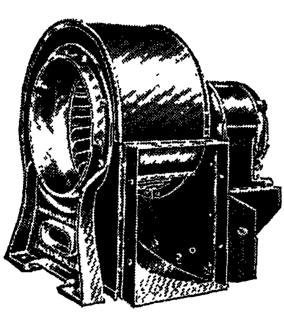
Representatives in Principal Cities



Type S Forward curve ventilating fans, single and double width, as well as direct motor drive.



Super Ventilating fans, direct motor drive up to 36 in. diameter. Motor belt drive up to 48 in. size.



Type BC Backward curve ventilating and exhaust fans, single and double width; belt driven and direct connected electric.

Ventilating Fans—For air conditioning systems and mechanical draft. Manufactured in the forward curved type for slow speed, and extremely quiet operation; also in the backward curved type with its flat horsepower curve characteristics and higher speeds suitable in the smaller sizes for direct connecting to synchronous speed motors. Ventilating Blowers manufactured in sizes up to 60 in. diameter wheel, in single and double width. Belt driven blowers equipped with either ball or high-grade babbit bearing. Direct motor drive can be equipped with any type or characteristic motor desired, in any arrangement.

Disc Fans—Super Ventilating Fans made in direct connected type up to 36 in. diameter, totally enclosed, ball thrust type motors. Slow speed motor belt driven type manufactured in sizes up to 48 in. for attic exhaust work and wherever large volumes of air are to be moved against low static pressures. All disc fans are quiet in operation. Decibel ratings on all fans are available.

Forced Draft Fans—All sizes for use on the smallest to largest boilers. Fans can be furnished with inlet or outlet adjustable louvers for controlling air volume.

Blast Wheels—We are well equipped to manufacture single and double width blast wheels in forward or backward curve type for oil burner and stoker manufacturers, as well as manufacturers of air conditioning units and other ventilating equipment.

Vibration Dampener Sub-Bases—For blower and ventilating equipment. Made with heavy channel iron and rubber vibration eliminator pads to suit size and weight of fan or blower.

Special Fan Equipment—We are in position to engineer and build fans, blowers, or exhausting equipment to meet customers' special needs. A card addressed to Div. 9 will bring you complete catalog data or information on any particular problem confronting you.

DeBothezat Ventilating Equipment Division

American Machine and Metals, Inc.

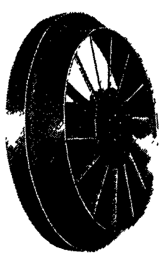
(Main Office and Factory)

901 DeBothezat St., **East Moline, Illinois**

Branches	**District Offices**			**Branches**
ATLANTA BOSTON CINCINNATI CLEVELAND DALLAS DES MOINES DETROIT FORT WAYNE HARTFORD	**Chicago**	**New York**	**San Francisco**	INDIANAPOLIS LOS ANGELES MILWAUKEE MINNEAPOLIS NEW ORLEANS PITTSBURGH PROVIDENCE SAGINAW ST. LOUIS

AXIAL FLOW
PRESSURE FANS

NON-OVERLOADING POWER CHARACTERISTICS
CERTIFIED RATINGS—GUARANTEED PERFORMANCE



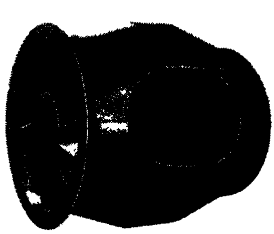
Ventilating Fan Axial Flow

Axial Flow Ventilating Sets

A complete series of volume and pressure axial-flow fans of high mechanical and static efficiencies with a non-overloading power characteristic. These fans offer savings in space, weight and power. Axial-Flow Ventilating Sets are available in a wide range of capacities in sizes 8 in. through 10 ft in diameter, and may be had arranged for direct motor drive or belt drive.

Bifurcators

Designed for handling corrosive or high temperature vapors with direct motor driven fan. Motor is located in chamber open to atmosphere but isolated from gases handled by fan. Installed as integral part of duct system, in any position.



Bifurcator

Multi-Stage Impeller Blowers

Units can be furnished in 2, 4, 6 or 8 stages. Direct motor or belt driven, producing high capacities and static pressures, with non-overloading power characteristics.

"Power-Flow" Roof Ventilators

Designed to provide positive ventilation at all times regardless of temperature, humidity and wind velocity. Guaranteed performance ratings. Equipped with high-efficiency Axial-Flow Pressure Fan, these Power-Flow Roof Ventilators possess the greatest air moving capacity per horse power! Low fan tip speeds permit unusual quietness of operation. Work efficiently against resistance of duct systems. Have non-overloading power characteristic available in a wide range of sizes, speeds, and for all standard electric current.

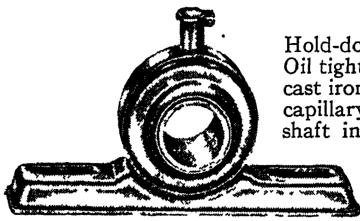
"Power-Flow" Ventilators are aerodynamically correct in design. Note trim appearance—low height.

The above is only a partial list of the ventilating units DeBothezat builds. Our engineers will be glad to give you expert assistance in your ventilating problems—offering you a solution in space, weight and power saving equipment. Catalog on all products sent on request.

The Lau Blower Company

2007 Home Avenue, **Dayton, Ohio**

Manufacturers of Package Unit Furnace Blowers, Blower Wheels, Housings, Pulleys, Pillow Blocks, Complete Assemblies, Window Fans and Attic Fans

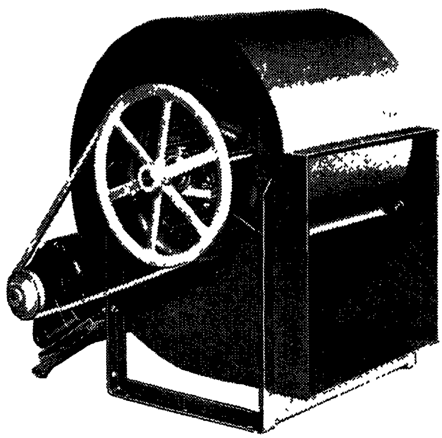


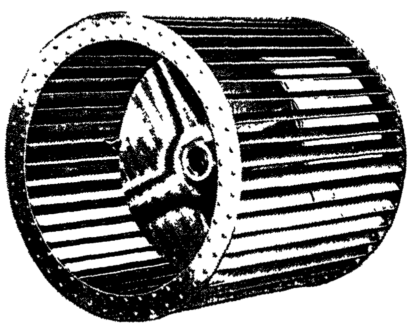
Self-Aligning Pillow Blocks

Hold-down bolts cannot affect the freedom of the bearing. Oil tight steel housing; reservoir holds twice as much oil as cast iron housings; Durex bushing feeds oil to the shaft by capillary action; 100 per cent bearing surface "floats" shaft in oil at all times. Spherical surface of bearing conforms to contour of housing, providing a universal joint action. (*Left*).

100 Series Assembly

A complete blower assembly for manufacturers who fabricate their own casings (also available with top motor mounting). Variable speed drive; automatic belt-tightening device; automatic cut-out on motor; 23 sizes. (*Right*).



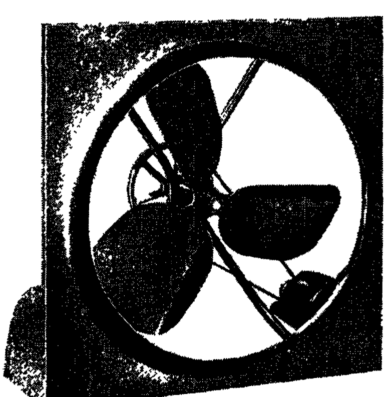


Blower Wheels

Squirrel cage, forward curve, multi-blade type wheels. Double inlet, double width (or single inlet, single width). Dynamically balanced; sizes 5 in. to 30 in. (*Left*).

Housings

Constructed of extra heavy guage steel, venturi die formed housing reduces turbulence. Scientifically designed scroll insures highest efficiency. Any angle discharge furnished. Blower housings available in 10 standard sizes—special sizes on request.



Niteair Panel Type Fan

Three broad, deep-pitched blades provide maximum volume air with minimum power consumption. Venturi type entrance panel assures maximum air delivery, reduces turbulence, and eliminates the most common cause of "air noise." Two self-aligning Durex bearings holding fan shaft rigidly, V-belt drive, the rubber mountings on motor, and end thrust combination avoid vibration. Eight sizes. (*Left*).

Write for the Lau Catalog on any of the products described above. Also for complete data on Lau Blower-Filter Units. Twenty Six Sizes . . . two speed operation available with most models.

The NEW YORK BLOWER COMPANY

GENERAL OFFICES
32nd STREET & SHIELDS AVENUE • CHICAGO
FACTORIES at LAPORTE, IND and CHICAGO, ILL

Experienced Air Engineers
Representatives in Principal Cities

BUYERS' KEY TO *Products*

Products . . . for all industrial processes, ventilation and heating. Specify them for every industrial, commercial and institutional requirement. Check these nine classifications which give thumb nail specifications:

- **CENTRIFUGAL FANS AND BLOWERS**
  Type ME Housed Centrifugal Wheels—Quiet Operating, Slow Speed Type and/or High Speed Wheels with Non-overloading Horsepower Characteristics. Range of Standard Wheel Sizes 15 in. to 80 in. Wheel Diameter. Sizes 15 inch to 80 inch Wheel Diameter.

- **JUNIOR CENTRIFUGAL BLOWERS**
  Type ME Junior Fans, direct connected Motor Driven Range of Sizes 6 inch to 12 inch Wheel Diameter.

- **DISC TYPE (or PROPELLER) PANEL FANS**
  Comet EXHAUSTAIR Ventilating Fans, Automatic Shutters, Power Roof Ventilators, direct connected Motor Driven. Size 10 inch to 30 inch Wheel Diameter. Heavy Duty Type, Pulley Driven, GIANT Disc Type Fans, Regular Sizes 36 inch to 108 inch Wheel Diameter with Round Body Frames.

- **INDUSTRIAL UNIT HEATERS**
  Disc or Propeller Fan, Ceiling Suspension Type, NYBCo COMET Unit Heaters with Molybdenum Alloy Corrosion-Proof and Freeze-Proof, Extra Heavy Welded, Longlife Ferrous Heating Element Suitable for low or high-pressure (unlimited) Steam Pressures. Capacities 24,000 to 300,000 BTU's per Hour. Excel AIR-FLOW Centrifugal Type Factory Unit Heaters. Blower-Type Unit Heaters with Encased Centrifugal Fans with NYBCo Molybdenum Alloy Welded Steel Heating Element (either Blow-through or Draw-through Type). Floor Type, Side-wall or Ceiling Suspension. Capacities 169,000 to 1,000,000 BTU's per Hour.

- **MECHANOVENT UNIT VENTILATORS**
  CLASSROOM Unit Ventilators, highly refined in design and appearance. A De-Luxe Product in every sense. Suited for fully Automatic Temperature and Humidity Regulation. Capacities 750 CFM to 1560 CFM for Classroom Duty.

- **MECHANOVENT UNIT VENTILATORS (Continued)**
  AUDITORIUM Unit Ventilators, Fully Encased Centrifugal Type Units, with or without Fresh Air and/or Recirculation Damper Assemblies, for use in Auditoriums or other places of large public gatherings. Capacities 2,000 CFM to 10,000 CFM.

- **AIR WASHERS**
  A completely engineered line of PEERLESS Air Washers, Air Cleansing, Air-Conditioning and Cooling. Complete with Single- and Double-bank Atomizing Spray Systems, Marine Type Doors, Eliminators, Entering and Back-spray Louvres, Water Strainers, Pumps and Motors, and with or without Humidity Flooding Provisions. Sizes and Capacities ranging from 3,600 CFM to 76,000 CFM.

- **HOT BLAST HEATING SURFACE**
  NYBCo "STEELFIN" Longlife High Pressure Molybdenum Alloy Steel, All Welded, Extra Heavy Duty, Homogeneous Fin-and-Oval-Tube Hot-Blast Heating Surface. Hot-dip Overall Metallic Coating, Including Headers. A Super-quality Product—proof against faults common to Surfaces constructed of Non-ferrous and Cast Iron Materials. An Engineered Product of Sizes and Capacities for Steam Pressures (or Hot Water Equivalents) from 2 lbs. to 150 lbs. duty, High Temperature or Low Temperature Fin Spacings, and a Range of Air Velocities from 400 to 1000 Ft. per Min.

- **VENTO, AEROFIN, and OTHER HEATING AND COOLING SURFACES**
  The various types and makes of Heat Transfer or Heat Exchange Surface, as regularly sold through the outlets of manufacturers of Fan-system Apparatus. These types of Surface are offered in various combinations and sizes, together with full and complete engineering recommendations.

- **MISCELLANEOUS**
  Dust and Shavings Exhausters and Material Conveyor Blowers (Centrifugal Fans with Special Housings and Wheels); Engines, Motors, and V-Belt and Other Drives; Air Filters—Automatic and Cartridge or Renewable Types; Control Devices, including Pressurestats, Thermostats, Humidistats; Turbine Ventilators; Gas-fired Unit Heaters; Specialties; and Other Apparatus for use in conjunction with Complete Blower Systems. Complete data and descriptive matter furnished on specific request.

MEMBER NAFM AND IUHA

Certified Ratings

Write for Catalogues

Full Catalog Matter, Descriptive Bulletins, Performance Tables, Engineering Data and Technical Presentation, Prices, and Complete Information with Illustrations will be furnished upon request of District Representative or by Home Office.

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Since 1889

B. F. Sturtevant Co.

Hyde Park, Boston, Mass.

Sturtevant
REG. U. S. PAT. OFF.
Puts Air to Work

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STURTEVANT, WIS. GALT, ONT. BERKELEY, CALIF.

The Cooling and Air Conditioning Division of B. F. Sturtevant Company

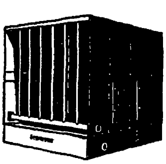
HYDE PARK BOSTON, MASS.

ATLANTA CAMDEN CLEVELAND GREENSBORO LOS ANGELES NEW YORK

DATA ON HEATING, VENTILATING, AIR CONDITIONING AND VACUUM CLEANING EQUIPMENT FOR ARCHITECTS, ENGINEERS, CONTRACTORS

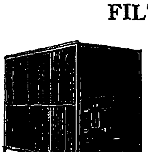
The Publications listed below, and on the following page, have been prepared to aid the architect, engineer and contractor in the selection of proper equipment for industrial, public, and private buildings of all types and sizes. We will gladly send copies upon request.

AIR WASHERS



Built in several types to meet varying requirements in cleaning, cooling, dehumidifying and humidifying air.
Catalog No. 295.

FILTERWASHERS

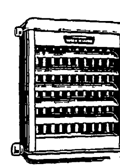


Employing filter pads in conjunction with water sprays, these airwashers clean and humidfiy air, removing all dust and dirt down to micronic fineness.
Catalog No. 453.

COOPERATION

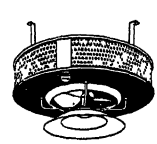
Sturtevant Engineers, located at each of the offices listed, are always ready to cooperate with architects, engineers and contractors in the selection of equipment suitable for any prospective installation.

SUSPENDED SPEED HEATERS



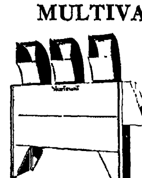
Propeller fan type, for wall or ceiling installation. Fin type heating element. For steam pressures up to 200 lbs, capacities up to 300,000 Btu.
Catalog No. 396.

DOWNBLAST SPEED HEATERS



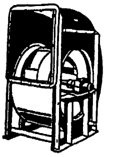
High velocity unit heaters for high ceiling installation. Fin type heating element. Steam Pressures up to 200 lbs. Capacities up to 400,000 Btu.
Catalog No. 454.

MULTIVANE SPEED HEATERS



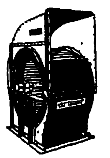
Centrifugal fan type for floor, wall, or ceiling installation. Fin type heating element. For steam pressures up to 200 lbs, capacities up to 2,010,000 Btu. **Catalog No. 452.**

SILENTVANE FANS



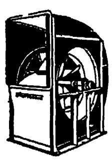
Backwardly curved blade type. A ventilating fan to meet the most exacting specifications, where very high efficiency and exceptionally low power consumption are required. **Catalog No. 457.**

MULTIVANE FANS



Forwardly curved blade type. A highly efficient centrifugal ventilating fan of sturdy construction to meet the general run of ventilating requirements. **Catalog No. 271.**

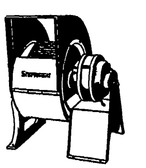
REXVANE FANS



The modern paddle wheel. Correct inlet blade curvature and stream line shrouding retain all good features of the old paddle wheel and greatly increase capacity.

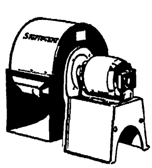
Catalog No. 414.

VOLUME FANS—CONVERTIBLE



Silentvane, Multivane and Rexvane fans shown above can all be furnished with convertible and reversible housings, permitting quick conversion to any horizontal or vertical discharge.

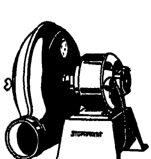
REXVANE VENTILATING SETS



Direct connected; motor driven. For use with ducts, in ventilating small rooms, laboratory hoods and, in general, any room up to 10,000 cu ft contents.

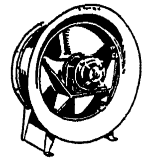
Catalog No. 406.

PRESSURE BLOWERS



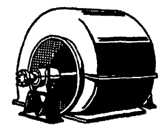
Small, compact forced draft fans, with direct connected electric motor, for coal burning heaters in schools and other buildings. **Catalog No. 297.**

PRESSURE FANS



Designed to operate against resistance of wind, ducts, etc., this rugged axial flow fan combines economical operation cost with high mechanical efficiency. **Catalog No. 444.**

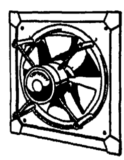
MECHANICAL DRAFT FANS



Forced and induced draft fans to meet any need. Can be furnished with Sturtevant reduction gears and steam turbine, motor or engine drive.

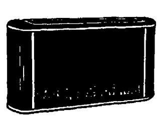
Catalogs No. 409, 445, 447, 448.

PROPELLER FANS



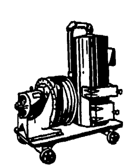
Built in fan sizes from 12 up to 45 in. Electric motors are available for both alternating and direct current, in wide variety of voltages. **Catalog No. 400.**

UNIT VENTILATORS



Combined steam heating and ventilating units widely used in schools and other places. Finished in duco of any standard color. Stainless steel trim. **Catalog No. 377.**

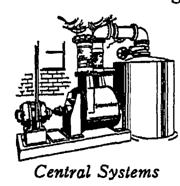
VACUUM CLEANERS



Portable

Sturtevant Vacuum Cleaners are made in both portable and stationary Central System types and in a variety of sizes to meet every commercial and industrial building need.

Features: Quiet operating vacuum producers, high suction for rapid cleaning under all conditions; tools designed individually for specific cleaning operations. Central Systems include piping recommendations to insure correct operation of the system.



Central Systems

Catalog No. 413 (Portable Cleaners).
Catalog No. 397 (Central Systems).

The Torrington Mfg. Co.

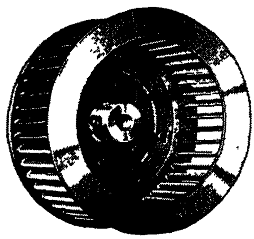
50 Franklin Street, **Torrington, Conn.**

Manufacturers of Blower Wheels and Propellor Type Fan Blades.

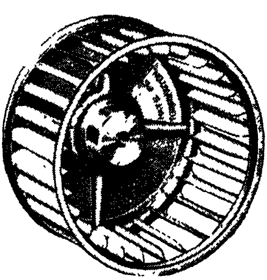
AIRISTOCRAT Quiet Propeller Fan Blades

AIROTOR Blower Wheels

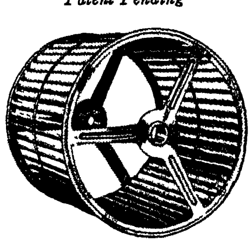
AUTOCRAT Fan Blades



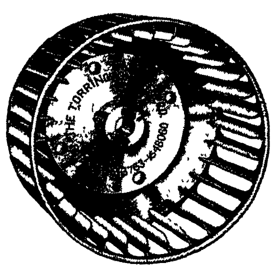
Single Inlet Aluminum Blower Wheel



Airotor Blower Wheel—Single Width—Single Inlet Patents 2,231,062; 2,231,063; Des. 126,043 Patent Pending



Airotor Blower Wheel—Double Width—Double Inlet Spider End Plates—Patents 2,231,062, 2,231,063 Patent Pending



Cup Type Blower Wheel. Pats. 1,513,763; 1,648,060

Torrington Aluminum Blower Wheels produce the smooth, quiet performance which is essential in modern heating and air conditioning units because the unique patented construction breaks up resonance and minimizes noise. Made of aluminum, they resist corrosion and their light weight facilitates quick starting —saves power. Every wheel is statically balanced.

Bulletin lists 34 sizes of single inlet single width and 34 sizes of double inlet double width wheels, including guaranteed capacities for each. Also gives detailed dimensions for all wheels and table of dimensions for housing scrolls. We do not manufacture housings.

Sizes 3 in. to 15 in. diameter in all standard widths.

Torrington Airotor Blower Wheels are light, sturdy and inexpensive—incorporate new principles of design and construction, which insure rigidity and concentricity. **Single Width—Single Inlet** wheel is of simple four-piece construction. **No rivets or welds are used**; concentric rib serving as backing for blade strip is formed at same time as hub socket, insuring trueness of wheel. Rigid radial ribs prevent deflection by thrust. Three thicknesses of metal in rims make for maximum strength. Manufactured in both aluminum and steel in 3⅝ in., 4½ in., 5 in., 6 in., 7½ in., 9 in. and 10½ in. diameters. Same sizes available in DA type double width, double inlet wheels.

Torrington Airotor Blower Wheel—Double Width—Double Inlet—Spider End Plates, has blades punched and formed in a single strip, rigidly held by flanged single piece end rings. Hubs are rigidly mounted by peening. Wheels of 3⅝ in., 10½ in., 12 in. and 16 in. diameter are available at present; 4½ in., 5 in., 6 in., 7½ in., 9 in. and 20 in. sizes are being developed.

Torrington Cup Type Wheels—Used for automobile heaters, gun type oil burners, windshield defrosters, small hair dryers, hand dryers, ice box and refrigerator circulators, window ventilators, exhausters, etc. Made for either clockwise or counter clockwise rotation, of steel, in sizes: 3 in. to 9 in. inclusive.

AIRISTOCRAT Quiet Propeller Fan Blades are widely recognized for their all-around excellent performance. The unique, patented construction embodies entirely new principles in the art of fan design—produces a blade unsurpassed for quiet operation, rugged construction and attractive appearance. Every **Airistocrat** unit is carefully built and the blades are hand gauged for correct contour and alignment. Statically balanced, these blades deliver full air volume with a minimum of noise. Aluminum alloy blades and steel spiders are standard except where otherwise noted. Rotation is clockwise only (facing air delivery side).

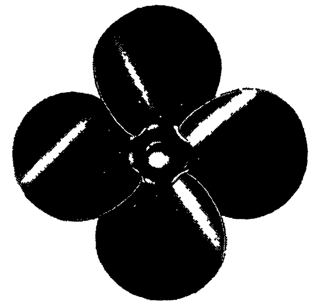
Available in the following finishes: 1. Plain—no finish on blades, spiders or hubs. 2. Blades with no finish; spider and hub with cadmium plate or black lacquer. 3. All black lacquered, with or without center button. 4. Buff and lacquered blades, black lacquered spider and hub, with or without center button. Catalog gives detailed dimensions and guaranteed performance curves recorded under *NEMA* and *NAFM* code tests at various speeds for each of the **Airistocrat** models described below.

"Standard" Series—Has blades mounted on a steel spider. Sturdy, attractive steel or aluminum blades which have withstood extreme laboratory breakdown tests. Sizes 8 in., 10 in., 12 in., 14 in., 16 in., 18 in. and 20 in. diameters in a variety of pitches to meet every need.

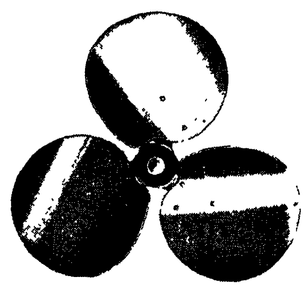
Three Blade "Y" Series—The design of this blade is the result of two years of laboratory experiment to produce a better air circulator blade. At recommended speed these blades produce a high velocity air stream effecting deep penetration with unusual quietness. Sizes 10, 12, 14, 16, 18, 20, 24 in. and 30 in. diameters, steel or aluminum blades.

Pressure "P" Series—Similar in construction to "Standard" Series but with blades especially designed for higher pressures. Sizes 10 in., 12 in., 14 in., 16 in. and 18 in. diameters.

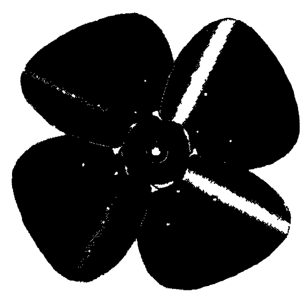
Pressure "U" Series—Two and four blade models of steel designed for pressure operation. Sizes 20 in., 22 in., 24 in., 26 in., 28 in. and 30 in. diameters. 24 in. and 30 in. sizes suitable for attic fans. Bulletin gives complete specifications and ratings.



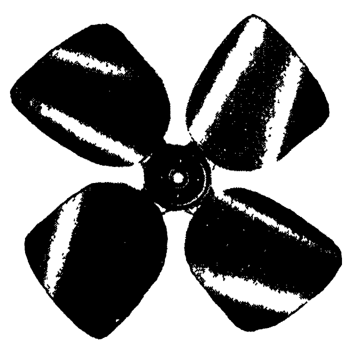
Airistocrat "Standard" Series
Pats. 2,072,322 and 2,021,707



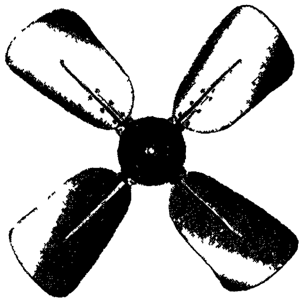
3-Blade Airistocrat "Y" Series



4-Blade Airistocrat Pressure Fan "P" Series

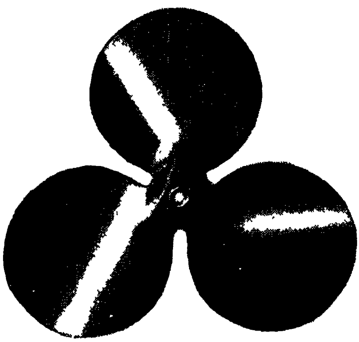


4-Blade Airistocrat Pressure Fan "U" Series
(also made in two blade model)



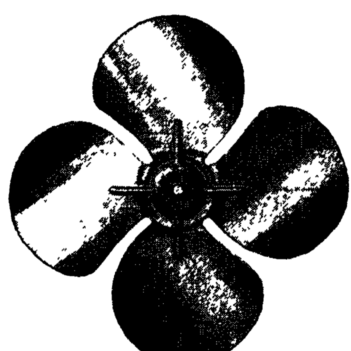
4-Blade Airistocrat Attic Fan "A" Series (also made in 2 and 3 blade models)

AIRISTOCRAT "A" Series Attic Fan Blades are the result of extensive study and experiment to produce blades having extraordinary efficiency, to sell at lower than average prices. LOW COST is possible because tools are interchangeable for production of either 2, 3 or 4 blade models in any diameters from 24 in. to 48 in. inclusive (sizes 24 in., 30 in., 36 in., 42 in. and 48 in. are standard). Construction approved only after severe breakdown tests. The extremely high EFFICIENCY is attained by the application of correct principles of design. Blades, spiders and hubs are of steel. Available in the following finishes: 1. Plain. 2. Aluminum lacquered blades, black lacquered spider and hub. 3. All one color lacquer. Bulletin gives detailed dimensions and specifications; also performance data.



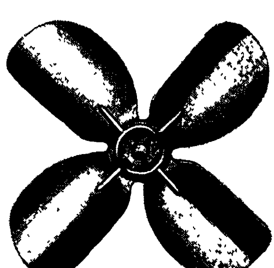
Airistocrat 3-Blade "One Piece" Series

3-Blade "One-Piece" Series Propellor Fan—An attractive, inexpensive one-piece blade incorporating the **Airistocrat** features for quiet operation. Available in both steel and aluminum. Sizes 8 in. and 10 in. diameters.



4-Blade "One Piece" Airistocrat Fan

4-Blade "One-Piece" Series Propellor Fan—An exceptionally rigid model blanked from one piece of metal with four wide blades. Quieter than narrow blade types. Made in both steel and aluminum. Clockwise rotation only (viewing air delivery side). Sizes 8 in., 9 in., 10 in., 12 in., and 16 in. diameters. Available in the following finishes: 1. Plain. 2. Lacquered. 3. Nickel or cadmium plated (steel only).



Autocrat Fan Blade

AUTOCRAT Fan Blades—For auto heaters, windshield defrosters, electric heaters, etc. Have been standard ever since these devices were first marketed. Made in sizes 3 in., 4 in., 4½ in., 5 in., 5¼ in., 5½ in., 6 in., 6¼ in., 6½ in., all four blades, also 7 in. 5-blade, in one piece of cold rolled steel or aluminum with brass hubs, complete with set screw. ¼ in. bore is standard. Either clockwise or counter clockwise rotation (expressed when looking at air delivery side of fan). White nickel is standard finish for steel blades. Bulletin gives complete specifications an ratings.

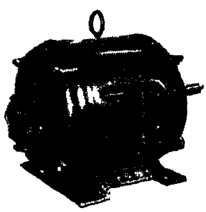
Wagner Electric Corporation

6403 Plymouth Avenue, **Saint Louis, Mo., U. S. A.**

Wagner Motors are built in a wide range of mechanical and electrical types to meet the varied requirements of the air-conditioning industry. These motors are carefully designed and skillfully constructed to make them quiet in operation and completely reliable.

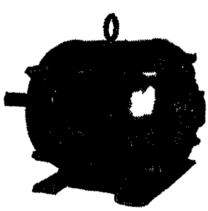
WAGNER POLYPHASE MOTORS

Single-Speed Squirrel-Cage Motors
(Type RP)



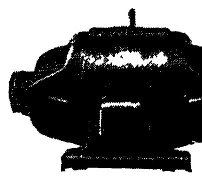
Normal starting-torque—normal starting-current for driving radial compressors, fans and blowers. High starting-torque—low starting-current for driving reciprocating compressors. 2 and 3 phase; 1/6 to 400 hp.

Multi-Speed Squirrel-Cage Motors
(Type MRP)



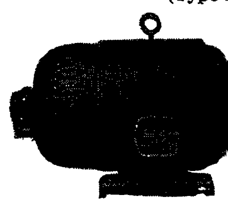
To use where several constant speeds reduce operating costs. Can be furnished with normal starting-torque or high starting-torque, with two, three or four speeds. Variable torque characteristics for fan service, constant torque characteristics for compressor service. 3 phase; 1/2 to 125 hp.

Fynn-Weichsel (Synchronous) Motors
(Type RN)



Wagner Fynn-Weichsel motors are leading-power-factor motors, having high starting-torque, low starting-current, high pull-in torque, constant speed at all loads, and ability to carry heavy intermittent overloads. Especially desirable where power-factor improvement or constant speed is required. 2 or 3 phase; 7 1/2 to 200 hp.

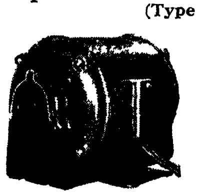
Special Compressor Motors
(Type RT)



The Wagner RT motor was specially developed to meet the demand for a motor with high starting-torque and very low starting-current. 2 and 3 phase; 40 to 150 hp.

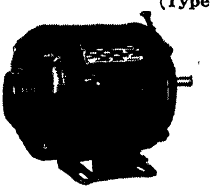
WAGNER SINGLE-PHASE MOTORS

Repulsion-Start-Induction Motors
(Type RA)



Brush-lifting (assures quiet operation, long brush and commutator life). For high-starting-torque heavy-duty applications. Open, totally-enclosed, and drip-proof; rigid or resilient-mounted. 1/6 to 1 1/2 hp; rigid mounted, 2 to 15 hp.

Split-Phase Motors
(Type RB)

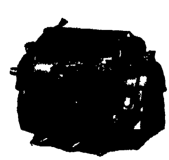


Long-life switch and unbreakable steel frames. Open, drip-proof and totally-enclosed; rigid or resilient-mounted. 1/20 to 1/3 hp.

Capacitor Motors
(Type RK)

Condenser-start induction-run. Drip-proof or totally-enclosed end-plates; riged or resilient-mounted. 1/8 to 3/4 hp.

Shaded-Pole Fan Motors
(Type M)



Single-phase induction of simple construction requiring no complicated starting equipment, ideally adapted to fan and blower drives in which the fans are mounted directly on the motor shaft. Totally-enclosed and open type, rigid base round frame, or resilient mounted, with or without 3-speed regulator. 1/25, 1/80, 1/40 and 1/30 hp ratings.

M42-6

GENERAL ELECTRIC COMPANY

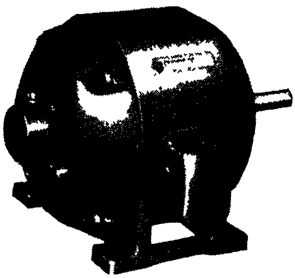
SCHENECTADY, N. Y.

Sales Offices, Warehouses, Service Shops and Distributors in Principal Cities
For Code Wire, Conduit Products, Wiring Devices, Insulating Materials, etc.,
Address—APPLIANCE AND MERCHANDISE DEPARTMENT, BRIDGEPORT, CONN.

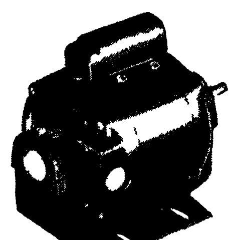
MOTORS FOR HEATING, VENTILATING, AND AIR CONDITIONING

General Electric offers a complete line of motors for compressors, fans, and pumps from which you can select easily the motors with electrical and mechanical characteristics best adapted to your equipment. Many of the most frequent applications are listed below. Complete information on other types of motors—vertical, enclosed, etc., with various electrical and mechanical modifications,—may be obtained at the G-E office near you.

For additional information ask for motor catalog GEA-623.



Tri-Clad induction motor. Type K, polyphase



Capacitor fractional-horsepower motor. Type KC

SOME G-E MOTORS AND THEIR USES

Application	Speed	Type Winding	Type	Horsepower Range	Classification
Fans and Centrifugal Pumps	Constant or Adjustable	Shunt	B & CD	1/8-200	Direct Current
Reciprocating Pumps and Compressors		Compound	B & CD	1/8-200	
	Constant	High Torque Capacitor	KC & KCJ	1/4-3	Single Phase Alternating Current
Small Direct Connected Fans	Constant	Resistance Split Phase	KH	1/40-1/3	
		Shaded-pole	KSP		
	Constant or 3-Speed	Low Torque Capacitor	KC	1/50-1	
Belted Fans, Centrifugal Pumps		General Purpose	KC	1/4-3	
		Capacitor	KC	1/8-3	
		Repulsion Induction	SCR	5-10	
Reciprocating Pumps and Compressors	Constant or Multispeed	Squirrel Cage	K or KB	1/4-1000	Polyphase Alternating Current
		High Starting Torque	K & KG	1/4-5 5-100	
Pumps, Compressors, Fans	Constant or Adjustable	Wound Rotor	M & MB	½-1000	
	Constant	Synchronous	TS	25-2000	

Types of Enclosures—Open—protected from falling objects or dripping liquids. Splashproof—where wetness is a factor.

Totally enclosed—for complete protection. Explosion-proof—where explosive fumes or dusts are encountered.

This Company will gladly assist in the solution of any electrical problems in relation to heating and ventilation

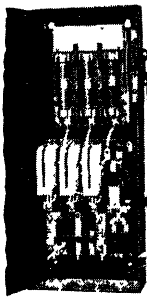
GENERAL ELECTRIC COMPANY

SCHENECTADY, N. Y.

Sales Offices, Warehouses, Service Shops and Distributors in Principal Cities

For Code Wire, Conduit Products, Wiring Devices, Insulating Materials, etc., Address—APPLIANCE AND MERCHANDISE DEPARTMENT, BRIDGEPORT, CONN.

CONTROL FOR HEATING, VENTILATING, AND AIR-CONDITIONING MOTORS

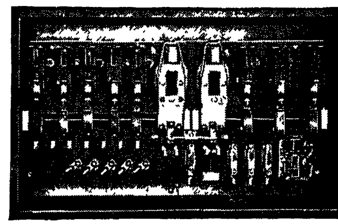


CR7008 combination a-c magnetic full-voltage starter. Provides short-circuit protection, as well as overload and under-voltage protection

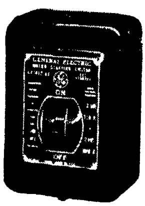
The General Electric line of standard control offers manual or automatic equipment for compressors, fans, or pumps driven by any type motor which you require, providing full protection for your motor, especially those listed on the preceding page.

For special applications General Electric controllers can be designed to meet your exact requirements.

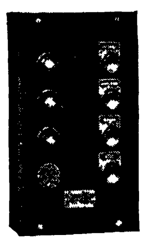
For additional information ask for control catalog GEA-606.



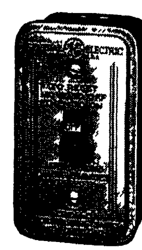
CR7107 a-c magnetic controller for use with multi-speed squirrel-cage motors driving pumps, fans, or blowers



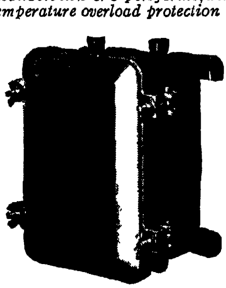
CR1062 manual full-voltage starter for small polyphase motors. Available in 2-or 3-pole forms, with temperature overload protection



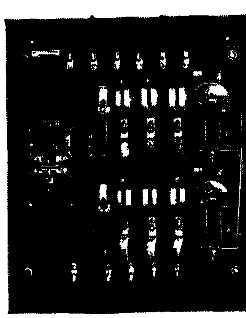
CR2940 indicating-light push-button station. Used with magnetic controllers to indicate speed of fan or blower



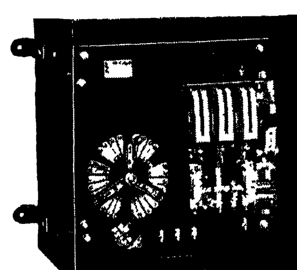
CR1061 manual starter for fractional-horsepower motors, a-c or d-c. Available in 1- or 2-pole forms, with temperature overload protection



CR7006-D51FS quiet magnetic switch. For full-voltage starting of squirrel-cage motors up to 5 hp, 220 volts. For applications where quiet operation is required, such as on fans or domestic air-conditioning systems



CR7896 throwover panels. To transfer motor or lighting load to emergency source of power if normal source fails. Retransfers load when power returns to normal source



CR7764 a-c speed-regulating controller for wound-rotor induction motors. For controlling the speed of motors driving ventilating fans and blowers. Provides undervoltage and overload protection

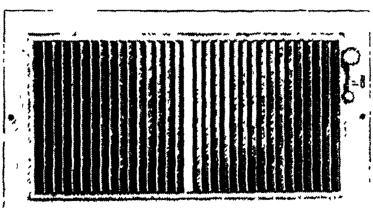
A complete line of accessories, including pressure governors, pressure switches, float switches, electrically operated valves, and indicating Selsyns, is available.

This Company will gladly assist in the solution of any electrical problem in relation to heating and ventilation

Air Control Products, Inc.

Coopersville, Michigan

AIR CONDITIONING REGISTERS AND GRILLES
ADJUSTABLE GRAVITY REGISTERS
FLOOR REGISTERS AND FACES
ATTIC-LOUVERS ● DAMPER CONTROL SETS

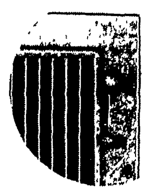


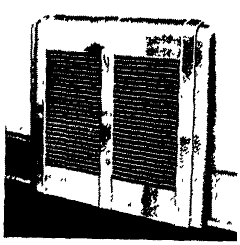
Each fin may be easily adjusted, even after the registers are installed, by means of a special tool that is provided.

Air Conditioning Registers

No. 10 Series Registers assure uniform air distribution for they offer a combination of both vertical and horizontal control of the air stream. Adjustable vertical fins plus adjustable horizontal louvers—immediately behind the register face—give positive air deflection in both planes. A small stop below the operating knob can be set for any deflection from 15 degrees upward (for cooling) to any desired downward deflection (for heating). See illustration below. This easy adjustment also makes the No. 10 Register suitable for Year-'Round Air Conditioning. Registers are regularly supplied in a tough, moisture resistant buff priming coat and are equipped with a sponge rubber gasket seal. This design furnished in baseboard and sidewall types—also available as grille without horizontal valve.

No. 110 Series Registers—an attractive low-priced register with a single shutter valve mechanism. A quality register in every detail—adjustable vertical fins provide directional flow—sponge rubber gasket eliminates streaked walls. The close similarity in appearance to the No. 10 Series makes it possible to use both on the same installation.





Adjustable Gravity Type Registers

No. 20 Registers offer modern styling; ample free area, for gravity installations, and are adjustable for forced air. A universal register—efficient for new gravity installations or for converted forced air installations. Self-sealing sponge rubber gasket, supplied on all registers, makes an air tight seal between wall and register. Beautiful Metalescent Finish furnished as standard. Also available in sidewall types.

Floor Registers and Faces are of grid type construction—resulting in a rigid floor register with ample free area and heel-proof mesh; shallow valve is also adaptable for sidewall installations. Available in a complete range of sizes.

Attic Louvers for attic ventilation and **Damper Control Sets** for forced air dampers are also included in the complete AIR CONTROL Line.

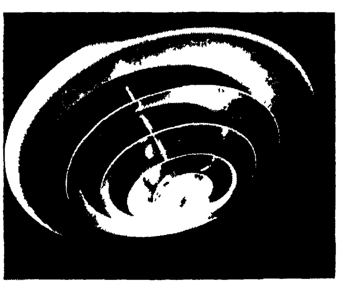
Write for catalog giving complete information on **Air Control's** entire line of products.

Anemostat Corporation of America

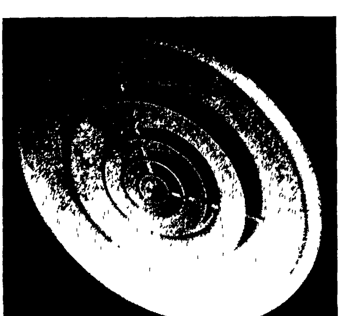
10 East 39th Street, **New York City, N. Y.**

THE ANEMOSTAT HIGH VELOCITY AIR DIFFUSER

Anemostat Type "A"



Anemostat Type "B"



Anemostat Type "C"

THE ANEMOSTAT PRINCIPLE

ANEMOSTATS produce unparalleled results because they are the only air diffusers which operate on the following interdependent principles:

1. Air expansion within the device, which reduces velocity instantly.
2. True Aspiration, which causes room air equal to 30 to 35 per cent of the supply air to be drawn into the device where it is mixed with the supply air. The percentage of aspiration depends on the type of Anemostat used.
3. Creation of a multiplicity of air currents and counter-currents at low velocities, which causes slow but adequate secondary air motion.

Type "A" Anemostat is a combination device for supplying air and either extracting it or returning it to the conditioner or heater. Designed to extract or return 75 cfm of room air for every 100 cfm of supply air. This percentage of extract or return may be increased or decreased by varying the extract velocities. It furthermore has an aspiration effect of 30 per cent. May be used with velocities up to 2500 fpm, and wherever both supply and return, or extract are required through the same unit. Should not be used with ceiling heights exceeding 16 ft.

Type "B" Anemostat is a diffusion device for supply air only. It has 35 per cent aspiration. May be used with velocities up to 4000 fpm and is suitable for industrial and commercial installations. Can be used on either exposed or concelaed duct work.

Type "C" Anemostat is a diffusion device for supply air only. It has 35 per cent aspiration. May be used with velocities up to 2500 fpm. Must be installed flush with ceiling and cannot be used on exposed duct work.

Type "W" Anemostat is a device for the diffusion of supply air from the wall. It has an aspiration effect of 35 per cent. Excessive air motion from the floor up to and including the breathing level is eliminated and the temperature differential, both horizontally and vertically between points in the occupied zone is reduced to a minimum. The effective diffusion covers an area within a radius of 180 deg. This result cannot be obtained by any other method which introduces air from a wall.

Type "CSL"—Cove Lighting Combination of an indirect cove lighting unit with a Type "CSL" Anemostat. Particularly suitable for use in theatres and auditoriums and wherever varied and unusual lighting effects are desired. The reflecting unit and wiring trough are mounted on top of the largest cone of the Anemostat and are entirely concealed when the Anemostat is installed.

Pendant Lighting Fixtures of all types, standard or special, can be hung from the center of the Ceiling Anemostat.

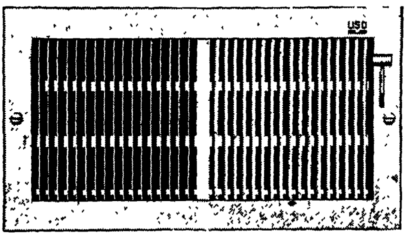
"No Air Conditioning System is better than its Air Distribution"

The Auer Register Co.

3608 Payne Avenue, **Cleveland, Ohio**

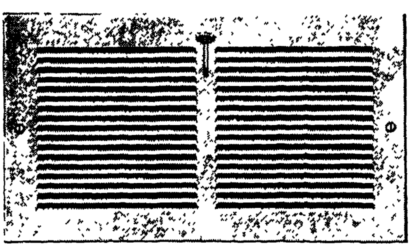
Manufacturers of Registers and Grilles for Gravity and Air Conditioning Systems; Wrought Metal Grilles for Concealing and Protecting Radiation

AIR CONDITIONING REGISTERS AND GRILLES



Airo-Flex No. 4432 Register—No Frame

Multi-louvres adjustable up, straight or down. Grille bars adjustable for right or left flow. Adjusting tool furnished. Grille to match.



Airo-Flex No. 7032 Register—No Frame

Grille bars set to direct air downward at 22½ deg., but adjustable for other angles. Single louvre. Grille to match.

Fin-Flex No. 5030 Register with Band Iron Frame

Flexible fins ⅜ in. on center offer satisfactory one-time adjustment. Adjusting tool furnished with every order. Horizontal fins, furnished as standard. Vertical fins furnished if specified. Same design furnished also without valve, as a return.

The Auer line of registers and grilles for heating and air conditioning systems is modern and complete, offering a wide choice of styles for every purpose. Only a few representative models are shown on this page.

Airo-Flex "4000" Registers have *vertical* grille bars adjustable with tool, as shown, for combination or single current, straight, right or left. For up-and-down flow, Multi-louvre Back Blades, operated by lever, direct current at desired angle up, straight, or as much as 22½ deg. down.

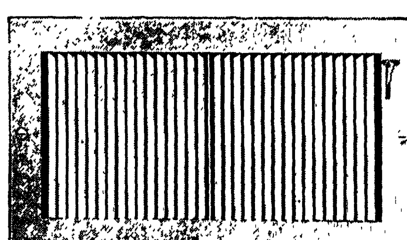
Airo-Flex "7000" Registers have *horizontal* grille bars adjustable in same manner as "4000" and are equipped with *single* blade louvre. A high quality *economy* register.

Fin-Flex Registers and Grilles are made with either vertical or horizontal fins easily adjusted at time of installation for single or multiple air current in any direction.

Dura-Flex Register and Grilles are also furnished with blades adjustable for any desired air flow. Other Dura-Flex designs are available with fixed blades.

DuraBilt Floor Registers and Cold Air Faces are assembled with steel cross-bar construction, all cross joints locked and mortised. These should be specified whereever extra strength is required. They come in medium or narrow mesh.

All Auer models are designed with due regard for air capacity, and supplied in all required sizes and finishes. Complete Catalog 41, showing all types for air conditioning and gravity heating, furnished on request Special Grille Catalog "G" also available.



Dura-Flex No. 8132 Register—No Frame

Adjustable bars ⅜ in. on center.
Also furnished with horizontal bars (adjustable). Small, convenient adjusting tool furnished with each order. Same design furnished also without valves, as a return.

Barber-Colman Company

Rockford, Illinois

ENGINEERED AIR DISTRIBUTION OUTLETS

Venturi-Flo

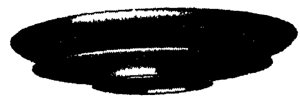
Venturi-Flo is a spun-steel overhead type air diffuser with flow characteristics similar to those of the well known fluid flow measuring device—the Venturi-Meter. The relationship between the neck area of the unit proper and the venturi throat area is so proportioned as to create a slight back pressure in the neck at all times, thereby automatically insuring uniform distribution around the entire periphery of the unit.

Three types of units are available, the recessed, the flush and the surface types. A wide range of sizes permits handling air volumes up to 15,000 cfm per unit.

Fittings for attaching any standard light fixtures to the outlets may be obtained for all three designs. They can also be furnished as combination supply and exhaust units, and with adjustable dampers.

Venturi-Flo—Recessed Type

Venturi-Flo—Flush Type



Venturi-Flo—Surface Type

Uni-Flo

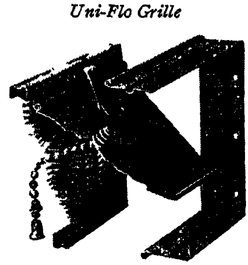
Uni-Flo grilles and registers are especially designed for air conditioning applications. They are engineered and prefabricated with directional flow aspirating fins for each individual installation. Proper air distribution is assured and the necessity of adjustment after installation obviated.

Uni-Flo grilles can be furnished in various shapes and sizes and for plane and curved surfaces.

Registers are similar in construction to grilles, but with the addition of spring loaded, positive closing chain or key operated dampers.

Electro plated finishes: Gunmetal, brushed bronze, plain zinc, buffed zinc, brushed zinc, satin copper. Also available in plain metal, grey prime coat, clear lacquer, and satin aluminum.

Uni-Flo Grille



Uni-Flo Register

Uni-Fin

Uni-Fin grilles and registers are designed especially for residential warm air installations, and are available in standard sizes with prime coat or electroplated finishes.

(*See also Page 990*)

Uni-Fin Register

W. B. CONNOR ENGINEERING CORP.

114 East 32nd Street, New York, N. Y.

DOREX Division

Offices in All Principal Cities

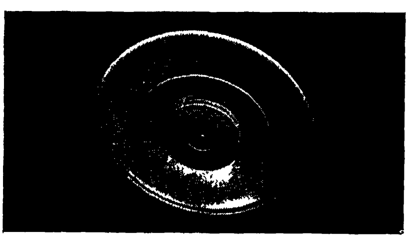
Canadian Representative: Arthur S. Leitch Co., Ltd., Toronto, Ontario

Manufacturers of KNO-DRAFT Adjustable Ceiling Air Diffusers

Dorex KNO-DRAFT Adjustable Air Diffusers insure efficient air distribution, adequate aspiration, noiseless and draftless diffusion, and uniform temperature throughout the occupied zone, regardless of the season or ventilation requirements. Every KNO-DRAFT unit is easily and quickly adjustable for system balancing or seasonal regulation. For instance, during the heating season warm air can be forced downward to obtain proper mixture of room and supply air. Adjustability is a highly advantageous Dorex feature provided at no extra cost.

The KNO-DRAFT Diffuser will effectively handle large volumes of air; pre-mixing room and supply air. It permits the use of higher duct velocities—resulting in smaller ducts and lower costs. Hence duct designs are simplified and fewer outlets are required. KNO-DRAFT Diffusers blend well with any architectural treatment—classical or modern. They are simple in construction, light in weight, and easily installed. Reasonably priced, the KNO-DRAFT Diffuser is a unit that will meet all specifications. Literature, catalogs, and complete engineering data will be furnished upon request.

Model F KNO-DRAFT Diffuser

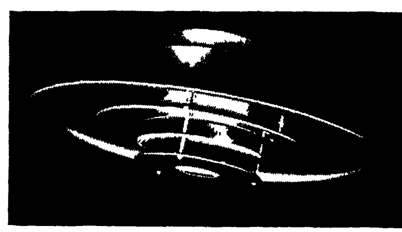


Model "F"—For Supply Air—attractive—light, yet sturdy—for high or low ceilings or attachment to exposed duct work. Anti-smudge rim prevents streaked ceilings. Sizes $2\frac{1}{2}$ in. to 42 in. in neck diameter for capacities from 10 cfm to 20,000 cfm per unit.

Model FL KNO-DRAFT Diffuser

Model "FL"—With built-in lighting. All Dorex KNO-DRAFT Diffusers are available with direct or indirect lighting—the ultimate in modern decorative effects.

Model SR KNO-DRAFT Diffuser



Model "SR"—For Combination Supply and Return air to simplify duct work. Sizes 6 in. to 42 in. supply air neck diameter for supply capacities from 10 cfm to 9,000 cfm per unit, with central return air throat for 75 per cent of supply capacity.

Model FT KNO-DRAFT Diffuser

Model "FT"—of extra sturdy construction for heavy duty in buses, trolley cars, marine craft, etc., sizes $2\frac{1}{2}$ in. to 42 in. neck diameter.

Hendrick Manufacturing Company

Hendrick Perforated Metal Grilles

48 Dundaff Street, **Carbondale, Pa.**

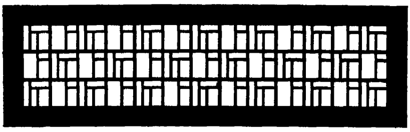
SALES OFFICES IN PRINCIPAL CITIES—CONSULT TELEPHONE DIRECTORY

PRODUCTS—Hendrick Perforated Metal Grilles; Mitco Open Steel Flooring; Mitco Armorgrids; Mitco Shur-Site Treads.

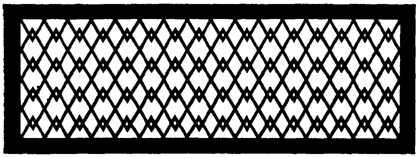
HENDRICK PERFORATED METAL GRILLES

To architects, engineers, contractors and others who buy or specify grilles, Hendrick offers literally hundreds of designs from which to select the pattern or patterns best suited to specific applications.

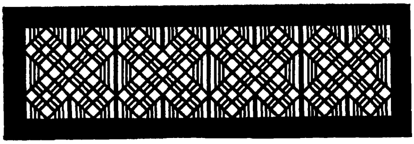
In addition to those popular designs which have been specified so consistently that they are today regarded as standard patterns, Hendrick offers a number of exclusive designs, many of them covered by design patents. Originally designed to meet specific requirements, these Hendrick patterns are available, without premium, to those who seek something that is distinctive as well as different.



Musak



M No, 9



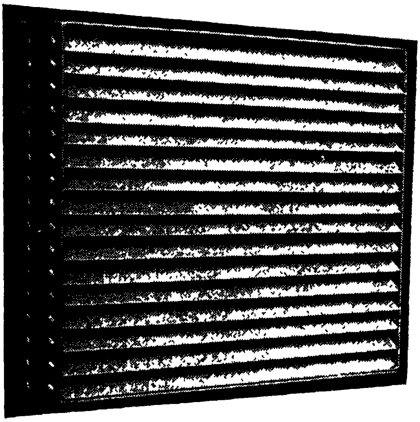
La Crosse

All Hendrick Grilles are characterized by clean-cut perforations and fine finish. In addition, Hendrick grilles are put through a special flattening operation which insures easy installation.

HENDRICK FIXED LOUVRE GRILLE

One of the most popular grilles in the Hendrick line is a door grille, developed originally for hotels and hospitals but equally ideal for bathroom doors in residences.

Hendrick Fixed Louvre Grille is built up of a series of strips bent to a fixed angle and rigidly fastened into a band frame, a construction permitting free circulation of air but preventing vision through the grille from any angle. Easily installed in any door.



Hendrick Fixed Louvre Grille

Regularly furnished in No. 18 U. S. Gauge Steel; also obtainable in other commercially available metals.

Write on your letterhead for a copy of 194-page handbook, "Hendrick Grilles."

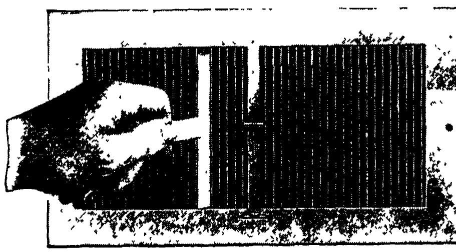
Hart & Cooley Manufacturing Co.

Established 1901

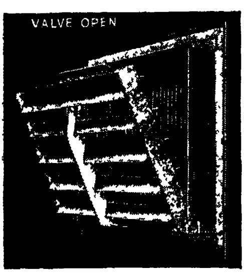
Air Conditioning Registers and Grilles - Warm Air Registers
Damper Regulators - Furnace Regulators - Pulleys - Chain

Holland, Mich.

NO. 75 DESIGN—FLEXIBLE FIN TYPE with TURNING BLADE VALVE to provide DOUBLE DEFLECTION. Also without Valve as Grille or Intake



CONTROL OF AIR FLOW IN TWO PLANES



Instant Adjustment of Air Flow (Up, Straight or Down)

Is obtained by turning the regulator on the register face to the proper setting with a key furnished with each register. When the valve is opened, as shown at the left, the individual valve louvres automatically stop in position to provide the proper air flow—Up (Fig. 1) for cooling systems to avoid drafts; Straight (Fig. 2) for ventilating systems; Down (Fig. 3) for heating systems to prevent stratification. When the valve is closed, as shown at the left below, it completely stops the flow of air.

Air Flow Can be Quickly Adjusted Sideways

No. 75 Design has a flexible fin-type face. Each fin may be twisted individually with a wrench furnished with each register or grille to provide any desired sideway deflection of the air flow.

Greatly Reduced Turbulence and Resistance

Figs. 1, 2, and 3 show the air flow with No. 75 Design; Fig. 4, with the conventional register. Compare the turbulence in the stackhead of the latter with the smooth flow obtained with No. 75 Design. So efficient is No. 75 Design that there is actually less resistance with this register, using a standard stackhead, than if no register at all were used.

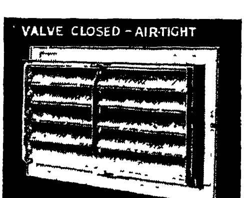


Fig. 1

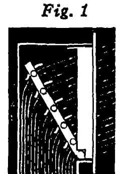


Fig. 2

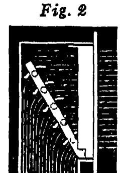


Fig. 3

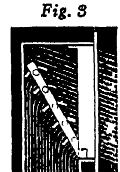
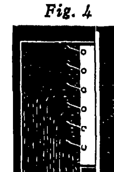
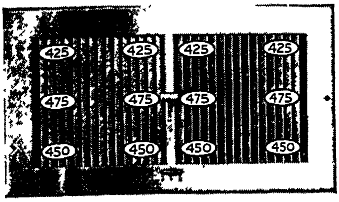
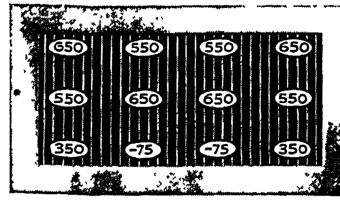


Fig. 4





Velocities with No. 75 Design



Velocities with Conventional Register

EVEN DISTRIBUTION OF AIR OVER ENTIRE FACE

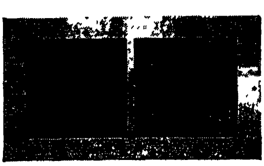
The turning blade valve distributes the air evenly with a uniform velocity over the entire face, as shown in Figs. 1, 2, and 3 on the preceding page. Note how the air rushes through the upper part of the face with a conventional register, as shown in Fig. 4. Since the entire face of No. 75 Design register is utilized for discharge of air, smaller and in some cases fewer registers can be used without causing excessive velocities.

Prevention of Streaked Ceilings—With either UP, STRAIGHT, OR DOWN deflections the air does not strike the ceiling immediately in front of the register; streaked ceilings are thus avoided.

Excellent Concealment of Duct—The depth and close spacing of the vertical bars, combined with the valve, provide almost complete concealment of the duct, adding considerably to the pleasing appearance of the register face.

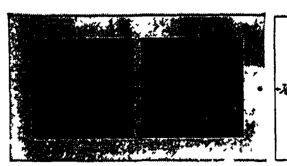
Special Settings—No. 75 Design functions equally well when located at the end of a horizontal duct or, by installing it upside down, when the air is delivered to it from above.

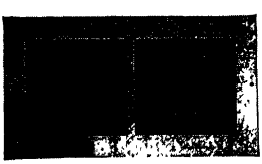
AVAILABLE IN FOUR TYPES



With Turning Blade Valve

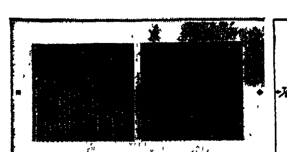
No. 751 Register (Left) has Sponge Rubber Gasket and $\frac{3}{16}$ in. turndown. No. 754 Register (Right) is similar except has $\frac{7}{8}$ in. projection.



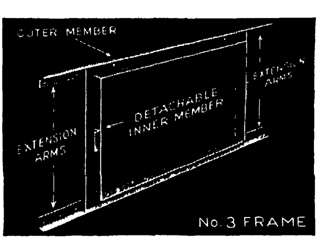


Without Valve

No. 750 Grille (Left) has Sponge Rubber Gasket and $\frac{3}{16}$ in. turndown. No. 757 Intake (Right) has $\frac{7}{8}$ in. projection.



FOUR TYPES OF INSTALLATION FRAMES AVAILABLE



No. 75 Design items can be used with or without installation frames. No. 3 Sidewall Stud Frame (illustrated), fastens directly to stud, forming a solid, streak-proof foundation for register. No. 8 Frame is similar for baseboard use. No. 5 Baseboard Stack Frame provides inexpensive, streak-proof installation. No. 2 Band Iron Frame provides for connecting register to stackhead.

CATALOG 42 showing the complete H & C line, available upon request.

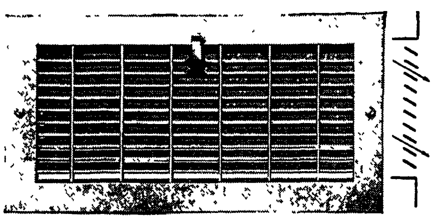
The Independent Register Co.

ESTABLISHED 1898

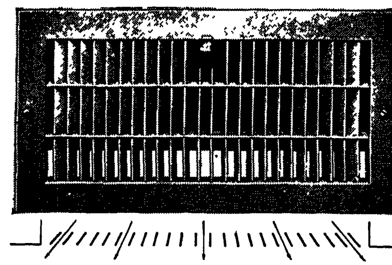
3747 East 93rd Street, **Cleveland, Ohio**

AIR CONDITIONING REGISTERS AND GRILLES

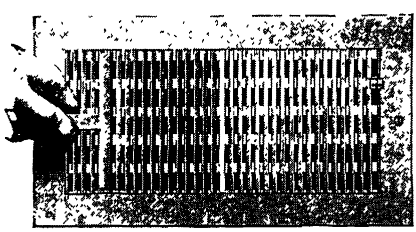
A Complete Line for Either Residential or Commercial Installations



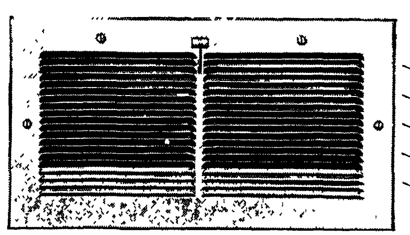
No. 311A "Fabrikated"—Grille Bars individually adjustable for upward or downward directed air flow.



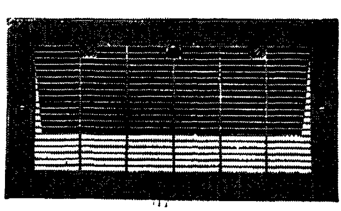
No. 321A "Fabrikated"—Grille bars individually adjustable for right or left directed air flow.



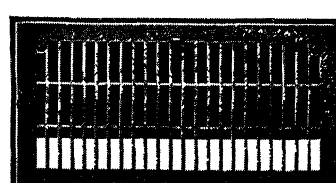
No. 238 Wrought Steel—4-way adjustable direction of air flow. Flexible vertical grille bars, multiple valves.



No. 139 Wrought Steel—Flexible horizontal grille bars, bendable for up, down or straight air flow. Single valves.



No. 136 Wrought Steel—Of fine appearance. Can be used to advantage on low priced installations. Single valves.



No. 137 Wrought Steel—A popular design, moderately priced. Single valves.

We manufacture many other types and styles of Registers and Grilles; a complete line.

You should have the Independent Register Catalogues—Yours for the Asking.

REGISTER & GRILLE MFG. CO.

Incorporated

70 Berry Street, **Brooklyn, N. Y.**

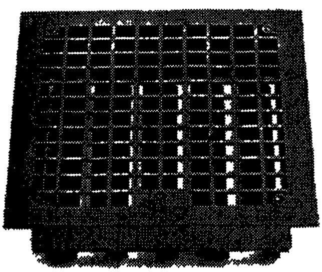
Headquarters for all types of Registers and Grilles

RESIDENTIAL AND COMMERCIAL

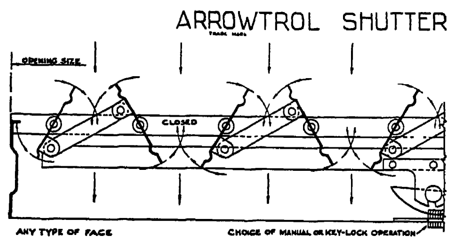
Register shutters of different types can be furnished with all types of Register Faces or Grilles.

All Register Shutters have our exclusive feature of brass collars inserted in the ends of the shutter to minimize rusting.

REGISTERS FOR VENTILATION

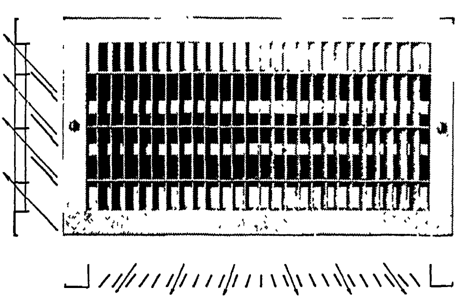


Style 3370 lock type Register allows directional flow of 135 deg either right and left or up and down. Will open 45 deg beyond 90 deg



The Arrowtrol, line cut shown above, gives straight throw in connection with volume control

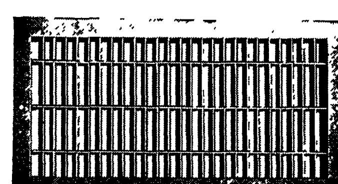
FOUR-WAY DEFLECTION TO AIR FLOWS



Style 3320 Grille and HMV deflecting vane

Front bars vertically adjustable, rear vanes horizontally adjustable; *or* Front bars horizontally adjustable, rear vanes vertically adjustable.

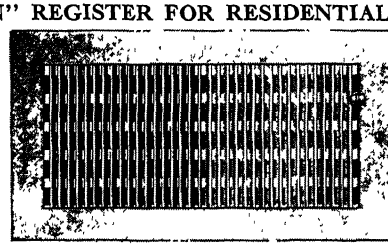
R & G ADJUSTABLE DIRECTED AIR FLOW TWO-WAY DEFLECTION



Use No. 3320 Grille for adjustable right and left deflection. Style 3310 has horizontal adjustable bars for up and down deflection.

THE "THIN MAN" REGISTER FOR RESIDENTIAL USE

Style 1108, shown, allows right and left deflection and up or down control at the back.



Other designs of faces are available.

Ask for our catalog which shows other types of air controls; also 81 different Stamped Metal designs and over 100 designs in Cast Metals—Iron, Brass or Bronze.

Tuttle & Bailey, Inc.

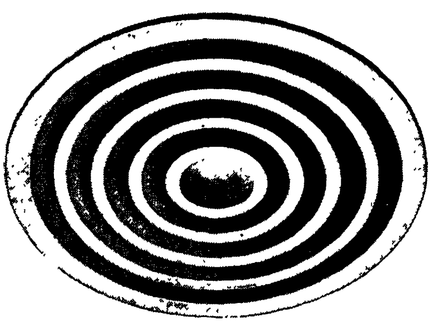
New Britain, Conn.

Branch Offices: New York, Chicago, Philadelphia, Houston

Ceiling Diffusers
Grilles, Registers and Intakes
Air Control Devices

Ornamental Grilles
Cast or Wrought Metals
Copper Convection Heaters

AEROFUSE OUTLET A Truly Flush Type Ceiling Diffuser

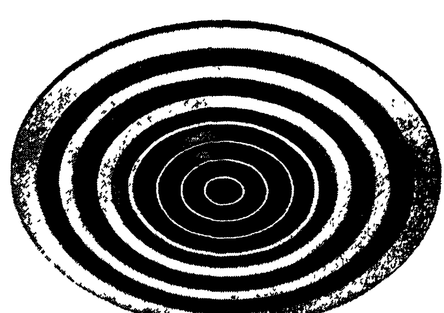


The Aerofuse Outlet is a perfected combination of real beauty and functional efficiency. Flush with the ceiling line and of simple attractive design it harmonizes unobtrusively with any style of interior decoration. Most important, however, is its superb performance. Because of unique construction, the supply air is brought into contact with room air over the largest possible area immediately after leaving the outlet. This results in a high rate of temperature equalization and eliminates the possibility of drafts.

1 Efficient Air Mixture . . . 2 Rapid Temperature Equalization . . . 3 Complete Air Distribution . . . 4 Total Elimination of Drafts

COMBINATION SUPPLY AND RETURN UNIT

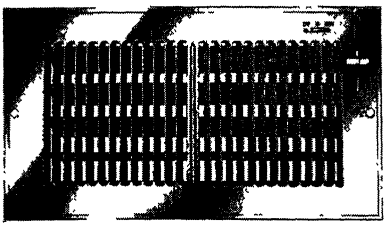
Particularly useful in installations where simplification of the duct layout is of primary importance, since the return (exhaust) duct may be run to the same point as the supply duct instead of to a grille at some other location. The removable center section provides Free Area for the return or exhaust of at least 60 per cent of the supply volume.



THE AEROFUSE OUTLET GIVES YOU BEAUTY
PLUS FUNCTIONAL EFFICIENCY

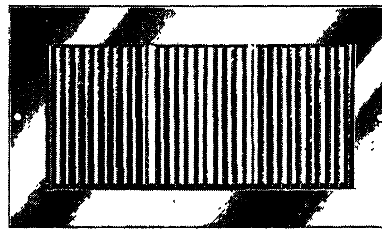
Tuttle & Bailey, Inc.

New Britain, Conn.



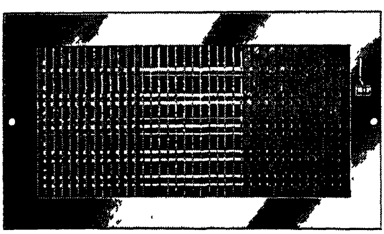
ADJUSTIBLADE REGISTER

A very inexpensive register. The air flow may be deflected sideways by the individually adjustable face vanes, and up and down by back blades. Also available with Flexair design (sectionally adjustable) face.



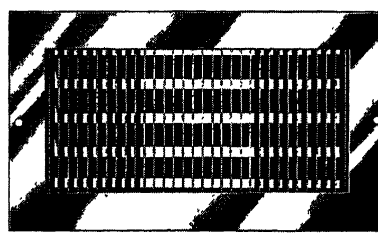
AIR CONDITIONING GRILLES

Furnished in a fixed deflection (Airline design) and a sectionally adjustable (Flexair design) type with bars running either vertically or horizontally.



DOUBLE CORE OUTLET

The vertical front bars are sectionally adjustable for sideways deflection. The horizontal back bars are either sectionally adjustable (Flexair design) or are of fixed deflection (Airline design).

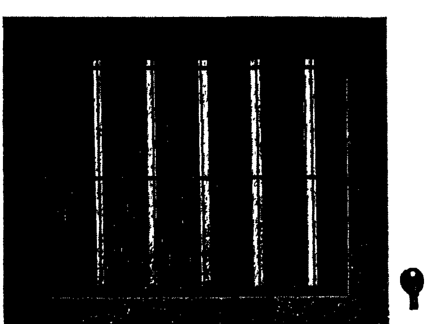


DOUBLE DEFLECTION OUTLET

Made with front bars of fixed deflection (Airline design) or with front bars sectionally adjustable (Flexair design) and individually adjustable back blades.

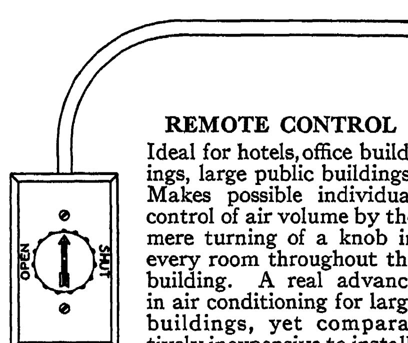
SANTROLS

A simple device to provide positive control of air volume throughout an entire duct system and insure even distribution of air over entire outlet face.



McKNIGHT REGISTER

Provides positive control of air volume at the outlet. Scientifically designed volume control louvers are operated by means of a special key furnished with each register.



REMOTE CONTROL

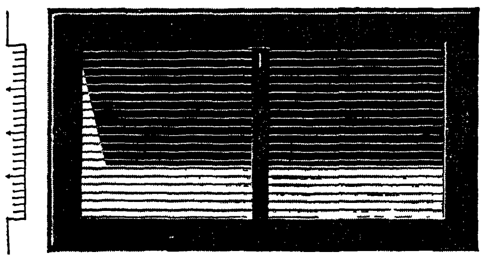
Ideal for hotels, office buildings, large public buildings. Makes possible individual control of air volume by the mere turning of a knob in every room throughout the building. A real advance in air conditioning for large buildings, yet comparatively inexpensive to install.

United States Register Company

General Offices: **Battle Creek, Mich., U.S.A.**

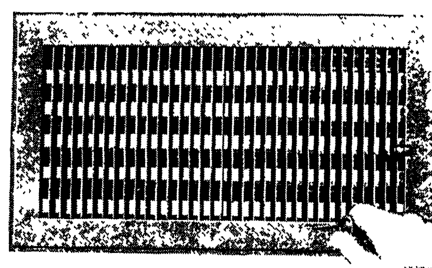
Branches: MINNEAPOLIS, MINN , KANSAS CITY, MO , ALBANY, N. Y., NEW YORK. N. Y , SAN FRANCISCO, CALIF.

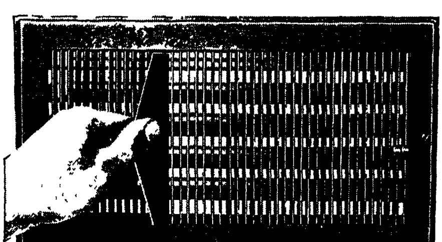
Air Conditioning Registers, Vents and Grilles



Style 153LF—Louver-Type Air Conditioning Register-Bars ¼ in. deep—Spaced 4 openings to the inch affords Non-Vision. Can be supplied in Directional Flow in either Horizontal or Vertical Bar Styles. Can be furnished with all styles of Setting Frames and with INSET PANELS which conveniently afford Multi-Flow.

Style 249LF—Duo-Deflection Air Conditioning Register. Gives complete Air Control. Vertical Front bars—Key-pin adjusted to provide 45 deg Right and Left or Two-way Side Flow. Lever operated Horizontal Back-valves give from Full Closed to any degree of Up-flow and to 45 deg Down-flow. FULL FACE COVERAGE. Can be supplied with any style of Setting Frame. Fits all Stack Heads.



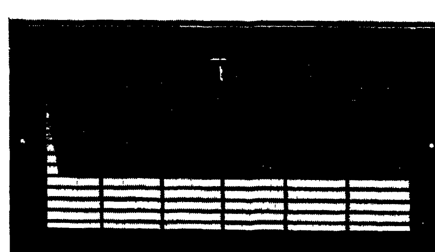


Style 256LF—Flex-bar Air Conditioning Register. Vertical Front Bars set 22 deg Right and Left. Side Flow Deflection attained by setting of Grille Bars with bending wrench to accommodate room condition. Back-valves give same Up and Down control of air flow as 249LF above. FULL FACE COVERAGE. Can be supplied with any style of Setting Frame. Fits all Stack Heads.

All of above Styles can be supplied with either Lever or Individually adjusted Multiple Valves or Louvers. I. E. 153VVI—Vertical Valves Individually adjusted. 145VVL—Lever operated Vertical Valves.

Grilles and Vents in Matching designs are available.

For Complete Information Write for Latest Catalogs.



Style 103LF—Horizontal Lattice Perforated Register for Forced Air Systems. Not directional flow.

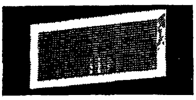
In Canada, United States registers, vents and grilles are manufactured and distributed by the CANADA REGISTER & GRILLE CO., Ltd., Toronto, Ontario

Waterloo Register Company

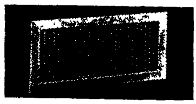
Waterloo, Iowa — **Seattle, Wash.**

Incorporated 1902

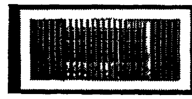
Representatives in Principal Cities



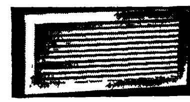
Supply Grille L-1A suitable for floor or window-bench application. Fine mesh grille surface with fixed air deflection. Individually adjustable rear louvres.



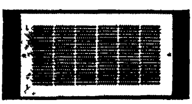
Supply Grille S-2A with individually adjustable streamlined louvres on ½ inch spacing.



Supply Grille S-2A complete with H-MLS Duo-trol multi-louvre damper.

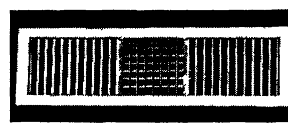


Return Grille L-2P with 3V vision-proof fixed fin arrangement.

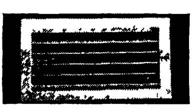


R-600 Forced Air Register provides attractive low cost design for housing projects. No directional features.

Zeph-O-Cone Marine Air Diffuser diffuses high entrance velocities rapidly and quietly. Sizes for 100, 150, 200 and 300 cfm at 2000 fpm inlet velocity.

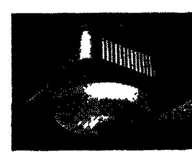


Supply Grille FGV-75 with both front and rear louvres streamlined and individually adjustable. ¾ in. blade spacing.

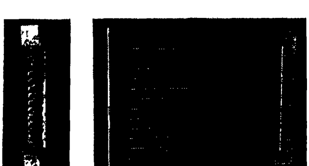


FH-100 Forced Air Register for residential application. Simple operation, quick shut-off. Easy to clean.

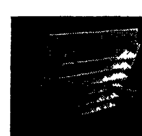
Techni-trol Air Volume Damper all louvres operate by linkage arrangement to reduce air volume without changing direction. Lever or key controls.



Multi-Plak Ceiling Outlet provides four-way direction control from each of four streamlined grille outlets. Light fixtures available on most models.



No-Ve-U Door Ventilator 19 gauge V-shape louvres provide strength and are sight-proof. Either stationary or adjustable louvres. Specify door thickness 1⅛ in., 1⅜ in. or 1¾ in. (at right).



Weather-Proof Louvre with replaceable screen, size 8 x 10, recommended for ventilation of attic and spaces under porches or buildings.

Engineering Data for horizontal diffusion of cool air is available for selection of proper velocities, number and size of outlets.

All products made of steel receive Parker "Bonderizing" process prior to painting.

The American Rolling Mill Company

Executive Offices, **Middletown, Ohio**

ATLANTA, GA.
1437 Citizens & Southern Natl. Bank Bldg.
BERKELEY, CALIF. ....Seventh and Parker Sts.
BOSTON, MASS. ....201 Devonshire St.
BUFFALO, N. Y. ....504 Seventeen Court St. Bldg.
CINCINNATI, OHIO......24 Cooper Bldg., Hyde Park
CHATTANOOGA, TENN., 712 Chattanooga Bank Bldg.
CHICAGO, ILL. ....310 S. Michigan Bldg.
CLEVELAND, OHIO.............. 1516 B. F. Keith Bldg.
COLUMBUS, OHIO.........................1020 Atlas Bldg.
DALLAS, TEXAS.....................1111 Santa Fe Bldg.
DAYTON, OHIO....................506 Mutual Home Bldg.
DES MOINES, IOWA...............703 Old Colony Bldg.
DETROIT, MICH.........5-261 General Motors Bldg.
INDIANAPOLIS, IND...........1106 Fletcher Trust Bldg.
KANSAS CITY, MO.............................7100 Roberts St.
LOS ANGELES, CALIF................331 Petroleum Bldg.
MIDDLETOWN, OHIO.................................703 Curtis St.
MILWAUKEE, WIS.,
627 First Wisconsin National Bank Bldg.
MINNEAPOLIS, MINN....................171-27th Ave., S. E.
NEW ORLEANS, LA.................3501 S. Carrollton Ave.
NEW YORK, N. Y......................................120 Broadway
PHILADELPHIA, PA.........1808 Lincoln-Liberty Bldg.
PITTSBURGH, PA..................................1627 Oliver Bldg.
RICHMOND, VA.......1308 State-Planters Bank Bldg.
ST. LOUIS, MO...........................1725 Ambassador Bldg.

Choose the Correct ARMCO Grade

These grades of ARMCO sheet metal are recommended for the air conditioning applications shown. For detailed information get in touch with the nearest district office or write direct to The American Rolling Mill Company, Middletown, Ohio.

ARMCO Ingot Iron
(Galvanized)

Ducts
Washer Chambers
Plenum Chambers
Steam Line Casings
Furnace Casings
Spray Towers
Drip Pans
Housings
Machine Guards
Unit Conditioners
Roof Ventilators
Eliminator Blades

ARMCO PAINTGRIP
(Galvanized)

Recommended for all galvanized sheet applications where the immediate beauty and extra protection of paint is desired. Write us for a *free* scratch sample.

Hot Rolled
(Sheets and Strip)

Fan Blades
Blower Casings
Fuel Oil Tanks
Unit Conditioners
Stoker Hoppers

ARMCO ZINCGRIP

A special galvanized sheet that can be severely formed without peeling or flaking of the tightly adherent zinc coating.

Cold Rolled
(Sheets and Strip)

Furnace Casings
Room Unit Casings

Plates
(ARMCO Ingot Iron)

Smoke Stacks
Coal Hoppers
Breeching
Unfired Pressure Vessels
Low-fired Boilers
Tanks

ARMCO High Tensile

A low alloy, high tensile steel possessing great strength. Used with proper design it results in weight reduction of framework, tanks and similar items. Under atmospheric service conditions it has four to six times the endurance of regular steel.

Stainless Steel
(Sheets, Strip and Plate)

Combustion Chambers
Heat Flues and Tubes
Humidifier Pans
Pre-heaters
Furnace Parts and Supports
Fan and Blower Blades

Special grades have excellent resistance to destructive heat-scaling up to 2000 F.

Other ARMCO Products

The grades for these applications are only a few that ARMCO makes. Others include copper-bearing sheets and plates and open-hearth steel, either galvanized or uncoated.

Bethlehem Steel Company

General Offices: Bethlehem, Pa.

BETHLEHEM STEEL COMPANY, GENERAL OFFICES: BETHLEHEM, PA. DISTRICT OFFICES: AKRON, ALBANY, ATLANTA, BALTIMORE, BOSTON, BUFFALO, CHATTANOOGA, CHICAGO, CINCINNATI, CLEVELAND, COLUMBUS, DALLAS, DETROIT, HONOLULU, HOUSTON, INDIANAPOLIS, JOHNSTOWN, PA., KANSAS CITY, MO., LOS ANGELES, MILWAUKEE, NEW HAVEN, NEW ORLEANS, NEW YORK, PHILADELPHIA, PITTSBURGH, PORTLAND, ORE., ST. LOUIS, ST. PAUL, SALT LAKE CITY, SAN ANTONIO, SAN FRANCISCO, SAVANNAH, SEATTLE, SPRINGFIELD, MASS., SYRACUSE, TOLEDO, TULSA, WASHINGTON, WILKES-BARRE, YORK. EXPORT DISTRIBUTORS: BETHLEHEM STEEL EXPORT CORPORATION, NEW YORK.

COPPER-BEARING BETH-CU-LOY FOR RUST RESISTANCE

The charts at the right show conclusively the superior rust resistance of copper-bearing steel. Sheets of the same composition as Beth-Cu-Loy, Bethlehem's copper-bearing steel, outlasted ordinary iron and steel by a wide margin when exposed to atmospheric corrosion.

Beth-Cu-Loy, available in the form of sheets, pipe and plates, offers 2 to 3 times longer life as indicated by these three corrosion tests—and Beth-Cu-Loy costs only 3 to 5 per cent more than ordinary steel, much less than open-hearth or copper-bearing iron.

Heating, ventilating and air conditioning engineers, architects and contractors are finding it pays to specify Beth-Cu-Loy wherever moisture or corrosion is a factor. Beth-Cu-Loy sheets are easily workable, durable and low in cost.

BETHLEHEM MAKES:

Sheet Steel—all grades, hot-rolled (black), cold-rolled, and galvanized—available in Beth-Cu-Loy.

Steel Pipe—In sizes up to 3 inches, now made by the Continuous-Weld Process and sold under the trade name Beth-Co-Weld.

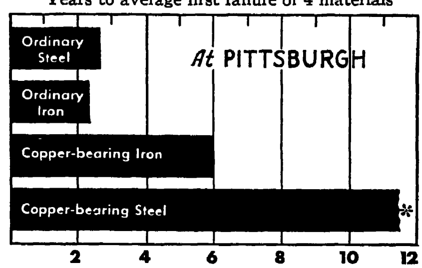
Ammonoduct—A new kind of pipe that has an outstanding advantage in its unusual ductility. It can be bent cold, without need for annealing, without danger of fracturing. Recommended for ammonia piping and for heater coils, water legs in furnaces and similar uses where pipe must be bent.

Boiler Tubes—Charcoal iron and steel.

Plates—all sizes; flanged and dished heads. Available in Beth-Cu-Loy.

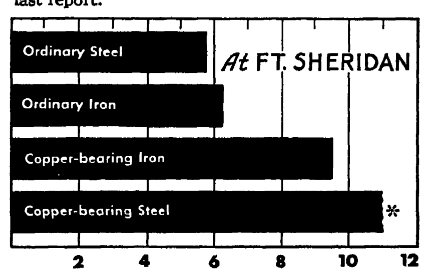
Literature and further information on any of these products can be secured from the nearest district office or from the general offices in Bethlehem, Pa.

A.S.T.M. Test of 22-gage black sheets
Years to average first failure of 4 materials



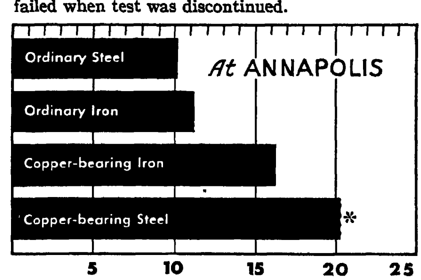
Sheets exposed April 21, 1926; tests still under way (see Proceedings of *A.S.T.M.*—Committee A-5, Vol. 38).

*No failures in copper-bearing steel sheets at last report.



Sheets exposed April 19, 1917; test discontinued April 16, 1928 (see Proceedings of *A.S.T.M.*—Committee A-5, Vol. 28).

*Only 10 of 61 copper-bearing steel sheets had failed when test was discontinued.



Sheets exposed October 17, 1916; tests still under way (see Proceedings of *A.S.T.M.*—Committee A-5, Vol. 38).

*Only 9 of 78 copper-bearing steel sheets had failed at last report.

A booklet "Beth-Cu-Loy Sheets," gives the story of these tests. A copy is yours for the asking.

Jones & Laughlin Steel Corporation

AMERICAN IRON AND STEEL WORKS

Jones & Laughlin Building, **Pittsburgh, Pa.**

WELDED and SEAMLESS STEEL TUBULAR PRODUCTS

J & L Welded Pipe

Jones & Laughlin manufactures Standard Weight, Extra Strong, and Double Extra Strong Welded Pipe, Black and Galvanized, for steam, gas, air, water, refrigeration and sprinkler work. Sizes: ⅛ in. to 16 in. O.D. inclusive.

J & L Copper-bearing Steel Pipe, when specified, can be supplied in standard weight, or extra strong, black or galvanized. Use of this product is recommended for long life, where piping is to be exposed to the atmosphere or other alternate wet and dry conditions.

Jones & Laughlin Steel Pipe is made of soft, weldable steel rolled from solid ingots made to a special analysis. The steel pipe produced is soft and ductile, free cutting, strong at the welds, and free from excess scale. J & L Pipe is commercially straight and free from blisters, cracks or other injurious defects.

Careful attention is given the threading of the pipe with good clean-cut threads fitted with sound couplings correctly tapped to give a tight joint. Soft, ductile steel of free cutting quality enables the contractor to cut clean, sound threads on the job.

The Jones & Laughlin process of galvanizing assures a thorough coating and insures against pipe being clogged with spelter. The galvanized coating adheres strongly and does not tend to flake off.

J & L Seamless Pipe

J & L Seamless Pipe is made in three weights; standard, extra strong and double extra strong. Sizes: ½ in. nominal to 14 in. O.D. inclusive.

J & L Seamless Steel Pipe is pierced from a solid billet—there are no welds. The result is dependable and uniform wall strength. The method of manufacture, and the use of only specially selected steel, assure exceptional ductility, a quality that is essential to successful coiling and bending, and flanging for Van Stone joints.

J & L Seamless Pipe can be used with full satisfaction in either threaded joint or completely welded installations.

Ductility, strength and safety—make this product especially adaptable for air, steam, gas and gasoline lines, boilers, refineries, dry kilns, refrigerating systems and other exacting applications.

J & L Hot Rolled Seamless Steel Boiler Tubes

J & L Seamless Boiler Tubes are manufactured in accordance with the *A.S.M.E.* Boiler Code and comply with the *A.S.T.M.* Specifications and the rules and regulations of the Bureau of Marine Inspection and Navigation of the U. S. Department of Commerce. They are supplied in a full range of standard sizes, from 1 in. O.D. to 6 in. O.D. inclusive.

The process by which Jones & Laughlin manufactures seamless boiler tubes is largely responsible for the unusually high ductility of the product. It is a process in which a forging action is predominant, and produces a desirable combination of strength with a highly ductile nature. J & L tubes therefore are installed with ease and safety.

Other J & L Tubular Products

J & L also manufactures Reamed and Drifted Pipe in sizes 1 in. to 6 in. inclusive, Dry Kiln Pipe, Pipe for Refrigeration Service, Water Well and Irrigation Casing, Line Pipe and a complete line of Oil Country Tubular Products in welded and seamless.

Also J & L Flat Galvanized Sheets for Air Conditioning and Ventilating Work

Pipes, ducts, stacks and trunk lines made of J. & L Flat Galvanized Sheets give a lasting, neat looking job. These sheets have a tight galvanized coating that will not spall or flake off during forming operations. J & L Flat Galvanized Sheets provide uniform resistance to corrosion. The superior quality of J & L Galvanized Sheet Products is assured in every shipment, because J & L's 96-inch Continuous Strip Mill, where J & L Galvanized Sheets are rolled under rigid control, is one of the most modern in the world.

United States Steel Corporation Subsidiaries

Carnegie-Illinois Steel Corporation, Pittsburgh and Chicago

Columbia Steel Company, San Francisco

Tennessee Coal, Iron & Railroad Company, Birmingham

United States Steel Export Company, New York

District Offices in all Principal Cities

U·S·S COPPER STEEL

For Superior Rust Resistance at Low Cost

Corrosion resistance and cost are two determining factors of the type of metal to be used for various air conditioning jobs.

Copper Steel has 2 to 3 times the atmospheric corrosion resistance of plain steel or pure iron as shown in the results of unbiased tests made at Pittsburgh, Ft. Sheridan and Annapolis by the American Society for Testing Metals.

The cost of U·S·S Copper Steel is less than that of pure iron or copper-bearing pure iron and only slightly more than plain steel. Thus there often is a dividend of 200 per cent to 300 per cent longer life and a saving in the first cost as well.

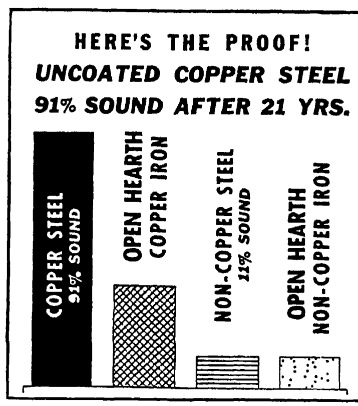
When galvanized, U·S·S Copper Steel produces a sheet that is rust resistant all the way through—not just on the surface. It should be used for all ducts carrying humidified air or placed in damp locations such as basements, shower rooms, etc.

U·S·S PAINTBOND

U·S·S Paintbond should be used whenever galvanized steel is to be painted. This special Bonderized sheet can be painted immediately, offers a much better surface for painting, lessens danger of the paint flaking and retards corrosion. It is used for ductwork, furnace housings and outdoor metal work.

U·S·S DUL-KOTE

U·S·S Dul-Kote is a specially treated non-spangled galvanized sheet which also can be painted immediately without ageing or otherwise preparing the surface. It is available in the South and in the West.



Corrosion test of A.S.T.M. on 22 gage black sheets exposed at Annapolis, Md., October, 1916. The copper steel sheets outlasted all others in the test.

OTHER U·S·S PRODUCTS INCLUDE:

Black Sheets—All grades, hot rolled, cold rolled in a number of different finishes.

Stainless—Heat resisting steel for various uses where temperatures are high and corrosion severe.

Cor-Ten—High Tensile steel—greater strength, greater atmospheric corrosion resistance for smokestacks, hoods, etc.

Efficient performance of the Air Conditioning apparatus and Air System Equipment shown in the two preceding sub-divisions is dependent upon proper adjustment and control. For these purposes there are many types of instruments for testing and adjusting the apparatus, and devices for control of its operation.

The functions and methods of using controls and instruments are described in Chapters 34 and 35 of the Technical Data Section. In the following sub-division of the Catalog Data Section — Controls and Instruments — these devices are illustrated and described, and factual data given by the manufacturers.

Following the data on Controls and Instruments is a sub-division on Steam and Hot Water Heating Systems, including the many parts required to make up the complete systems. This type of apparatus too, requires testing, adjusting, and control; many of the same devices used with air systems are also suitable for use with steam and hot water systems.

CONTROLS AND INSTRUMENTS

Automatic controls form an essential part of modern heating, ventilating and air conditioning equipment, and for the refrigerating equipment which performs important functions in many air conditioning operations. Their use makes possible accurate maintenance of desired physical conditions, with an operating efficiency and economy which are not obtainable with manually operated controls.

Instruments of many types and for many uses are available for determining the capacity and operating efficiency of apparatus. These instruments are designed to obtain results in conformity with adopted test methods and operating standards.

CONTROLS (p. 988-1013)

Thermostats—room, immersion, insertion and surface types; humidity controls, pressure controllers, damper motors, control valves, solenoid valves, relays, etc.

For control of air, gases, temperatures, humidity and liquids; for automatic fuel burning apparatus; for all types of heating, ventilating and air conditioning apparatus operating as separate units, or as integral parts of central systems.

The various types of automatic controls include electric, pnuematic, and self-contained control systems—two-position, or on-and-off, and the modulating or graduated control. They are adaptable for individual room control, or for zone control in large buildings, and also for industrial process control.

Technical data on automatic controls will be found in Chapter 34.

INSTRUMENTS (p. 988-1013)

For measuring, indicating and recording air velocity, temperature, humidity, pressure, flow and liquid levels; and for testing and rating heating, ventilating and air conditioning equipment.

They include gauges, meters, recorders and indicators, hygrometers, pyrometers, psychrometers, thermometers, velometers.

Technical data on instruments is contained in Chapter 35.

Manufacturer's products shown in this division are designed for specific applications. Consult the Index to Modern Equipment for additional products of these manufacturers.

Alco Valve Company

ENGINEERED REFRIGERANT CONTROLS

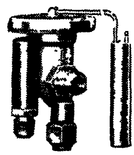
2638 Big Bend Blvd., **St. Louis, Mo.**

NEW YORK OFFICE: 381 Fourth Ave — CHICAGO OFFICE: 433 East Erie St.

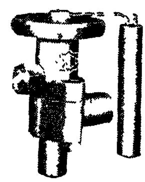
A complete line of Engineered Refrigerant Controls

THERMO EXPANSION VALVES

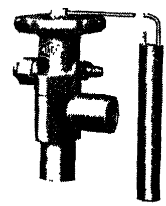
For automatic control of liquid refrigerant on all types of air conditioning and refrigeration systems.



Type TK



Type TJL



Type THL

CAPACITIES—From fractional tonnage to 100 tons Methyl Chloride, 50 tons Freon-12.

MAGNETIC STOP VALVES

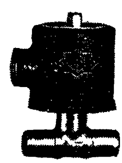
For all types of service

Magnetic Liquid Stop Valves

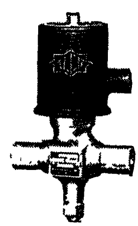
Freon—up to 75 tons, Methyl Chloride—up to 150 tons.

Magnetic Suction Stop Valves

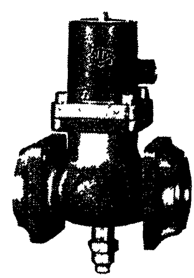
Freon—up to 1½″ or 8.8 tons Methyl Chloride—up to 1½″ or 17 tons



Type S1



Type M8



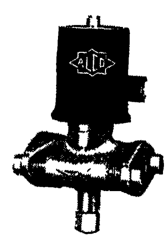
Type R2

AMMONIA CONTROLS

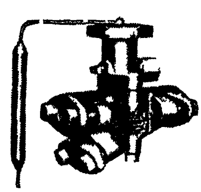
Magnetic Liquid Stop Valves— up to 172 tons.

Magnetic Suction Stop Valves— up to 1½″ or 28 tons.

Thermo Expansion Valves— from fractional tonnage to 60 tons.



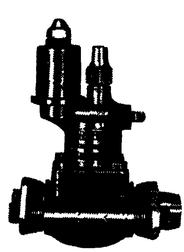
Type M5



Type TGS

EVAPORATOR PRESSURE REGULATORS

For Freon, Methyl Chloride and Ammonia, with port sizes up to 2 in., and a wide variety of connection sizes.



ALCO also offers Magnetic Stop Valves for brine, water, gas, air and steam; specially designed Magnetic Compressor Discharge Valves and Magnetic Pilot Check Suction Stop Valves (for lines subject to reverse Flow).

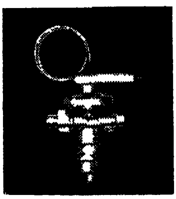
In addition, the Alco line of Engineered Refrigerant Controls includes Float Valves, Float Switches, High Pressure Float Valves, Constant Pressure Expansion Valves and liquid and suction line Filters.

AUTOMATIC PRODUCTS COMPANY

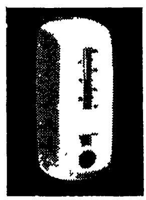
2450 NORTH THIRTY — SECOND STREET
MILWAUKEE WISCONSIN

A-P DEPENDABLE CONTROLS

For Heating, Refrigeration and Air Conditioning

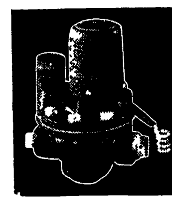


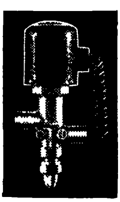
• A-P Thermostatic Expansion Valves. Several models and sizes, for capacities up to 16 tons Freon or 32 tons Methyl Chloride.



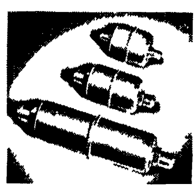
A-P Thermostats.
For Cooling or Heating

A-P Solenoid • Operated Water Valve. Made especially for Deep Well Cooling.



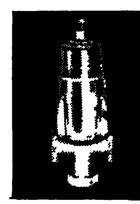


• A-P Solenoid Refrigerant Valves. Capacities up to 50 tons Freon.

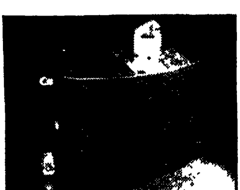


A-P "Trap-It."
Traps dirt, scale, moisture in refrigeration systems.

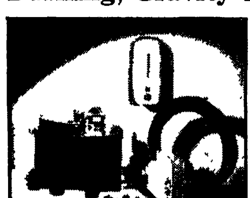
A-P Water • Regulating Valves. Capacities up to 1440 Gallons per hour.



A-P Controls for Oil Burning, Gravity-Feed Heating Plants.



A-P Constant Level Oil Control Valve—With Fuel Compensator. Used on Gravity Oil Burning Appliances.



A-P Complete Furnace Control Set—Made in variety of types for Gravity-Feed Oil Burning Furnaces.

A-P Fuel Oil "Trap-It"—Traps dirt and water in fuel systems. Improves operation of all oil burning devices.

A-P Valve DEPENDABILITY

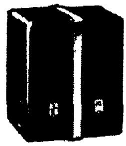
is widely recognized in Refrigeration, Air Conditioning and Heating. This reputation is born of close adherence to a rigid standard of perfection—in materials used, careful testing and inspection, simplicity of construction, and many unusual features.

Barber-Colman Company

Rockford, Illinois

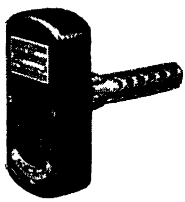
Automatic Control Systems for Heating, Ventilating, Air Conditioning

Room Thermostat

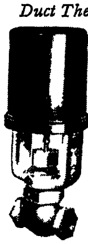


Room Type Microtherm

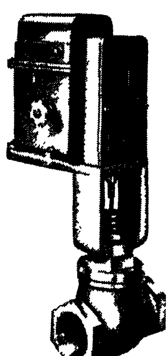
Hygrostat



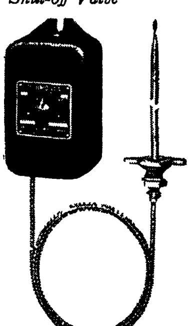
Duct Thermostat



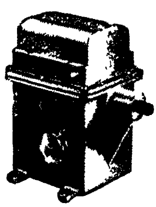
Motor-operated Shut-off Valve



Motor-operated Proportioning Valve



Remote Bulb Thermostat



Proportioning Type Control Motor

Barber-Colman Controls are all electric; precision built to insure long, continuous and dependable service; easy to install in either new or existing buildings; and ready for instant service, even after long shut-down periods.

Thermostats. All types—room, duct, immersion, air stream and remote bulb. For 2-position, floating, and proportioning control.

Hygrostats. Room and duct types.

Motor-Operated Valves. Packless, packed single-seat, pilot piston, V-ported, balanced, 3-way, and butterfly. For 2-position, floating, or proportioning control. Also Solenoid Valves for air, oil, water and gas. *Motor-operated valves are powered with Barcol motors which have only one moving part and require no attention except oiling; oil submerged operators require no attention.*

Control Motors. Uni-directional, or reversible fixed or adjustable speed. For 2-position, floating or proportioning operation of dampers in heating, ventilating or air conditioning applications. *Oil submerged models have the motor and gear train entirely submerged in oil.*

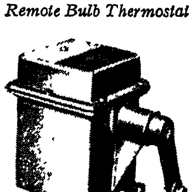
Program Switches. Automatic contact making mechanism for multi-compressor control or similar applications.

Econostat (not illustrated), a complete self-contained thermostatic unit for automatic regulation of the heat supplied to a building in accordance with outdoor temperatures.

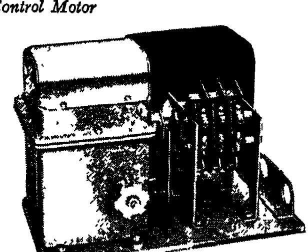
Write for descriptive literature.

LISTED AS STANDARD BY
UNDERWRITERS LABORATORIES

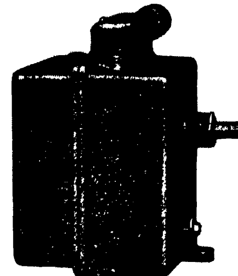
(*See also Page 971*)



Stall Type Control Motor



Program Switch



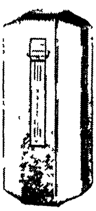
Heavy Duty Industrial Type Control Motor

Julien P. Friez & Sons

(Division of Bendix Aviation Corporation)

Baltimore **Maryland**
Established in 1876

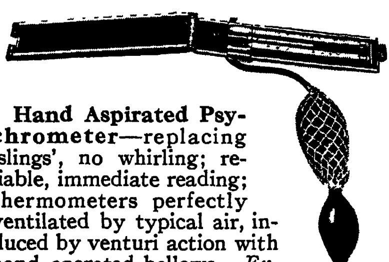
Manufacturers of a Complete Line of Automatic Electric Controls for Industrial and Comfort Applications. Also a Complete Range of Recording and Accurate Measuring Instruments for Indoor and Outdoor Applications



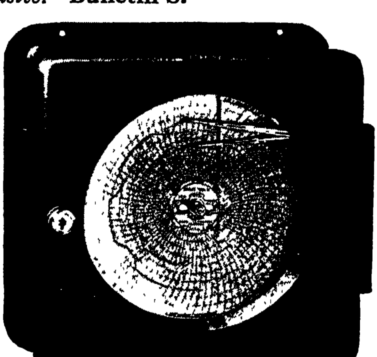
Humidstat—accurate for long periods and over complete range: double length human hair element. **Bulletin AA.**

Thermostat — sensitive, accurate for highest grade work. **Bulletin TT.**

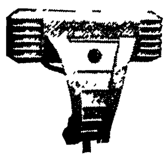
Comfortrol — effective temperature Thermostat resetting itself as prevailing humidity varies, using human hair compensating element. *Exclusive.* **Bulletin E.**



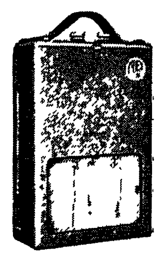
Hand Aspirated Psychrometer—replacing 'slings', no whirling; reliable, immediate reading; thermometers perfectly ventilated by typical air, induced by venturi action with hand operated bellows. *Exclusive.* **Bulletin S.**



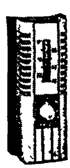
Remote Reading Temperature and Humidity Recorder—Electrically operated; humidity uniquely recorded from distant location directly in percent relative. *Exclusive.* **Bulletin R.**



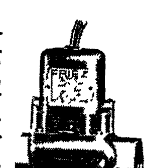
Windowstat— placed at window indoors, positively preventing condensation from excess humidity. Controls humidity supply just below critical point as outdoor temperature varies. *Exclusive.* **Bulletin W.**



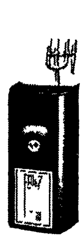
Portable-Recorder— for surveys or tests of humidity and temperature. Inked records on charts the size of filing cards (3″ x 5″). *Exclusive.* **Bulletin G.**



Microstat—Small sized thermostat, featuring powerful Alnico magnets. Though priced with the lowest, unsurpassed for accuracy and fine appearance. **Bulletin TM.**



Magnetic Gas Valve— New principle, free floating disc, no diaphragm; quiet, durable; range of sizes; low priced, low voltage, especially suited for control by Microstat pictured above. *Exclusive.* **Bulletin VG.**



Hydraulic Action

A complete and proven line of controls for warm air, hot water and steam systems, industrial ovens, space heating, air conditioning and refrigeration. Permanently accurate, rugged, at competitive prices. **Bulletins CR, LC and TS** and Data Sheet 225.

Write for Bulletins

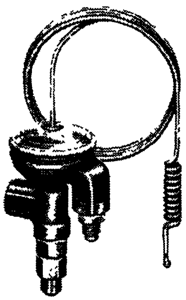
MODERN ADVANCED CONTROLS FOR MODERN NEEDS

Detroit Lubricator Company

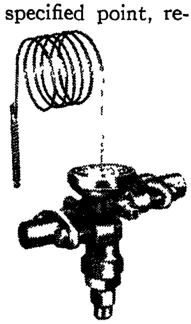
Detroit, Michigan, U. S. A.

New York, N. Y., 40 West 40th Street
Chicago, Ill., 816 S. Michigan Avenue   Los Angeles, Calif., 320 Crocker Street
Canadian Representative:
Railway and Engineering Specialties Limited, Montreal, Toronto, Winnipeg

Dura-Fram Expansion Valves



"Detroit" Expansion Valves are used in the liquid lines to evaporator coils of refrigeration and air conditioning installations. They operate at variable back pressures to keep them completely refrigerated regardless of the variations in load. Power elements are charged with gas at a definite pressure instead of a liquid. By this means the valves remain tightly closed whenever the suction pressure rises above a specified point, regardless of the temperature of the thermostatic bulb. This prevents overloading the motor when starting up a warm system or when operating under excessive temperature conditions. Also provides instant action with no temperature lag. Capacities range from $\frac{1}{8}$ ton to 30 tons with Freon.

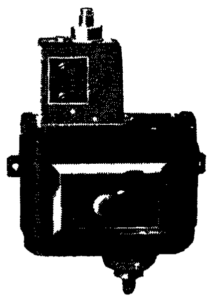


Detroit Solenoid Valves

Detroit Solenoid Valves control refrigerants, water or gas. They are available in the following orifice sizes: $\frac{1}{8}$ in., $\frac{3}{16}$ in., $\frac{7}{32}$ in., $\frac{3}{8}$ in., $\frac{1}{2}$ in., $\frac{5}{8}$ in.—and in capacities up to 17 tons Freon at a 2 lb pressure drop.

Other Controls

This Company can also supply a full line of Boiler and Furnace Limit Controls—Room Thermostats for both heating and cooling — Fan controls — Humidity and Stoker Controls and Convector Valves for concealed radiators.



FIBA

No. 450 Series Refrigeration Controls

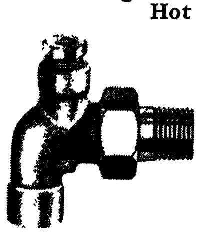
Supplied in several models for various refrigeration and air conditioning requirements. Line voltage type. Model FB-3 controls refrigeration compressor from pressure changes in the suction line. Model FIBA has, in addition, a high pressure cutout, which stops the compressor if high side pressures become excessive. Other models control from temperature changes. Range of either −40 deg. to plus 25 deg. or −5 deg. to plus 60 deg. is available.

Other features include alarm circuit, external "cold control," and high pressure cut-in for meat box applications, which prevents long "off" cycles and guards against "slimy meat" in cold weather.

The New No. 411 Room Thermostat

The "Genuine Detroit" No. 411 is a very sensitive room thermostat for heating and is supplied with or without compensation (pre-heater). With compensation it provides straight line temperature control, eliminates "cold 70" and prevents costly overheating.

Balancing Fittings for Circulated Hot Water

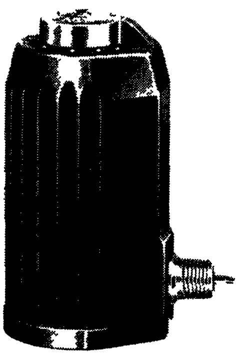


With "Genuine Detroit" Balancing Elbows and Straight Way Fittings on each radiator return the entire system can be accurately balanced after it is all tightened up and in operation—it is done with the turn of a screw driver and without breaking a single joint.

IDEAL FAST-VENTING SYSTEM

For Automatically Fired One-Pipe Steam Jobs

An automatic, oil, gas or coal burner operates on a pronounced *on* and *off* cycle. Therefore, on automatically fired one-pipe steam jobs, all venting and all heating must be accomplished during the limited *on* period—the venting first. The time factor in venting is of utmost importance, especially when using the compensated or "preheater" type thermostat which causes heating cycles even shorter.



New No. 300 Multiport

About four years ago, the Detroit Lubricator Company brought out the Ideal Fast-Venting System which consists of No. 300 Multiport for radiators and the No. 861 Hurivent for mains which have very large ports, and allow both radiators and mains to vent in only a small fraction of the time needed when conventional valves are used.

Experience has demonstrated the outstanding advantages of large port fast venting for automatically fired one-pipe steam systems. The more important of these advantages are:

1. Quicker response to the thermostat's call for heat.
2. System balance—all radiators heat up simultaneously.
3. Definite elimination of cold rooms.
4. Material reduction in fuel costs.

The New 300 Multiport

With the principle of large port fast-venting definitely proven, Detroit engineers have taken the next logical step in the progress of fast-venting. A step that carries this practice to the full limit of its possibilities. They have redesigned the No. 300 Multiport adding several new and very valuable features.

The new No. 300 Arco-Detroit Multiport has a venting capacity even greater than the old No. 300, a greater range of adjustment, and permits even, fast heat delivery as the burner comes on.

Boiler pressure, in nearly all cases, need never exceed a few ounces—still further reducing firing cycle and fuel cost. Recommended only for automatically fired systems controlled to a maximum operating pressure of 3 lb.

Other Important Changes

1. Minimum venting rate is automatically controlled which makes it possible to secure the absolute minimum venting rate for oversized radiators or radiators very near the boiler.
2. A siphon tongue replaces the old siphon tube so that the new valve can be used on thin radiation without the siphon conflicting with the wall opposite the connection.
3. Together with a simple adapter, the new valve can be used on installations which otherwise would require a straight shank valve.
4. Dealers and contractors stock only one model, that illustrated herewith.
5. New Inner-shell construction prevents both noise and spitting.
6. Size reduced and more pleasingly styled.

New No. 861 Hurivent

No. 861 Arco-Detroit Hurivent for Mains

In order to take full advantage of fast-venting, large port vent valves are necessary to vent the mains at the same time air valves are venting risers and radiators. The No. 861 has a full ½ in. port and has for the past several years proved its marked efficiency—the No. 861 will vent more than 260 ft of two-inch main in the short period of 60 seconds at only 4 ounces of pressure.

In addition, the Detroit line includes the new No. 5000 Airid Variport, an adjustable large port valve for installations operating at more than 3 lb. pressure; the No. 500 Airid, an inexpensive non-adjustable air valve; and the No. 841 Ideal Quick Vent for mains. For a vacuum operated hand fired coal job there is the No. 510 Vac-Airid, an air valve, and the No. 862 Vac-Hurivent, and the No. 842 Ideal Vac-Vent, both for the mains.

The Fulton Sylphon Company

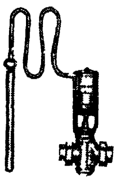
Manufacturers of Sylphon Automatic Temperature Controlling Instruments and Packless Expansion Joints

Sales Representatives in Principal Cities

Knoxville, Tenn.

HOT WATER SUPPLY

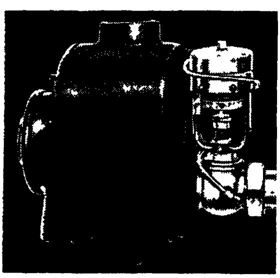
No. 923 Temperature Regulator—For controlling water temperature in heaters, open or closed tanks and other equipment. Operation unaffected by temperature fluctuations at the valve, either above or below bulb temperature. All parts, except steel adjustment spring, made of non-ferrous metals. May be installed in any position. Ranges from 40 −80 F to 290 −330 F. Bulletin HVG-20.



No. 923 Temperature Regulator

Sylphon Thermostatic Water Mixers

Utilize hot water from any storage tank or instantaneous heater, and effectively regulate the amount of cold water required to temper it to the desired degree, actually mixing the hot and cold water together before delivery. Temperature remains constant in spite of fluctuations in supply water temperatures or pressures.

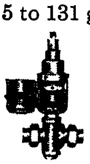


No. 902 Sylphon Thermostatic Water Mixer—14 to 131 gpm depending on water pressure

Four sizes with capacities ranging from 5 to 131 gpm. Bulletin HVG-40.

REFRIGERATION CONTROLS

Adaptable wherever brine is used as the refrigerant. Latest development is a "freeze-proof" valve (illustrated at left on the popular Sylphon No. 945-Z Regulator). Bulletin HVG-20.



No. 945-Z Regulator

PACKLESS EXPANSION JOINTS

The Sylphon Packless Expansion Joint eliminates useless building height, expensive construction and non-revenue producing space. No costly leaks and repairs, no repacking, always tight, allows heating system to operate at full efficiency. Write for Bulletin HVG-140.

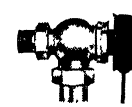
No. 110 Sylphon Expansion Joint

SPACE HEATING CONTROL

No. 885 Automatic Radiator Valve—For exposed radiation. Small, neat, finely finished, adjustable to room temperature desired. Simply replace ordinary radiator valves with these Sylphon Automatic Regulators—no wiring, piping or auxiliary equipment are required. These valves answer the demand for an inexpensive means of providing accurate, dependable space temperature control in rooms, sections or throughout large buildings, new or old. Similar type valves for concealed radiation — get Bulletin HVG-80.

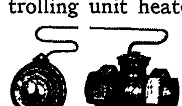
Sylphon No. 885 Automatic Radiator Valve

No. 890 Electric Radiator Control Valve—For either exposed or concealed radiation. Similar in appearance and action to Sylphon Automatic Valves, but operated by an electric wall thermostat. The closing of the thermostat circuit energizes a low voltage electric heater coil surrounding a bulb containing a volatile liquid. This liquid expansion causes pressure on a bellows in the valve head operating the valve. This provides radiator valve control from a remote location, permits regulation of several radiators from a single thermostat, enables a time switch to be installed, if desired, offers effective zone control of large areas at a fraction of the cost of conventional motor-operated valve systems. Bulletin HVG-70.



Sylphon No. 890 Electric Control Valve

No. 7 Temperature Control—A self-contained, self-powered regulator for controlling unit heaters, wall or ceiling type radiators, heating coils in duct-type heating systems, etc. Quickly installed, holds temperatures within close limits. Valve placed in steam line to one or a battery of heaters, thermostat mounted on wall or column. For use on regular heating pressures up to 15 lb. Similar regulators, Nos. 7-2 and 7-3 for 50 and 75 lb pressure and temperatures up to 170 F. Bulletin HVG-50.



Sylphon No. 7 Temperature Control (Self-operating)

HEATING AND AIR CONDITIONING CONTROL

Almost any type of heating, ventilating or air conditioning system can be advantageously controlled wholly or in part by Sylphon Regulators. Basic advantages of Sylphon Controls are:

Modulating—Maintains ideal conditions—not continually correcting too hot, too cold, too humid or too dry conditions.

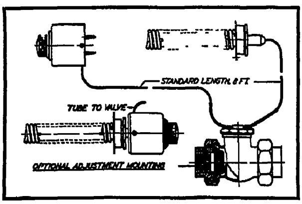
Compensating—Many Sylphon Regulators offer compensating control, automatically raising their low limit setting at a predetermined rate as outside temperatures fall.

Sensitive—Close operating temperature differentials. Quick response.

Simple—in design.

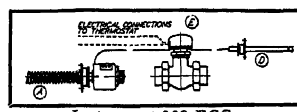
Rugged Construction—To give years of satisfactory service.

Adaptable—Any one of many combinations of Sylphon Instruments can be arranged to control any air conditioning system and to provide exactly the conditions desired. *Write for Bulletin SAC—820.*



Regulator 928-C

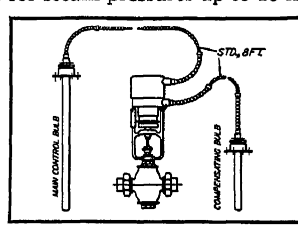
The No. 928-C Regulator—Simple, compact yet highly sensitive. Suitable for modulating control of air temperatures in ducts. Bulb is constructed of numerous coils of copper tubing giving sensitivity to the slightest temperature variation. Packless valve eliminates service problem and makes this regulator ideal for installation in inaccessible locations. Suitable for steam pressures up to 15 lb; other types available for pressures up to 75 lb.



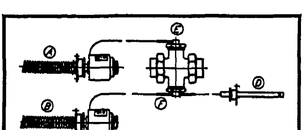
Instrument 928-ECC

The No. 928-ECC Sylphon Instrument—Room control and low-limit control in a single valve regulator for modulating control of ventilating systems. Main control from an electric room thermostat operating through the electric head "E" on the valve. Low-limit control by Bulb "A" located in discharge duct from the heater. Bulb "D", located in inlet side of the duct to the heater, compensates Bulb "A". Compensating thermostat can be furnished to raise low-limit setting at predetermined rate with falling outside temperature. Suitable for steam pressures up to 15 lb.

Sylphon No. 971 Differential Regulator—For controlling room temperature on the cooling cycle, where chilled water or brine is used as cooling medium and where it is desired to have a gradual increase in room temperature as outside temperature increases. This regulator is modulating in action, thereby affords better control over humidity than is procured when usual on-and-off type control is employed.



Regulator 971

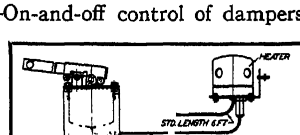


Control 889-C

The Sylphon No. 889-C Control—A modulating, dual-function regulator for control of duct heating and ventilating systems—two independent valves in a single body.

Adjustable Thermostat "A" governing Valve "E" functions to maintain room temperature from temperature of recirculated air. Adjustable Thermostat "B" acts as a low-limit ductstat controlling Valve "F" to maintain minimum discharge air temperature. Bulb "D" compensates Bulb "B" to maintain even discharge air temperature irrespective of demand. Compensated Thermostat "B" can also be furnished to raise its setting at a predetermined rate with falling fresh air temperatures, if desired. Suitable for steam pressures up to 15 lb.

Sylphon No. 371 Positive Type Damper Motor—On-and-off control of dampers. Operation controlled by room thermostat, by hand-operated switch, by motor starting switch, etc. Advantages include: (a) motor returns to closed or safety position in event of current failure; (b) heat-motor bulb and motor separate enhances convenience of installation; (c) damper motor lever adjustable; (d) positive, powerful operation.



Damper Motor 371

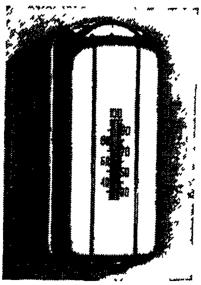
GENERAL ELECTRIC COMPANY

SCHENECTADY, N. Y.

Sales Offices, Warehouses, Service Shops and Distributors in Principal Cities

For Code Wire, Conduit Products, Wiring Devices, Insulating Materials, etc., Address—APPLIANCE AND MERCHANDISE DEPARTMENT, BRIDGEPORT, CONN.

GENERAL ELECTRIC CONDITIONING CONTROLS



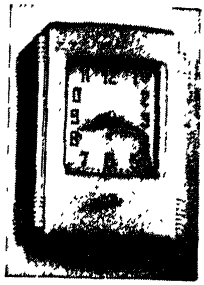
G-E ROOM THERMOSTAT . . . CR7865-Z

The CR7865-Z group of instruments offers a complete line of room thermostats for heating and cooling service. These high-grade instruments, embodying all modern features for comfort or process temperature control, have been styled to present a pleasing appearance in any surrounding.

For additional information ask for Bulletin 8951, Page 101.

APPLICATIONS· 1. Control of domestic or commercial heating equipment, when used with suitable relays or combustion controls. 2. Control of cooling equipment, when used with suitable relays or magnetic switches. 3. Control of year 'round air-conditioning systems, when used with CR7865-C5 cycling relay. 4. Control of motor valves, damper motors, etc.

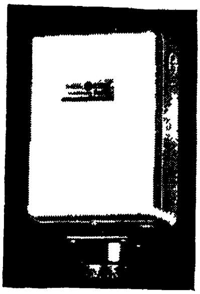
G-E TIME SWITCH . . . TYPE T-41 CR7865-X



The Type T-41 time switch, when used with a thermostat that has thermal night setback, provides day-night temperature control for domestic heating systems. The Telechron driven time mechanism operates a single-pole, single-throw contact which energizes the night-setback resistor during the low-temperature interval.

For additional information ask for Bulletin 8956, Page 101.

APPLICATIONS: 1. Day-night temperature control, in conjunction with room thermostat having thermal setback feature (CR7865-Z). 2. Time-switch operation of unit coolers, unit heaters, etc. 3. In conjunction with room-thermostat limit controls and relay, as a complete stoker-control system.



G-E SELECTOR CONTROL . . . CR7865-C5

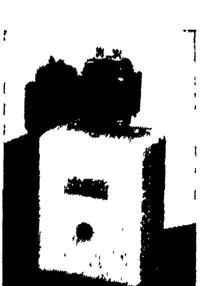
Heating-cooling Relay

The CR7865-C5 selector-control unit, when used with a three-wire, "floating"-type thermostat or humidistat, allows close control of temperature, year 'round, or humdity under widely varying conditions. While the most widespread application of this control is in process air conditioning, this control is well suited to general air-conditioning applications.

For additional information ask for Bulletin 8952, Page 109.

APPLICATIONS· 1 Heating and cooling control for maintaining constant temperature in process air conditioning. 2. Humidification and dehumidification control in process air conditioning. 3. Automatic year 'round comfort control.

G-E MAGNETIC SWITCH WITH TRANSFORMER . . . CR-7865-C3A



The CR7865-C3A is a magnetic switch for control of air-conditioning-equipment motors directly from the contacts of a room thermostat or other low-voltage control instrument. It consists of a sturdy three-pole magnetic-switch mechanism, overload relays, and a low-voltage control transformer (24 volts, open circuit) assembled as a unit.

For additional information ask for Bulletin 8962, Page 101.

APPLICATIONS: 1. Control of condensing-unit motor from room thermostat in air-conditioning installations. 2. Control of cooling-tower or evaporative-condenser motors from room thermostat; compressor motor controlled through auxiliary contact. 3. Control of fan motor from room thermostat on constant-steam-supply heating systems. 4. Control of electric heating units from remote low-voltage thermostat 5. Control of water-pump motors from sensitive water-level or pressure-control devices.

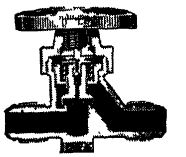
(*See also Pages 966-967*)

Henry Valve Company

1001-19 North Spaulding Ave., Chicago, Ill.

Manufacturers of

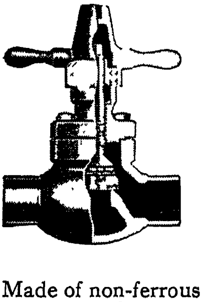
Complete Line of Valves, Strainers and Driers for Freon, Methyl Chloride and Sulphur Dioxide. Also Ammonia Valves and Forged Steel Fittings. These products are widely used by many branches of the government.



Balanced-Action Diaphragm Packless Valves

VALUE OF BALANCED-ACTION
Regardless of operating conditions or the differential in the pressure above and below the valve seat, balanced-action assures positive and instantaneous opening. The balancing action is really the equalizing of the pressures on the two sides of the valve seat at the instant of opening. This is accomplished by a channel in the axis of the valve stem. When the valve is closed, the upper port of this channel is sealed by the diaphragm assembly itself. At the instant of opening, the pressure above the seat forces the diaphragm assembly upward, exposing the upper port of the balancing channel. The pressure is released through the channel to the region below the seat, equalizing the pressures, thus assuring positive opening. A spring-tensioned ball check seals the channel for diaphragm inspection.

Other important features are Ovaline handwheel, ports-in-line, non-rotating bearing plate to protect diaphragm from rotating friction of stem, and use of multiple puncture and fracture-proof diaphragms designed to resist wear and corrosion. Available in a complete range of sizes with flare and solder connections.

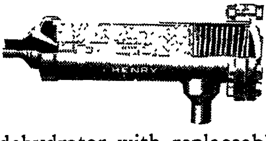


WING CAP VALVES
Designed especially for Freon and Methyl Chloride. Have patented rotating self-aligning stem disc. Special resilient packing. May be repacked under pressure. Wing cap can be inverted and socket used for operating valve. Made of non-ferrous alloy to meet government specifications. Solder connections machined directly in valve body.

ABSO-DRY PRESSURE SEALED DRYERS For Refrigeration and Air Conditioning
The exclusive Henry vacuum process first removes every trace of moisture, then the dryer is charged with dehydrated air. Loosening a seal cap prior to installation produces hissing sound, a guarantee of original factory dryness and freedom from leaks

OTHER FEATURES OF HENRY DRYERS—*Perforated Dispersion tube* is connected to inlet port and exposes entire volume of dehydrant to penetration by refrigerant. *Minimum* pressure drop. No channelling. *Compression Spring* maintains uniform tension on dehydrant at all times and compensates for changes in volume. *Soldered or Flanged Shells*—models are available with either soldered cap or flanged end shells. Flange is distortion-proof. Shells not exceeding 5½ in. in length are drawn in dies, so that they have only one joint.

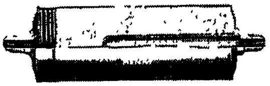
TWO DEHYDRANTS—Choice of the two most popular dehydrants at same price: Activated Alumina or Silica gel.



Type 757 Cartridge Dehydrator

A flanged shell dehydrator with replaceable cartridge.

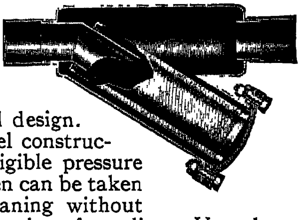
Type 712 Dehydrator



Soldered brass shell dehydrator with dispersion tube.

HENRY STRAINERS
There is a size and type of Henry Strainer for every installation requirement.

Type 895 "Y" Strainer



With solder fittings for use with copper pipe. Exceptional design. Welded steel construction. Negligible pressure drop. Screen can be taken out for cleaning without removing strainer from line. Very large screen area. Light weight. Baffle prevents heavy particles injuring screen.

Johnson Service Company

AUTOMATIC TEMPERATURE AND AIR CONDITIONING CONTROL

General Offices and Factory

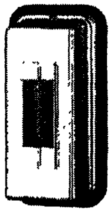
Milwaukee, Wis.

Branch Offices in all Large Cities

JOHNSON TEMPERATURE REGULATING CO. OF CANADA, LTD., 113 SIMCOE ST., TORONTO, ONT.
HALIFAX, N. S. MONTREAL, QUE. WINNIPEG, MAN. CALGARY, ALTA. VANCOUVER, B. C.

PRODUCTS AND SERVICES

Manufacturers, Engineers, and Contractors—For automatic temperature and humidity control systems applied to all types of heating, cooling, ventilating, and air conditioning installations.

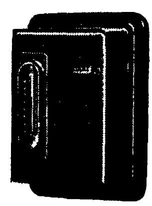


Single Room Thermostat

Space Control—Automatic control of room temperatures and humidities, applied to radiators, unit ventilators, unit heaters, and heat delivery ducts. Johnson "Duo-Stats" to maintain the proper relationship between outdoor and heating system temperatures for groups of radiators, or "heating zones." A complete line of devices for automatic control of air conditioning systems, heating, cooling, humidifying, dehumidifying.

Process Control—Automatic temperature and humidity control devices for manufacturing and industrial processing, applied to tanks, dryers, vats, kettles, curing rooms, coolers, kilns, etc.

Nation-wide Service—Johnson sales engineers, technicians, and trained installation men are available at all branch offices. None of the men in the nation-wide Johnson organization are agents, jobbers, or part-time representatives. All are salaried employees, devoting their entire efforts to the interests of the JOHNSON SERVICE COMPANY and its customers.



Dual Room Thermostat

Send for Bulletins describing the detailed characteristics of any of the Johnson devices.

JOHNSON THERMOSTATS

Room Thermostats—Intermediate (gradual) or positive (snap) action, maintaining temperatures accurately within one degree above or below point of setting. "Dual" (two-temperature) and "Summer-Winter" types, as well as standard instruments. Various types of covers allow wide selection of adjusting features, guards, and method of mounting. Red-reading thermometers with magnifying tube attached to each cover.

Insertion and Immersion Thermostats—Sense temperatures in ducts, tanks, and similar locations. High grade insertion or immersion thermometers for mounting adjacent to the thermostats, including the distinctive Johnson "Sunrise" insertion thermometer, with red-reading mercury column in heavy lens glass tube and 9-in. scale with patented adjustable tilting feature.

Room Humidostat

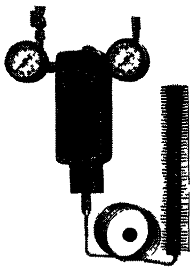
Extended Tube Thermostats—Mercury or vapor tension type, to sense temperatures at a point remote from the location of the operating mechanism. Various types of bulbs. Connecting tubing up to 50 ft in length for vapor tension, 75 ft for mercury actuated systems.

Special Thermostats—For applications encountered in industrial control, including the "Record-O-Stat," combination extended-tube temperature controller and recorder. Full 10-in. chart and vapor tension or mercury actuated systems. Single or duplex type, the latter controlling and recording both wet and dry bulb temperatures.

Remotely Adjusted Thermostats—A distinctive Johnson feature, applied to various types of instruments where readjustment must be accomplished from a remote point, such as another thermostat or a manual switch.

"Sylphon" Radiator Valve

Johnson Sensitivity Adjustment—An important development in automatic temperature and humidity control for air conditioning. A unique and convenient means of adjusting the sensitivity of Johnson thermostats and humidostats, on the job, balancing "time-lag" with respect to the capacity of conditioning apparatus. "Hunting" and temperature fluctuation prevented. Available on all Johnson gradual action insertion and immersion thermostats, insertion humidostats, and certain room type thermostats and humidostats.



Extended Tube Thermostat

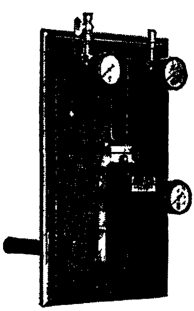
JOHNSON HUMIDITY CONTROL

Johnson Humidostats—Automatically control the supply of moisture delivered by a humidifier or by other means, maintaining a constant percentage of relative humidity. Available in both room and insertion patterns and with various types of elements as determined by requirements, the most sensitive controlling within 1 per cent at relative humidities as high as 95 per cent at 100 F. Humidostatic elements are wood-strip, human hair, animal membrane, or other suitable substances as selected.

Johnson Humidifiers—"Steam grid" type (perforated pipe supplied with low pressure steam) or pan type with copper evaporating pan, brass heating coils, and float control.

JOHNSON VALVES

Johnson Diaphragm Valves—Simple and rugged. Seamless metal bellows and heavy spring operate the valve stem. Available, if desired, with diaphragms of special molded rubber, resistant to aging, heat deterioration and oxidation. No complicated moving parts. Made in all standard sizes and patterns. Direct acting (normally open) or reverse acting (normally closed). Also, three-way mixing and by-pass valves. For steam, water, brine, and freon.



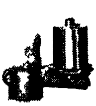
Remotely Adjusted Duct Thermostat

Johnson "Streamline" Diaphragm Valves—Modulating discs and special internal construction, insure superior gradual control . . . Where maximum power is required for repositioning at the slightest demand of controlling instruments, Johnson molded rubber diaphragm valves are fitted with Johnson's dependable pilot feature, for smooth gradual operation, independent of friction and pressure variations.

JOHNSON DAMPERS AND SWITCHES

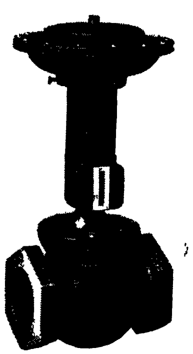
Standard Johnson Dampers—Steel blades in flat steel frames with adequate bracing to form a rigid assembly. Finished in two coats of black lacquer. Special corrosion-resisting finishes on order. Angle iron frames optional. **Special Dampers**—Galvanized iron, monel metal, aluminum, copper, rust-resisting steel, etc. Brass pins in steel bearings or ball bearings.

Johnson Damper Motors—In principle, similar to valves. Seamless metal or specially molded rubber diaphragm operates damper through suitable linkage. Various types of brackets. Distinctive Johnson "Piston-type" damper motors afford long travel at full power, a feature not found in other such devices. With or without pilot mechanism, as described above under "Valves."



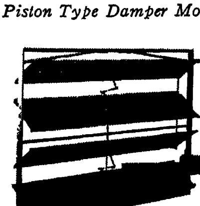
Modulating Attachment for Expansion Valves

Johnson Pneumatic Switches—Various patterns for operation of dampers and for placing thermostats and other devices in and out of service, as required, from remote points. Standard switchboards are oiled slate. Ebony asbestos, polished oak, and genuine or imitation marble on order.

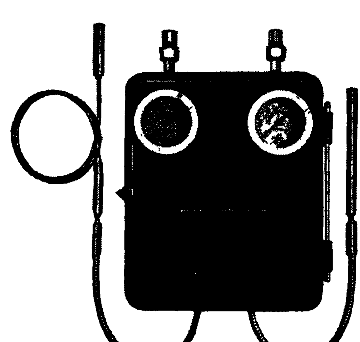


Rubber Diaphragm Coil Valve

Piston Type Damper Motor



Louvered Damper



Johnson "Duo-Stat"

Illinois Testing Laboratories, Inc.

422 N. LaSalle Street, **Chicago, Illinois**

TESTING ENGINEERS AND MANUFACTURERS

Pyrometers—Portable—Wall Type—Surface Temperatures
Distant Reading Resistance Thermometers
Automatic Temperature Controllers—Air Velocity Meters

"ALNOR" VELOMETER

The Only All-Purpose Air Velocity Meter

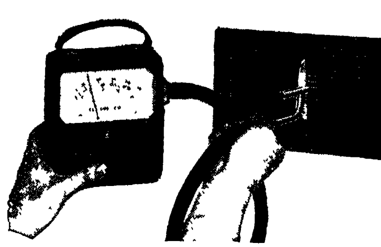
Velometer with averaging jet used for checking velocity from supply grille. Tube is attached to left side.

The Velometer is a versatile direct reading air velocity meter which gives instantaneous readings of the speed of air.

The Velometer can be scaled to not only read velocities directly, but also to read static or total pressures when using suitable jets.

The Velometer is ideal for measuring duct Velocities and pressures, velocities at grilles or registers. It also offers a convenient and satisfactory instrument for checking drafts, leaks around doors or windows or in ducts, velocities of ceiling outlets and similar air diffusers.

Anyone can use the Velometer. No mathematical calculations, no leveling—no timing.



Velometer with new exhaust grille jet. Tube is attached to right side.

Made in standard ranges for velocity readings from 20 fpm to 6000 fpm and 3 in. static or total pressure. Special ranges available as low as 10 fpm and up to 24,000 fpm velocity and 20 in. pressure.

New Intake or Exhaust Grille Jet—To provide more accurate velocity readings at exhaust grilles a new type jet is now offered for use with the Velometer. This jet compensates for the change from static pressure to velocity pressure at or near the face of the grille.

Spot jets—for velocities over very small areas.

Averaging jets—for obtaining average velocities over a definite area or grille face.

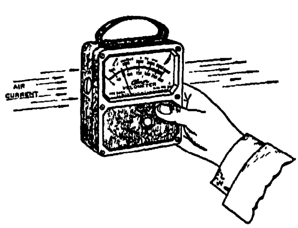
Duct jets—for determining velocities directly within ducts or pipes.

Static pressure jet—for static pressures in inches of water.

Total pressure jet—for total pressure in inches of water.

Other jets—Standard jets offered in several lengths and sizes.

Special jets—can be designed for unusual applications.



View shows how instrument is used to read low velocities without jets.

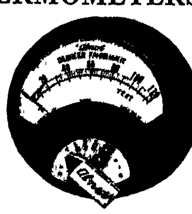
Ask for Bulletin No. 2448-D

"ALNOR" DISTANT READING ELECTRIC THERMOMETERS

The use of "Alnor" multi-point resistance type thermometers is rapidly increasing in air-conditioning, as well as for heating and refrigerator installations.

The instrument can be located in the machinery room or boiler room with the elements located on various floors in any part of the building, or outdoors, thus providing the engineers with constant and convenient temperature readings.

"Alnor" thermometers are made in several styles and sizes, both portable and mounted types.



"Alnor" round type multi-point resistance thermometer with built-in switch.

Ask for Bulletin No. 2451-B

Leeds & Northrup Company

General Office and Works: 4941 Stenton Avenue, **Philadelphia, Pa.**

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SAN FRANCISCO
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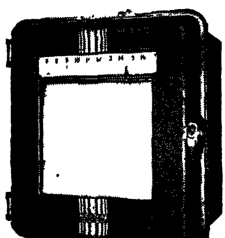
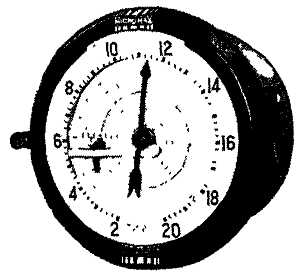
RUGGED, ELECTRICAL-BALANCE INSTRUMENTS



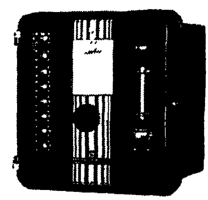
Model S Micromax Recorder

Records from 1 to 16 points on a single strip-chart. Extremely open record. Can operate signals. (About 1/15th size)



Model R Micromax Recorder

Records 1 or 2 points on a round-chart. Has extremely readable dial. Can operate signals. (About 1/15th size)



Panel Indicator

Hand-operated. Can be connected through selector switches to any number of points. (About 1/12th size)

Electrical Thermometers for Air Conditioning

No method for measuring temperatures fits the specific needs of air conditioning as does the three-lead, null-type resistance thermometer method. It is independent of distance and disregards all temperatures except those right at points of measurement. Thermohms (electrical resistance thermometers) can be placed anywhere—in rooms, air ducts or water lines. They are connected by simple electrical wiring to instruments at a central location. Instruments may be: Micromax Recorders, Model S for up to sixteen Thermohms, Model R, for related pairs such as wet and dry bulb; indicators with switches for any number of Thermohms; or indicating and recording combinations.

Sound in principle, this equipment is reliable in operation. Instruments and Thermohms are highly responsive, yet rugged in construction. A complete system is easy and economical to install, regardless of distances. It is easy to operate and demands minimum maintenance. Thermohms and instruments are interchangeable, and can be replaced without disturbing wiring or returning anything to the factory.

L&N Resistance Thermometers make it possible to operate efficiently; to maintain comfort or correct process atmosphere constantly . . . so that maximum return is realized on the conditioning investment.

Jrl Ad-N-225(2)

Electrical Instruments for the Steam Plant

The facts needed to operate a modern heating plant so as to save fuel, to protect equipment, and to operate efficiently at varying loads are provided reliably by rugged L&N instruments. Readings can be indicated or recorded or both. Recorders can be equipped to operate signals or alarms that warn the operator of extreme conditions. In some cases the instruments control automatically.

Micromax Model S provides a permanent record of conditions at from 1 to 16 points on one wide-scale chart. Micromax Model R concentrates on conditions at one point, provides a permanent record, has a giant indicating dial that can be read at a glance. A Panel Indicator provides intermittent checks on conditions at one or several points.

In the heating plant, L&N measuring, signalling or controlling equipment is used for:

Metermax Combustion Control
Furnace Pressure Control
Smoke Density Analysis
Flue Gas Analysis (Per Cent CO_2)
Flue Gas Temperatures
Steam and Water Temperatures
Boiler-Furnace Temperatures
Condenser Leakage

The Mercoid Corporation

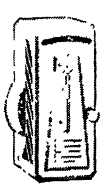
COMPLETE LINE OF AUTOMATIC CONTROLS

Main Office and Factory
4201 BELMONT AVE., CHICAGO, ILL.
Distributors and Jobbers in all Principal Cities

Branch Offices
NEW YORK, N. Y., 393-7th Avenue
PHILADELPHIA, PA., 3137 N. Broad St.
BOSTON, MASS., 839 Beacon St.

Mercoid Controls are noted for their accuracy, trouble-free service and long life. They are equipped exclusively with sealed mercury contact switches—the switch that cannot be affected by dust, dirt or corrosion. See Mercoid catalog No. 500AS showing the complete line with detailed description.

MERCOID SENSATHERMS

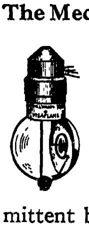


Mercoid room thermostats known as Sensatherms operate on a total differential of 1 deg F (plus or minus ½ deg F). Type H is the standard thermostat for heating, etc. Type DNH (illustrated) is a hand wound day-night thermostat for maintaining lowered temperatures for any period up to 12 hours. At a set time in the morning, it automatically reverts back to the day setting. Type HBH is a two-stage thermostat for control of high-low gas or oil burners. Prevents overshooting on stoker systems, etc. Type HH is a dual thermostat for heating and cooling operations.

PRESSURE AND TEMPERATURE LIMIT CONTROLS

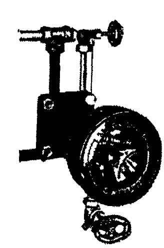
These instruments are of proven reliability and long life. The outside double adjustment with calibrated dial is a time saving feature when making adjustments. Available for steam, hot water and warm air.

VISAFLAME
The Mechanical Eye Actuated by Light



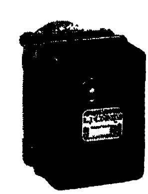
A control system for direct burner mounting. It represents a decided improvement in oil burner safety control. Operates direct from the light of the flame instead of from the heat in the stack. Used in conjunction with the K-2-I panel unit for intermittent burners and the K-2 for constant ignition burners.

COMBINED PRESSURE AND LOW WATER CONTROL



Type DA-131Q prevents firing into dry boilers and guards against building up excess steam pressure. Has quick hook-up fittings designed in accord with the A. S. M. E. Code. Instead of a packing gland, a flexible diaphragm is used which eliminates sticking or erratic operation. Has outside double adjustments. Other types of low water and boiler water feed pump controls available.

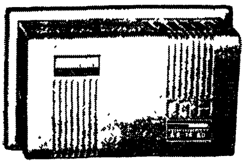
OIL BURNER SAFETY AND IGNITION CONTROL



Type JMI provides positive protection against flame or ignition failure on intermittent ignition oil burners. This control insures having ignition circuit closed before every starting operation of burner. Type JM is used for constant ignition burners.

STOKER TIMER CONTROLS

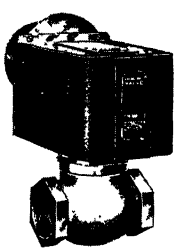
Type TV2 Stok-A-Timer combines a Mercoid Transformer-Relay and a synchronous motor timer mechanism for maintaining the stoker fire during periods when thermostat is not calling for heat. Interval adjustment can be set for ½ hr. or 1 hr. merely by moving a lever. No change of cams required.



"Gradustat"
Pneumatic Thermostat

THE GRADUTROL SYSTEM OF PNEUMATIC CONTROL

The Gradutrol System designation is applied to any combination of Minneapolis-Honeywell Automatic Pneumatic Controls used to govern the operation of air conditioning or heating systems. Such features as infinite positioning with the Gradutrol Relay and accurate graduation of valve and damper motors makes the Gradutrol System a truly remarkable advance in pneumatic control of commercial air conditioning and space heating installations.



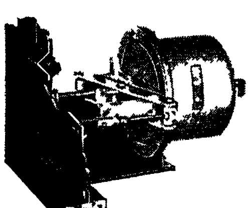
"Grad-U-Valve"
Pneumatic Control Valve

COMBINATION ELECTRIC AND PNEUMATIC SYSTEMS

The outstanding advantages of both the electric Modutrol System and pneumatic Gradutrol System of control may be combined in a single installation. Thus maximum flexibility and low installation cost are obtained. Minneapolis-Honeywell can offer either an electric or pneumatic system, or a combination of the two. This is your guarantee of an unprejudiced recommendation.

BROWN INSTRUMENTS

The extent to which air conditioning equipment is being used in office buildings, theatres, stores, industrial buildings, etc., has opened up a wide demand for indicating and recording resistance thermometers because the temperatures throughout these air conditioning systems should be checked periodically in order to obtain the best results at minimum operating cost. To obtain uniform conditions from modern equipment, it is necessary that the engineer in charge of operation have a visual picture of actual conditions.

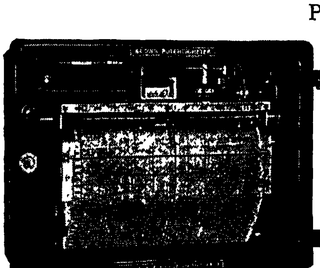


"Grad-U-Motor"
Pneumatic Damper Motor

Brown Resistance thermometers are available for indicating, recording, and controlling service and are applicable to all types of air conditioning and space heating installations.

In addition to Resistance Thermometers, The M-H Brown Instrument Division manufactures:

Thermometers	Flow Meters
Hygrometers	CO_2 Meters
Pressure Gauges	Tachometers
Vacuum Gauges	Liquid Level Gauges
Potentiometer Pyrometers	Protectoglo System



Brown Recording Resistance Thermometer

RESPONSIBILITY FOR ENTIRE CONTROL SYSTEM

Minneapolis-Honeywell Regulator Co. is equipped to assume the entire responsibility for any control installation, thereby eliminating the difficulties and misunderstandings which division of responsibility may create.

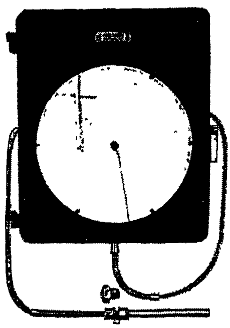
The Palmer Company

Main Plant: 2506 Norwood Ave., **Cincinnati (Norwood), Ohio**

Canadian Factory: King and George Sts., Toronto

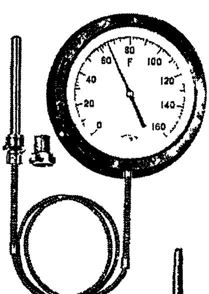
Manufacturers and Originators—"Red-Reading-Mercury" Thermometers

RECORDING THERMOMETERS

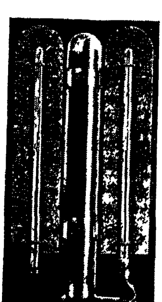


Mercury Actuated. 12 in. die-cast aluminum case. Wrinkle or Satin finish. All parts are rust-proof. Flexible armoured tubing and bulb of stainless-steel. Fittings: Plain, Union, Separable-socket and adjustable or union flange. All ranges up to 1000 F or 550 C. Guaranteed extremely accurate and sensitive. Every part strengthened for long and satisfactory service. *Write for Bulletin No. 1300.*

DIAL THERMOMETER



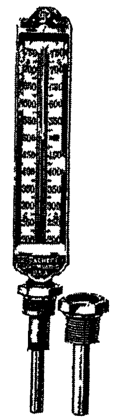
Mercury Actuated. 8 in. case. Black rubberized finish. Flexible armoured tubing and bulb of stainless-steel. All parts are rust-proof. Fittings: Plain, Union, Separable-socket and Adjustable or Union flange. All ranges up to 1000 F or 550 C. Guaranteed sensitive and accurate and to give long and satisfactory service. *Write for Bulletin No. 1500.*



WALL HYGROMETER and SLING PSYCHROMETER

Wet and Dry bulb; Mercury tube, with RED column. Chart furnished. Guaranteed sensitive and accurate. *Send for Bulletin No. 500.*

"RED-READING-MERCURY"

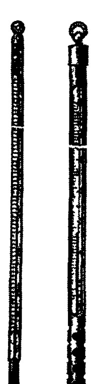


Industrial Thermometers—These mercury tubes will show a bright RED color, visible at a great distance. The color is reflected and cannot fade. (Patented by Palmer). Thoroughly annealed and guaranteed permanently accurate. Costs no more. STRAIGHT, ANGLE, SIDE - ANGLE, RECLINING AND INCLINING Case, OBLIQUE STEM, etc. 7, 9 and 12 in. case, with or without glass front. Standard 3½ in. stem and longer lengths. Fittings: Fixed Thread, Union, Separable-socket and Adjustable or Union Flange. All ranges up to 750 F or 400 C. For ranges up to 1000 F or 550 C, with plain mercury tube, borosilicate glass. *Write for Catalog No. 200-F.*

REPAIRS—To all makes of Industrial Mercury Thermometers, furnishing "Red-Reading-Mercury" tube, at no extra cost and replacing all worn or broken parts, making the thermometer as good as new. Guaranteed accurate. A trial order will convince you.

LABORATORY THERMOMETERS

Glass engraved mercury tube; show bright RED column . . . so easy to see. With or without metal armour; Round or Lens glass; ranges to 750 F. or 400 C. Plain mercury tube borosilicate glass on ranges 1000 F. or 550 C. Correctly annealed and guaranteed accurate.



POCKET THERMOMETERS . . . for quick tests. Reliable and accurate. With RED column.

$-20 + 120$ F.
$0 + 220$ F.

Write for Catalog No. 300-D.

Penn Electric Switch Co.

Goshen, Indiana

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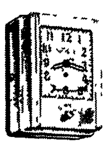
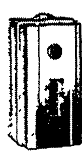
IN CANADA—POWERLITE DEVICES LTD., PENN ELECTRIC SWITCH DIV., TORONTO, ONT.

Distributors and Jobbers in All Principal Cities

Automatic Controls for Heating, Refrigeration, Air Conditioning, Pumps, Air Compressors


Tem-Clock

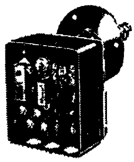

Temtrols

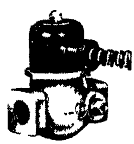
Temperature and Humidity Controls

For control of temperatures and humidity in heating, cooling and air conditioning equipment.


Humidistat

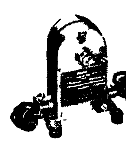

Heavy Duty Thermostats

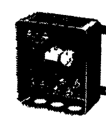

Oil Burner Stack Switches

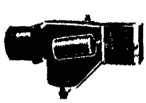

Solenoid Gas Valves

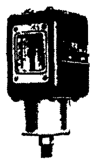
Combustion Controls for all Fuels

For automatic fuel burning equipment, and for stack combustion control.


Damper Motors

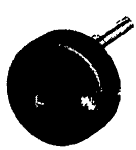

Stoker Timer Relays


Cut-offs and Feeders

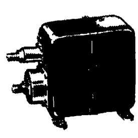

Steam Pressure Controls

Boiler and Furnace Controls

For feedwater, steam pressure, liquids and warm air.


Liquid Immersion Temperature Controls

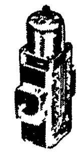

Warm Air Bonnet Controls

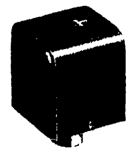

Refrigeration Compressor Controls


Water and Refrigerant Solenoids

Many Others

For control of refrigerants, water and air; and for pumps and compressors.


Water Valves


Pump and Air Compressor Controls

Write for catalog on Penn Controls to cover your particular applications, or 'phone the nearest Penn office or representative. Penn engineers always are available for consulation on control problems, without obligation, of course.

Penn control engineers have simplified design and production problems for others! Let them assist you.

THE POWERS REGULATOR CO.

51 Years of Specialization in Temperature and Humidity Control

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231 East 46th St., New York City

General Offices and Factory
2719-2 Greenview Ave., Chicago, Ill.

The Canadian Powers Regulator Co., Ltd.
195 Spadina Ave., Toronto, Ont.

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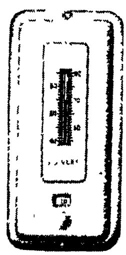
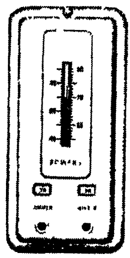
Atlanta	Calgary	Dallas	Halifax	Kansas City	Nashville	Portland	Seattle
Baltimore	Chattanooga	Denver	Hartford	Los Angeles	New Orleans	Rochester	Syracuse
Birmingham	Chicago	Des Moines	Honolulu	Memphis	New York	St. Louis	Toronto
Boston	Cincinnati	Detroit	Houston	Milwaukee	Oklahoma City	Salt Lake City	Vancouver
Buffalo	Cleveland	El Paso	Indianapolis	Minneapolis	Philadelphia	San Antonio	Winnipeg

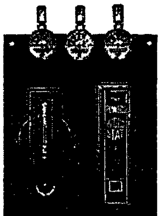


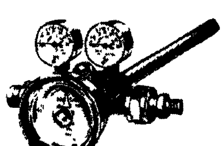


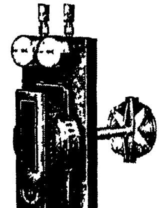
PRODUCTS—A very complete line of compressed air operated and self-operating temperature, humidity and air flow controls for automatically regulating heating, cooling, ventilating and air conditioning systems and industrial processes.

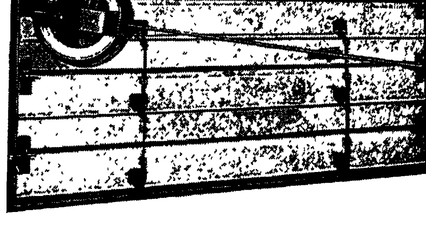
A complete line of self-operating and compressed air operated valves and regulators made for: Controlling steam heated hot water heaters and submerged type heaters; and for automatically mixing hot and cold water or steam and cold water delivering a mixture at a predetermined temperature. Dial Indicating and Recording Thermometers. Thermometer-Regulators. High pressure steam traps and pressure reducing valves.

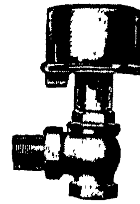
Powers Compressed Air Operated Apparatus



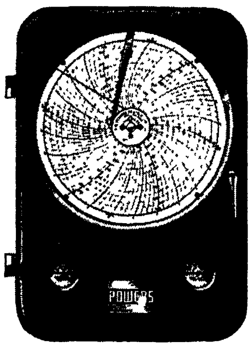


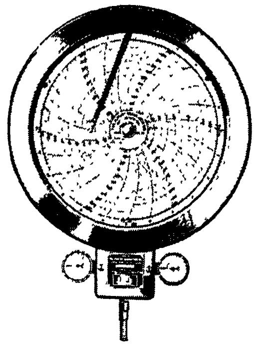


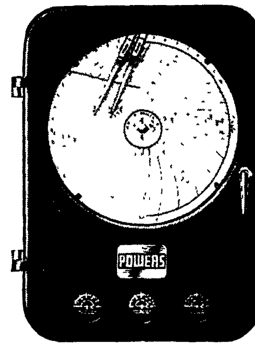


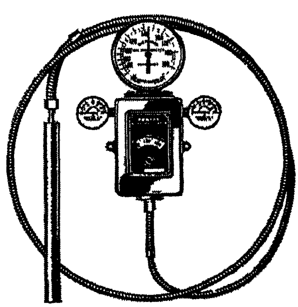


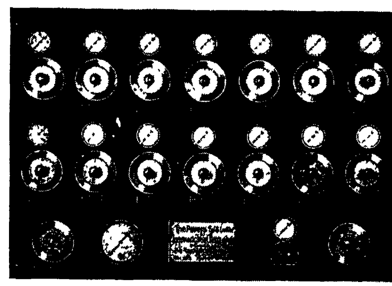
Compressed Air Operated Apparatus—Continued



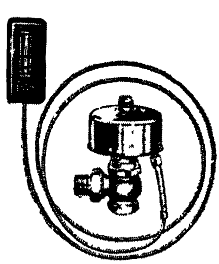


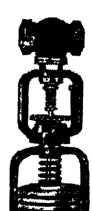


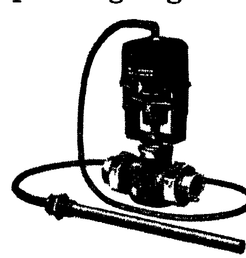




Three of the Many Types of Powers Self-Operating Regulators







POWERS ENGINEERING SERVICE

As the accurate performance of a heating, cooling, ventilating or an air conditioning system, or an industrial process is so dependent upon its automatic control equipment, and as the cost of such control is but a fraction of the entire system, the use of the proper type of regulation is always sound economy.

To secure the maximum return on the investment in automatic control equipment, it is exceedingly important that proper selection of control apparatus be made when each installation is being planned.

Forty-nine years of experience in furnishing and installing temperature and humidity control for every conceivable purpose in all types of buildings have given us a wealth of experience from which you can draw in selecting the proper type of control for any purpose.

CATALOGS AND BULLETINS describing any or all of our products furnished upon request. Phone or write our nearest office. See your phone directory.

Taylor Instrument Companies

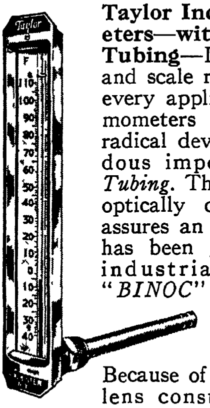
Rochester, N. Y., U. S. A.

IN CANADA—TAYLOR INSTRUMENT COMPANIES OF CANADA, LTD., TORONTO

NEW YORK	PHILADELPHIA	LOS ANGELES	ST. LOUIS	DETROIT
CHICAGO	PITTSBURGH	BALTIMORE	CINCINNATI	ATLANTA
BOSTON	CLEVELAND	SAN FRANCISCO	TULSA	MINNEAPOLIS

Manufacturing Distributors in Great Britain, Short & Mason, Ltd. London

Taylor Instruments for Indicating, Recording and Controlling Temperature, Pressure, Humidity, Flow and Liquid Level

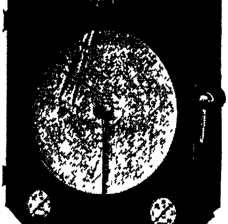


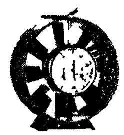
Taylor Industrial Thermometers—with new "BINOC" Tubing—Includes many styles and scale ranges with bulbs for every application. These thermometers contain a new and radical development of tremendous importance—"*BINOC*" *Tubing*. This newly designed and optically correct glass tubing assures an ease of reading that has been generally lacking in industrial thermometers. "*BINOC*" *Tubing* more than doubles the angle of vision within which readings can be made. Because of the patented Triple-lens construction its broad mercury column can be read easily and accurately with both eyes. Bore reflection is absent.

Taylor "BINOC" Pocket Test Thermometer—Ideal for frequent testing of important temperatures. Taylor patented "*BINOC*" *Tubing* eliminates juggling and guesswork. High accuracy—Easier to Read.

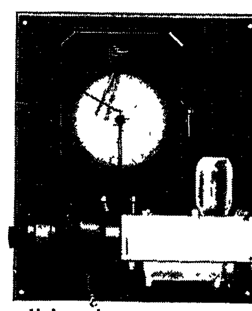
The New Taylor "Fulscope" Recording Controller—An air-operated controller that gives practically any character of process control regardless of *time lag* in apparatus.

Available for controlling temperature, pressure, humidity, rate of flow, liquid level. Where extreme load changes or badly balanced operating conditions exist, precision control can be maintained by the automatic reset feature. For applications where a record is not needed, Taylor supplies an Indicating "Fulscope" Controller.





Taylor Biram's Anemometer—Ideal for measuring air velocities with the fan revolutions indicated on the dial. Various models for a wide range of air speeds and registration limits.



Taylor Recording Hygrometer—Records both wet- and dry-bulb temperatures on the same chart in different colored inks, making comparison very easy.

Type shown has motor-driven fan for conditioned rooms or passages where circulation is poor. Furnished without fan for installations where circulation across bulb is good.

Taylor Sling Psychrometer—The advantage of this form of Wet- and Dry-Bulb Hygrometer over the stationary form is the facility with which tests can be made and the accuracy of the readings obtainable, as the whirling bulbs are subjected to perfect circulation. Two accurate etched stem thermometers are mounted on a die-cast frame, with the bulb of one covered with a wick to be moistened.

These thermometers have scales of 20 to 120 degree F, graduated in ½-deg divisions. A copper case protects the tubes when not in use.

Taylor also offers a complete line of the famous Taylor Recording and Dial Thermometers, Self-Acting and Type "P" Controllers, the 10-BG Hygrometer and many types of Humidiguides.

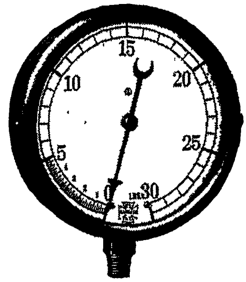
UNITED STATES GAUGE COMPANY

U. S. GAUGE CO. N.Y.

Indicating and Recording Pressure Gauges

44 BEAVER STREET · NEW YORK

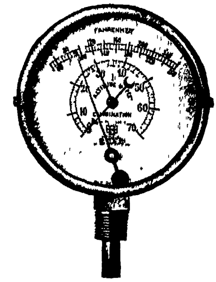
FACTORY SELLERSVILLE PENNSYLVANIA
BRANCHES NEW YORK CHICAGO PHILADELPHIA
BOSTON CLEVELAND DETROIT · ST LOUIS
HOUSTON SEATTLE LOS ANGELES MONTREAL



U. S. GAUGES—U. S. Gauges are made in all standard sizes from 2 in. to 12 in. dial inclusive for pressures up to 50,000 lb and for vacuum. Gauges may be supplied with cast-iron, cast-brass, drawn steel, or drawn brass cases for wall mounting or flush mounting. For severe service requirements we can supply long wearing hardened steel movements or bushed movements.

For service on Steam Heating Systems the following gauges may be supplied—

Steam Gauges . . . Compound Pressure and Vacuum Gauges . . . Retard Gauges . . . Compound Retard Gauges . . . Steam Gauges with Internal Siphons.



For service on Hot Water Heating Systems the following instruments and gauges may be supplied— Altitude Gauges . . . Tank-in-Basement Gauges . . . Altitude and Pressure Gauges . . . Combination Altitude Gauges, and (a) Bimetal Thermometers, (b) Glass Tube Thermometers, (c) Vapor Tension Distance Type Thermometers . . . Glass Tube Hot Water Thermometers.

U. S. RECORDING GAUGES—U. S. Recording Gauges are supplied in 8½ in., 10 in. and 12 in. sizes for pressures up to 50,000 lb and for vacuum. These Recording Gauges can be supplied with either cast-iron or cast-brass cases for wall mounting or flush mounting. Pen arms are made of non-corrosive metal. Especially designed clock movements are used. Charts can be furnished for customary time periods.

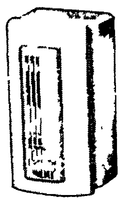
U. S. DIAL THERMOMETERS—U. S. Dial Thermometers are of the vapor tension type with open scale reading in the center and upper portion of the scale, or of the glass filled type with even scale reading. Cases may be cast-iron, cast-brass, drawn steel, or drawn brass for either wall or flush mounting. Supplied in all standard sizes from 2 in. to 12 in. dial inclusive, for temperature ranges from minus 40 deg F to 800 deg F. Furnished with rigid connection bulb or with bulb at end of flexible capillary tubing up to 100 ft long.

White-Rodgers Electric Company

1293 Cass Avenue, **St.Louis, Mo.**

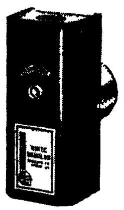
NEW YORK CITY SYRACUSE PHILADELPHIA CHICAGO CLEVELAND
COLUMBUS DETROIT DAVENPORT

Distributors in Principal Cities



Low Voltage Room Thermostat—anticipating type.

Line voltage Thermostat for Unit Heater and Air-Conditioning Installations.



Steam Pressure Control—for safety limit service.

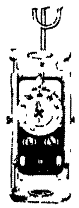
PUT "HYDRAULIC ACTION" TO WORK FOR YOU

The powerful, uniform expansion and contraction of a solid liquid charge against a stainless steel diaphragm, combined with the mechanical simplicity of White-Rodgers Controls contributes these important features to the field of automatic temperature control:

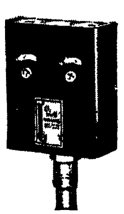
1. *Visible, uniformly calibrated dials.*
2. *Easily set differential adjustor.*
3. *Fast acting thermal elements.*
4. *Combination controls with independently adjustable switches.*

High capacity switch—the tremendous force available with Hydraulic Action has resulted in a sturdy switch with Underwriters' approved rating of 1½ hp 240 v AC, 1 hp 120 v AC.

Take advantage of Hydraulic Action on your next installation. Specify White-Rodgers Controls. The latest condensed control catalog is awaiting your request. Write for it today!

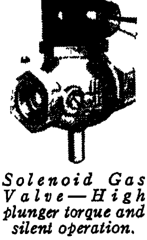


Single speed fan control-cover removed showing visible dial.

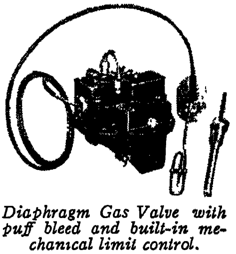


Dual Immersion Control—Limit-Circulator or Summer-Winter service

Stoker Timer—with or without night set-back or fused line switch.



Solenoid Gas Valve—High plunger torque and silent operation.



Diaphragm Gas Valve with puff bleed and built-in mechanical limit control.

International Heating & Ventilating Exposition

THE AIR CONDITIONING EXPOSITION

Permanent Address—Grand Central Palace, **New York, N. Y.**

EXPOSITION HELD

The first in Philadelphia, 1930.
The second in Cleveland, 1932.
The third in New York, 1934.
The fourth in Chicago, 1936.
The fifth in New York, 1938.
The sixth in Cleveland, 1940.
The seventh in Philadelphia, 1942.

Subsequent Expositions will be held on alternate, even numbered years.

These are held coincident with the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and are directed by the International Exposition Company, under the auspices of the A.S.H.V.E.

EXHIBITORS

Comprise leading firms in each phase of the industry; number has varied from 150 to 327 exhibitors.

EXHIBITS

These range from and comprise all the types of articles discussed or advertised in this copy of THE A.S.H.V.E. GUIDE.

1. *The* COMBUSTION *Group:*
   Furnaces, burners (coal, oil and gas), grates, stokers, boilers, radiators (various types), refractories and auxiliaries.
2. *The* OIL BURNER *Group.*
3. *The* HYDRAULIC *Group:*
   Water feeders, water heaters, pumps, traps, valves, piping, fittings, expansion joints, pipe hangers, etc.
4. *The* STEAM HEATING *Group:*
   Vapor heating and steam specialties.
5. *The* HOT WATER HEATING *Group.*
6. *The* AIR *Group:*
   Warm Air furnaces and stoves, registers and grilles, cooling towers, air filters, motors, fans, blowers, conditioning equipment, ventilators (room and industrial types), unit heaters, etc.
7. *The* AIR CONDITIONING *Group:*
   Equipment which circulates and filters the air, in summer dehumidifies and cools; in winter heats and humidifies, and does all these in proper season for complete, all year round air conditioning.
8. *The* CONTROL *Group:*
   Instruments of precision for indicating, controlling or recording temperature, pressure, volume, time, flow, draft or any other function to be measured.
9. *The* REFRIGERATING *Group:*
   Compressors, condensers, cooling apparatus, contingent apparatus and refrigerants.
10. *The* CENTRAL HEATING *Group:*
    Apparatus and materials especially designed or adapted to the uses of central heating and central heating station supplies.
11. *The* INSULATING *Group:*
    Structural insulators (refractory and cellulose materials), asbestos, magnesia clays and combinations thereof, pipe and conduit covering, etc., weather-stripping, etc.
12. *The* MISCELLANEOUS *Group:*
    Electric Heaters, boiler and pipe repair alloys, liquids and compounds, tools of all kinds, and equipment not specifically included in the above groups, but related thereto.
13. *The* MACHINEEY AND GENERAL EQUIPMENT *Group.*
14. BOOKS AND PUBLICATIONS

VISITOR ATTENDANCE

Comprises a registered attendance invited to the exposition and includes:

(Figures are 1940 analysis)

INDUSTRIES

Governmental	401
Distribution Channels Contractors, Dealers, Jobbers, Supply Houses	7,031
Home Owners	333
Industrial Users	9,371
Professional and Service Organizations	689
Public Utilities	900
Real Estate Management and Operation	630
Educational Institutions	500
Miscellaneous	797
TOTAL	20,652

OCCUPATIONS

Executive	11,433
Construction	2,632
Operation	2,353
Technical	2,688
Not Classified including Educators, Publishers, Home Owners, etc	1,546
TOTAL	20,652

Industrial Expositions in America lead the expositions of the world in style, business effectiveness, industrial influence and educational value. This Exposition stands among the leaders in Industrial Expositions in America. It is an educational institution which biennially brings together the research developments and improvements in equipment and materials for use in heating, ventilating and air conditioning all types of buildings.

HEATING SYSTEMS

•

Steam and hot water heating systems with their many parts and accessories are classified according to their specific type of design and the service required. These systems and their component parts include:

HEATING SYSTEMS (p. 1016-1090)

Combinations of parts forming steam vapor and vacuum systems and hot water systems.

Technical data on steam heating systems are contained in Chapter 14; hot water systems in Chapter 16. Other references to heating systems will be found in the Index to the Technical Data Section.

BOILERS (p. 1040-1059)

Water tube, fire tube and firebox types; cast iron and steel construction; for coal, coke, gas or oil firing.

Technical data on heating boilers are given in Chapter 12.

In connection with steam and hot water heating systems various types of radiators and convectors are required. Complete manufacturers references will be found in the Index to Modern Equipment—pages 1137-1160.

Technical data is contained in Chapter 13.

BURNERS (p. 1060-1069)

Automatic fuel burning equipment suitable for use as an integral part of heating boilers and furnaces, and also for conversion of hand-fired heaters to automatic operation. Gas burners, oil burners, stokers.

Technical data are given in Chapter 10.

PUMPS (p. 1070-1073)

For use in conjunction with heating systems, and other purposes in heating, ventilating and air conditioning service; and for handling air, gases, ammonia, brine and other refrigerants.

References to technical data on pumps will be found in the Index to the Technical Data Section.

SPECIALTIES (p. 1074-1086)

Feed water devices, pressure and draft regulators, combustion controls, strainers, traps, valves, etc.—all essential for efficient operation of heating equipment.

References to technical data on heating specialties may be found in the Index to the Technical Data Section, each indexed under its respective title.

PIPE AND FITTINGS (p. 1087-1090)

Iron, steel, wrought iron, copper, brass—seamless or welded.

Technical data will be found in Chapter 18.

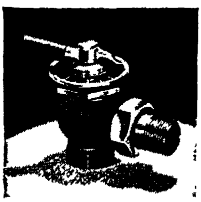
Manufacturer's products shown in this division are designed for specific applications. Consult the Index to Modern Equipment for additional products of these manufacturers.

Barnes & Jones
INCORPORATED

129 Brookside Avenue Boston, Mass.
New York Office: 101 Park Avenue

Barnes & Jones Vapor and Vacuum Systems of Steam Heating; Modulation Valves; Adjustable-Orifice Radiator Valves; Packless Quick-Opening Radiator Valves; Thermostatic Radiator Traps; Thermostatic Trap Replacement Units; Condensators (Boiler Return Traps); Float and Thermostatic Traps; Strainers; Damper Regulators; Gages; Systems of Zone Control for Steam Heating.
Complete Catalog on Request

Modulation Valves, Type K—Packless Quick Opening Valves, Type F



Types K and F Valves have non-tarnishable indicating dial, non-rising stem, renewable disc seat. Tail piece extra heavy. Extra long to facilitate installation. Three models: lever handle, wheel handle, lock shield. Type F Valve furnished with wheel handle only.

Type K Valve

Size	½″	¾″	1″	1¼″
Cap. Sq Ft Rad.* .	30	60	100	180

Type F Valve

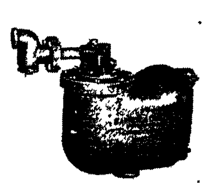
Size......	½″	¾″	1″	1¼″	1½″	2″
Cap. Sq Ft Rad.*. . ..	30	60	100	180	270	400

*Based on 2 oz pressure differential.

Adjustable Orifice Valves, Type H

May be adjusted for different capacities after installation. At all times provides indication of the adjustment. Operation is quiet. Unauthorized tampering with adjustment is virtually impossible.

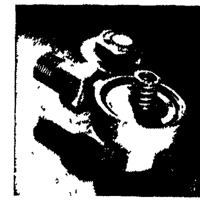
Condensators



For returning water of condensation to boiler from open return line systems independently of boiler pressure, without change in operating conditions, air binding, or admitting steam to the return side.

No.	31	32	33	34	35	36
Cap. Sq Ft Rad.*....	700	1600	3500	6000	10,000	16,000

Thermostatic Radiator Traps



Sturdily made to precision standards. Sensitive in operation. Provide instant discharge of air and water, prevents passage of steam. Contains unique Cage Type Thermostatic Unit, which carries its own thermostatic element, valve piece and valve seat, factory calibrated and locked in correct adjustment.

Trap No.	120	12	123	124	134	13	14
Inlet Tapping... .	½″	½″	½″	½″	¾″	¾″	1″
Outlet Tapping ...	½″	¾″	½″	¾″	¾″	¾″	1″
Cap. Sq Ft Rad.* ..	200	200	400	400	400	700	1200

*Based on 1½ lb pressure differential.

Thermostatic Radiator Cage Replacement Units

Offer complete and reliable trap renewal in practically every make of thermostatic trap. You simply (1) remove the old cover and unit, (2) insert the new Barnes & Jones Cage Unit, (3) replace the cover, and the old trap will operate with its original efficiency.

Float and Thermostatic Traps

Handle large and sudden condensation loads. Large air and water capacity. Large float assures instant opening of the discharge valve. Cage Type Thermostatic Unit assures quick elimination of air.

Trap No......	41	42	43	43A	44B	45B
Inlet Tapping.	¾″	1″	1¼″	1½″	1½″	2″
Outlet Tapping	¾″	1″	1¼″	1½″	1½″	2″
Cap. Lb. Water per Hour* ..	200	700	1200	1200	2400	5000

*Based on 2 lb pressure differential.

Bell and Gossett Company

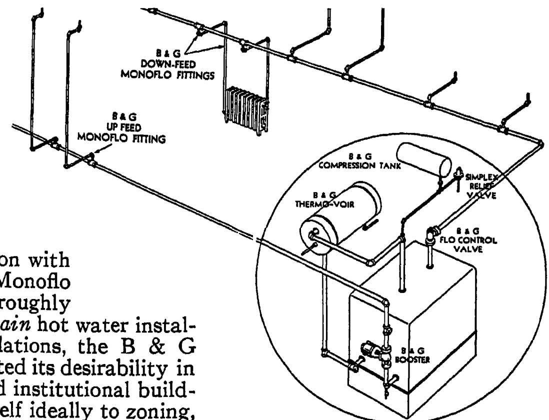
Morton Grove, Illinois

(*Suburb of Chicago*)

HOT WATER SYSTEMS AND SPECIALTIES

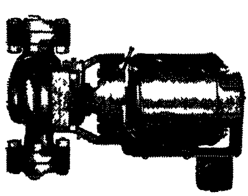
B & G MONOFLO HEATING SYSTEMS

B & G Monoflo Fitting



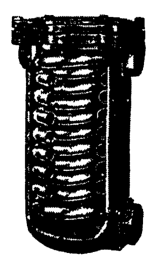
A genuine advance in controlled and economical heating is offered by the B & G Monoflo System. In conjunction with forced circulation, the B & G Monoflo Fitting makes possible a thoroughly practical, well balanced *single main* hot water installation. In over 100,000 installations, the B & G Monoflo System has demonstrated its desirability in homes, apartments, factories and institutional buildings. The equipment lends itself ideally to zoning, yet is exceedingly simple in application.

EQUIPMENT REQUIRED



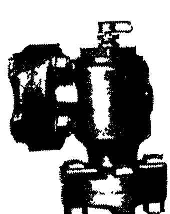
B & G Booster

An electrically-driven centrifugal pump, which mechanically circulates hot water through the system—distinguished by genuine oil lubrication, patented water-tight seal and precision manufacture throughout.



B & G Indirect Water Heater

Any one of five B & G Heater types can be installed to furnish year around domestic hot water at smallest possible cost.

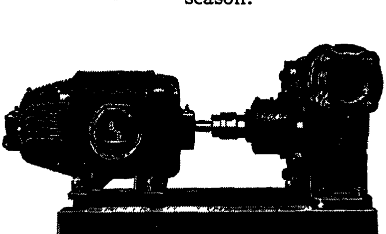


B & G Angle Flo-Control Valve

This valve, installed in the main, controls circulation of hot water to radiators, permitting summer operation of the Indirect Water Heater. It also helps maintain a uniform room temperature during the heating season.

B & G Motorized Valves

B & G Motorized Valves are ideal for controlling boiler water flow through the individual circuits of zoned heating systems.



B & G Universal Pump

THE B & G UNIVERSAL PUMP is designed primarily for large warm water heating systems in apartment buildings, office buildings, factories, schools, etc. The installation can be operated as one large single zone or divided into several zones by controlling the circulation of the pumped water through each circuit with a B & G Motorized Valve, operated by a zone thermostat; the pump being either operated continuously or until all valves are in the closed position.

See the B & G Handbook for Complete Engineering Data

C. A. Dunham Company

Administrative and General Offices

450 E. Ohio Street, **Chicago, Ill.**

Factories: MARSHALLTOWN, IOWA; MICHIGAN CITY, IND.; TORONTO, CANADA; LONDON, ENGLAND

C. A. Dunham Co., Ltd.
1523 DAVENPORT ROAD,
TORONTO, ONT., CANADA

The DUNHAM HEATING SERVICE

C. A. Dunham Co., Ltd.,
(Of the United Kingdom)
Morden Road, LONDON, S W. 19
ENGLAND

See "Dunham Heating Service" in local telephone directory in all principal cities.

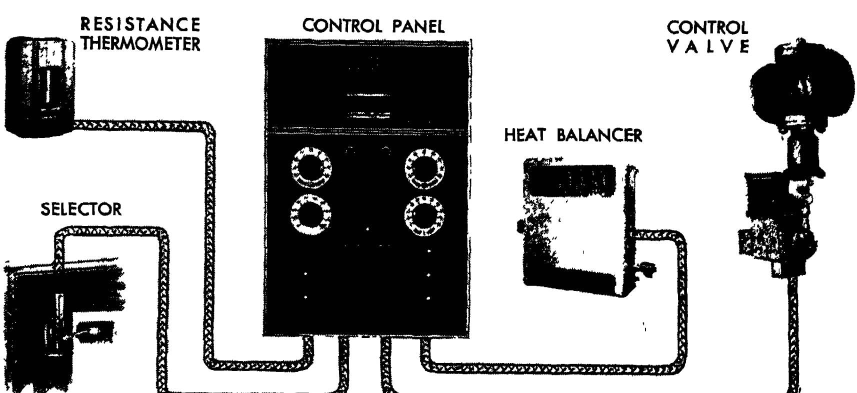
The accumulated experience of the entire Dunham Organization is put at the disposal of the Heating, Ventilating and Air Conditioning Engineer. This cooperation is available for *Modernization Work and Repairs*, as well as for new construction in industrial, commercial, housing and other projects.

The Dunham Differential Heating System, circulating sub-atmospheric steam, maintains desirable temperatures throughout a building by automatic control of both steam temperature and steam volume.

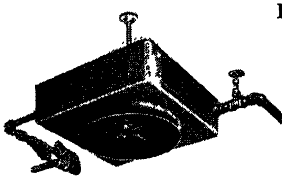
The system is a simple two-pipe system in which all the essentials of circulation, distribution and control are co-ordinated. Control of the temperature of the steam is accomplished by controlling the pressure or vacuum of the steam in the supply piping and radiators to balance exactly the heat input with building-heat-loss.

The Dunham Differential System distributes a varying supply of heat equally, automatically and continuously through the heated space. Desirable building temperatures are automatically maintained under varying weather conditions. A positive *continuous* circulation is maintained as a fundamental function of the system. This maintains unusually constant temperature levels throughout the building.

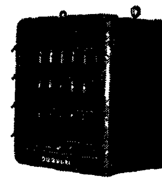
The Control is Fully Automatic. The RT (Resistance Thermometer) Control Equipment, which is an integral part of the System, is fully automatic. Beginning with a maximum radiator heat output obtained by steam circulation at a pressure of 2 pounds and a temperature of 218 F or more as required, the output is progressively reduced according to the demands of the weather, by a reduction in the rate of steam admission to the system, which automatically causes a reduction in steam pressure and temperature so that steam may be circulated at varying temperatures down to about 133 F. Further reduction in heat-output is obtained by partial filling of radiators with sub-atmospheric steam until the point is reached at which the need for heat ceases and the supply of steam is completely shut off. The distribution of the steam supply is automatically maintained under all variations in supply by the co-ordinated functioning of the Traps, Pump, Differential Controller and Regulating Orifices at radiator inlets.



DUNHAM UNIT HEATERS

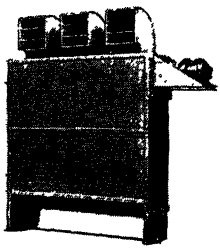


Type C



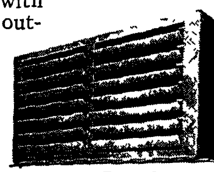
Type V

Type C—A specially designed unit heater for industrial and commercial applications. Designed for discharging large volumes of heated air downward to working levels, distributing heat evenly over large areas. 10 sizes from 136 to 2000 sq ft EDR. Four types of diffusers.



Type R

Type R—Blower type unit for industrial and commercial applications. Available in 16 standard sizes, each with various combinations of Btu and cfm output. Floor, wall and suspended types, with and without by-pass or mixing dampers. All sizes and types either direct connected or belt drive.



Type D

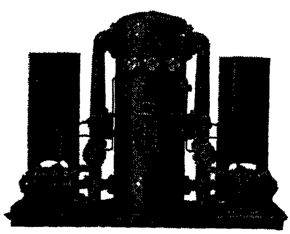
Type V—Horizontal propeller fan type. 35 sizes. Capacities from 65 to 1200 sq ft EDR.

Type D—Horizontal propeller fan type. 12 sizes. Capacities from 558 to 2360 sq ft EDR.

DUNHAM PUMPS

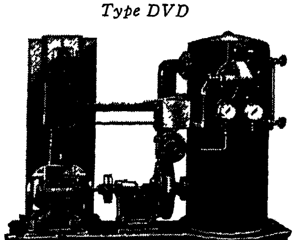
Tested and Rated with A.S.H.V.E. Code and Code of Vacuum Return Line Heating Pump Manufacturers' Section of Hydraulic Institute.

Vacuum Pumps



Type DVD

Type DVD—Designed for use with Dunham Differential Heating. Capable of maintaining whole systems under vacuums as high as 25 in. Built in 9 sizes. Capacities 2500 to 65,000 sq ft EDR. VRD (Vac-Return Line) built in same sizes.



Type VR

Type VR—Meets all code tests for air and simultaneous air and water handling capacities. No moving parts or close clearances in exhauster unit. Built in 11 sizes. Capacities 2500 to 150,000 sq ft EDR. DV (Differential Vacuum) built in same sizes.

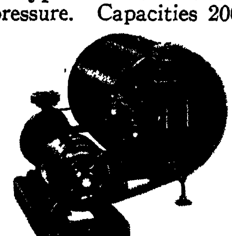
Condensation Pumps

Pump and motor assembled on rigid cast iron base. Bronze fitted centrifugal pump has non-corrosive shaft. Enclosed type Impeller. Liberal size ball bearings. Receiver tank equipped with float switch and push button starting switch with over-load protection.

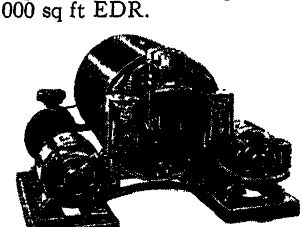
Type CH-Model B, Single and Duplex—66 sizes of varying capacities and discharge pressures. Capacities 2000 to 50,000 sq ft EDR; 60 cycle d.c. or a.c. 1750 rpm; 25 or 50 cycle a.c., 1450 rpm.

Type CHH-Model B, Single and Duplex—49 sizes of varying capacities and discharge pressures. Capacities 2000 to 50,000 sq ft EDR; 60 cycle d.c. or a.c. 3450 rpm; 25 or 50 cycle a.c., 2850 rpm.

Type CV—48 sizes of varying capacities and discharge pressure. Capacities 2000 to 25,000 sq ft EDR.



Single Unit Type CH—Model B
CHH—Model B, same connections



Duplex Unit Type CH—Model B
CHH—Model B, same connections

Type CV

GRINNELL COMPANY INC.

Heating, Industrial and Power Plant Piping, Fittings, Hangers, Valves, Pipe Bending, Welding, Piping Supplies, Etc.

Executive Offices: **Providence, R. I.**

Offices, Plants and Branches

Albany, N. Y.
Atlanta, Ga. (Plant and Foundry)
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GRINNELL COMPANY OF THE PACIFIC

Los Angeles, Cal. (Branch) Oakland, Cal. (Branch) San Francisco, Cal. (Branch) Seattle, Wash. (Branch)

GRINNELL COMPANY OF CANADA, LTD.

Montreal, Que (Branch) Vancouver, B. C. (Branch) Toronto, Ont. (Plant and Foundry) Winnipeg, Man. Oshawa, Ont. (Foundry)

PRODUCTS AND SERVICES—

Complete Service on materials to Specification on Power Plant Piping, Industrial Piping, and Industrial Heating Systems; Prefabricated Piping including Pipe Cutting and Threading, Pipe Bends, Welded Headers, Welded and Welding Fittings, Lap Joints and the Grinnell line of products for Super Power.

Grinnell Equiflo Valves for forced hot water heating systems; Grinnell Adjustable Pipe Hangers and Supports; Grinnell Cast Iron and Malleable Iron Pipe Fittings; Grinnell Malleable Iron Unions; Grinnell Welding Fittings; Grinnell Thermoliers (Unit Heaters); Grinnell Thermofin (Convectors); Thermoflex Traps and Heating Specialties.

Also Humidifying Systems; Constant Level Size Circulating Systems; Piping for acids and other special materials.

Malleable Iron, Brass, Bronze and other Castings; Brass, Cast Iron, Wrought Iron and Steel Pipe; Seamless Steel Tubing in Iron Pipe Sizes.

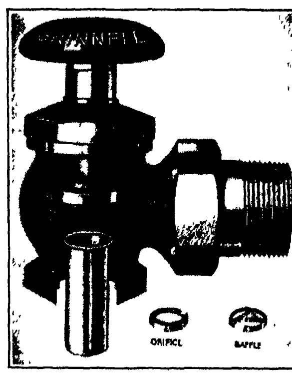
Valves: Check, Globe, Pressure Reducing and Regulating, Quick Opening, Safety and Y.

Automatic Sprinkler Systems; Stand Pipes; Underground Supply Mains; Hydrants; Fire Pumps; Pressure and Gravity Tanks.

Grinnell "Junior" Automatic Sprinkler Systems for Basements and other hazardous areas of Dwellings, Small Apartment Buildings, Schools, Churches, Stores, etc.

Grinnell Equiflo Valves

For Forced Hot Water Heating



Equiflo Valve

The designing of forced circulation hot water heating systems is so simplified by the Grinnell Equiflo Valve that they can be laid out and installed as easily as vapor or steam systems. This valve consists of a regular type packless radiator valve with a cartridge or tube made up of a series of orifices and baffles capable of setting up any required frictional resistance. This method of establishing any desired resistance does away with elaborate calculation of pipe sizes. Grinnell guarantees perfectly balanced circulation to each and every radiator where these valves are installed throughout the system.

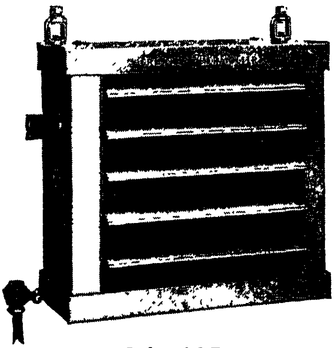
Equiflo Data Book sent to interested parties.

For Data On Thermoflex Traps and Heating Specialties, see page 1079

THERMOLIER

Patented

THE GRINNELL UNIT HEATER



Industrial Type

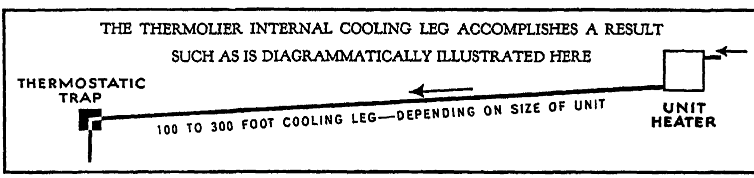
De Luxe, Industrial and Factory Types—125 Lb W.S.P.

Thermolier is a ruggedly built unit heater whose efficiency and dependability have been proved by actual performance in field service. Thousands of them are installed in industrial buildings and commercial structures of all types of occupancy.

Thermolier has 14 points of superiority, the most outstanding of which is the internal cooling leg built right into the unit, an exclusive Thermolier feature. See drawing below.

Radiation is from brass-finned seamless copper U-tubes rolled into a cast-iron tube sheet. No solder is used for strengthening joints and there are no flat horizontal surfaces to catch dirt.

Units may be controlled manually or automatically, singly or in groups. Installation and piping are extremely simple and inexpensive, hence the unit may be moved from one



location to another at small cost if found desirable on account of changes in building or occupancy. The complete line includes 27 Models in DeLuxe Type, and 35 Models each in Industrial and Factory Types.

Thermoliers provide maximum distribution of heat without objectionable drafts.

Specifications

Fan—Grinnell special of rugged construction. **Motor**—heavy duty, oversize, enclosed, moisture-proof. **Housing**—Art Metal Slate gray finish with chromium trim on DeLuxe Types. Copper on Industrial Type with rubbed lacquer finish; steel on Factory Type finished in gray lacquer. **Frame**—Heavy pressed steel, providing rugged support for motor and fan. **Special Features**—Adjustable swivel hanger rod couplings; louvers rigid, but easily adjustable: integral cooling leg insuring perfect drainage through one thermostatic trap for pressures up to 25 lb.

For pressures not exceeding 125 lb, a thermostatic trap of proper construction can be used and should be attached directly to the unit.

CAPACITIES

60 F Entering Air Temperature—2 lbs Steam Pressure

Model Nos.	Btu per Hour	Model Nos.	Btu per Hour	Model Nos.	Btu per Hour
21	35,600	46L	63,500	81L	143,600
21L	28,350	51	98,600	91	196,000
22	37,100	51L	73,700	91L	160,600
26	38,650	57	101,300	101	237,800
26L	31,750	57L	83,000	101L	205,400
31	48,700	61	102,100	111	275,300
31L	36,200	61L	77,500	111L	229,700
37	62,200	66	128,700	141	325,300
41	70,900	66L	110,500	141L	269,700
41L	56,600	71	151,700	181	373,300
44	84,800	71L	126,800	181L	303,200
46	86,400	81	174,900		

Data Book covering other pressures and temperatures, dimensions and complete installation information on application. Address GRINNELL COMPANY, INC., 277 West Exchange Street, Providence, R. I.

GRINNELL ADJUSTABLE PIPE HANGERS AND SUPPORTS

One of the chief advantages of Grinnell Adjustable Hangers is that they permit adjustment of pipe lines after installation, thus obviating the necessity of turnbuckles or the removal of hangers. Their time and trouble-saving qualities during installation are equally exceptional. Below are shown a few Grinnell Hangers and Supports of particular interest to heating engineers. Send for Hanger Catalogue showing complete line.

Adjustable Swivel Rings (Patented)

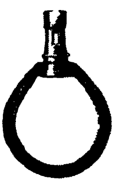


Fig. No. 101
Solid Ring

These Malleable Iron Adjustable Swivel Rings can be used with Coach Screw Rod or Machine Threaded Rod in connection with practically any type of Ceiling Flange, Expansion Case, Insert, etc.

Adjustment of at least 1½ in. is secured by turning Swivel Shank. Swivel Shank automatically locks, preventing loosening due to vibration in the pipe line.

The Split Ring permits adjustment either before or after Ring is closed. A wedge type pin is loosely but inseparably cast into the hinged section for fastening this section after pipe is in place.

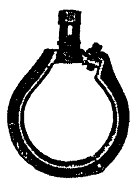


Fig. No. 104
Split Ring

Adjustable Swivel Pipe Rolls (Patented)

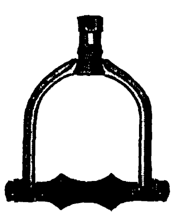


Fig. No. 174
Swivel Pipe Roll

An adjustable type of pipe roll using a single hanger rod. Swivel Shank allows vertical adjustment and automatically locks, preventing loosening from vibration.

CB-Universal Concrete Inserts
(Patented)

Made of air furnace malleable iron, in one body size, to take a special removable nut, tapped for ⅜ in., ½ in., ⅝ in. or ¾ in. rod as required. Nuts automatically lock by means of V-type teeth on both insert and nuts.

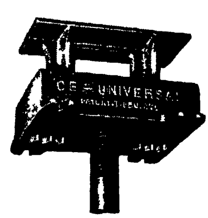
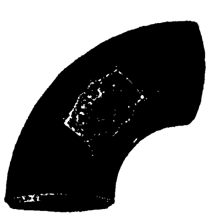


Fig. No. 288
CB-Universal Insert

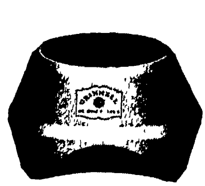
GRINNELL WELDING FITTINGS



90° Elbow, Long Turn

Grinnell Welding Fittings are made from Seamless Steel Pipe or tubing and possess the same physical characteristics as standard, extra strong and o.d. steel pipe or seamless steel pipe of comparable size. They can be used under the same conditions, pressures and temperatures as the pipe itself.

All Grinnell Welding Fittings have welding faces for all plain circumferential butt welds scarfed or beveled as follows: For wall thicknesses $\frac{3}{16}$ to ¾ inch inclusive, 37½ deg. ± 2½ deg., straight bevel. Angles of bevel other than 37½ deg. can be furnished on special order.



Welding Outlet

Welding Tee

Lap-Joint Stub End with Lap Flange attached

Threaded Outlet

William S. Haines & Company

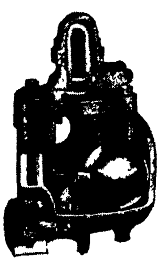
12th and Buttonwood Sts., **Philadelphia, Pa.**

Manufacturers of

EQUIPMENT FOR VAPOR AND VACUUM HEATING SYSTEMS

PRODUCTS—Haines Vento Radiator Traps, Medium Pressure and Blast Type Traps, Combined Float and Thermostatic Traps, Air Eliminators, High Pressure Thermostatic Traps, Boiler Return Traps, Radiator Valves.

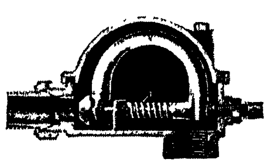
HAINES F & T TRAPS



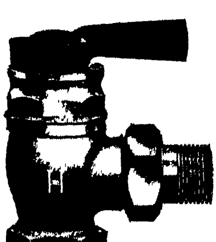
Designed to handle large quantities of condensation. For dripping steam mains, unit heaters, hot water generators. etc. Cannot become air bound as it has a thermostatically controlled air by pass. Sizes ¾ in., 1 in., 1¼ in., 1½ in.

HAINES MEDIUM PRESSURE TRAPS

A ruggedly constructed bolted case trap. Ideal for hospital and kitchen equipment and all process work operating on pressure up to 60 pounds.



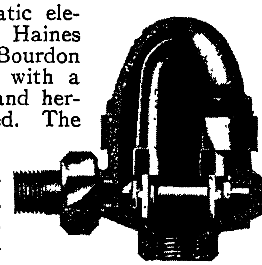
HAINES MODULATING VALVES



A packless valve assuring positive and leak proof performance. Completely opens or closes on less than a full turn of handle. Can be furnished with wheel or lever handle or lockshield.

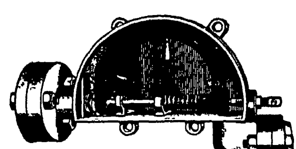
HAINES RADIATOR TRAPS

The thermostatic element in all Haines Traps is a Bourdon tube, charged with a volatile fluid and hermetically sealed. The expansion and contraction of the fluid, under varying temperatures, furnishes the operating power.



The vertical seat of this trap prevents it from becoming inoperative from scale or other foreign matter.

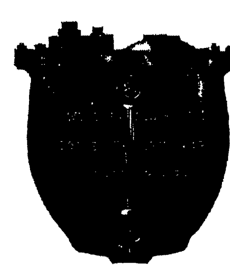
HAINES HIGH PRESSURE TRAPS



For dripping high pressure mains, laundry equipment and all process fixtures with working pressures up to 125 pounds.

HAINES BOILER RETURN TRAPS

For vapor and atmospheric heating systems. Assures positive circulation by venting the air and returning the water of condensation to the boiler. Has no stuffing boxes or packed joints to leak air or water.



Each device is individually tested, factory adjusted and guaranteed.

Hoffman Specialty Co., Inc.

General Office and Factory

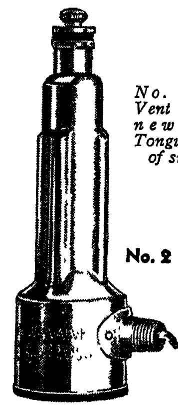
1001 York Street, **Indianapolis, Ind.**

Sales Representatives in Principal Cities

Manufacturers of Adjustable Port Radiator Venting Valves, Quick Vents and Air Eliminators for One and Two Pipe Steam and Vacuum Systems; Hoffman Supply Valves, Traps and Basement Specialties for Controlled Heat Systems, Air Conditioner Hoffman-Economy Vacuum and Condensation Pumps, and Hot Water Controlled Heat Equipment.

AIR VALVES

The Nos. 1, 1A and 40 are used for venting radiators on One and Two Pipe oil or gas automatic fired Steam Systems, and the Nos. 4, 4A, 75 and 75A are used in conjunction with these valves for venting steam mains, risers and other quick venting service.



No. 2 Vacuum Vent Valve with new "Short Tongue" in place of siphon tube.

No. 2

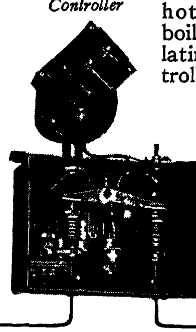
VACUUM VALVES

The Nos. 2, 2A Vacuum Air Valves feature the Hoffman Double Air Lock consisting of the vacuum check and vacuum diaphragm. These valves are for use on coal burning hand or stoker fired One Pipe Vacuum Systems; and for venting ends of steam mains or heating risers, where it is also desired to prevent the return of air into the system, the Nos. 6, 16, 16A, 76 and 76A vents are used.

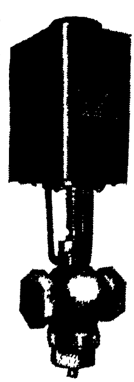
HOT WATER CONTROLLED HEAT EQUIPMENT

The Hoffman Temperature Controller is connected by capillary tubing to the Outdoor Temperature Bulb, and to the Water Temperature Bulb installed in the supply main. Variations in outdoor and circulating water temperatures are instantly transmitted by these two Bulbs to the Temperature Controller which electrically opens or closes the Control Valve.

Temperature Controller

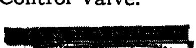


The Hoffman Control Valve. Admission of hot water from the boiler into the circulating system is controlled by this valve. It is opened and closed electrically when actuated by demands for more or less heat from the Hoffman Temperature Controller.

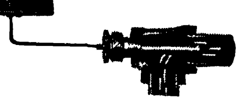


Control Valve

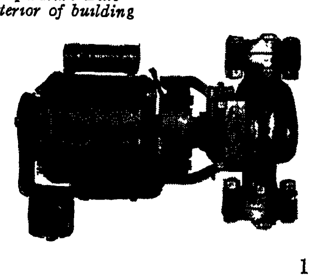
Available in sizes to correspond with Hoffman Circulator.



Outdoor Temperature Bulb located on exterior of building



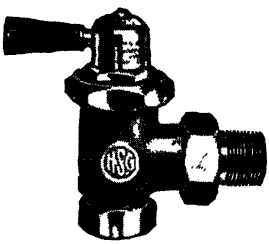
Water Temperature Bulb



Hoffman Circulator

The Hoffman Circulator is a centrifugal pump of large capacity, low power consumption and furnished in all standard sizes. It is installed in the return main and operates continuously except when outdoor temperature rises above 65 deg.

HOFFMAN CONTROLLED HEAT



No. 7 Modulating Valve

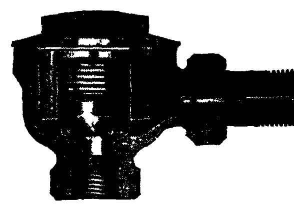
A Hoffman Controlled Heat System consists of the No. 7 Adjustable Orifice Modulating Valve on the supply end of the radiator, the No. 8-A Thermostatic Trap on the return end and either a Hoffman Differential Loop (for coal-fired installations operating at pressures up to 8 oz), or a Boiler Return Trap where higher pressures are encountered, for returning the condensate to the boiler.

SUPPLY VALVES

Besides the No. 7 Adjustable Orifice Modulating Valve the Nos. 37 and 80 series (not illustrated) represent a complete line of Packless Supply Valves that meet the exacting requirements of architects and engineers.

THERMOSTATIC TRAPS

The line of Bellows Type Thermostatic Traps, with hydraulically formed and tested bellows, consists of the Nos. 17-A, 8-A and 9-A, and are principally used for low pressure steam or vapor systems. These traps have nominal capacities from 200 sq ft up to 700 sq ft of radiation. The Nos. 8-A and 9-A have renewable elements, which combine the thermostat, valve pin and renewable seat into a single unit.

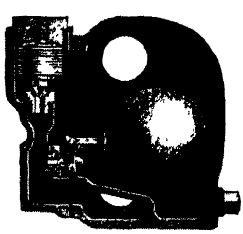


No. 8A—½ in.

The No. 10-A Hoffman Trap, which is equipped with waterhammer proof bellows, 1 in. connection, has a nominal capacity of 2800 sq ft.

The Nos. 8 and 9 Traps have a thermostatic element consisting of three chambers each having a top and bottom diaphragm. These chambers are all joined together and the complete thermostatic member is housed in a cage and is not attached to the valve body or cap. This allows the thermostatic element and valve pin to be easily removed and replaced without adjustment. These traps range in sizes from ⅜ in. to 1 in., are medium pressure traps, and are recommended where pressures up to 50 lb are encountered.

The Nos. 20-H and 21-H High Pressure Traps are equipped with waterhammer proof bellows, for use on pressures up to 125 lb. Available in ⅜ in. to 1 in. connection.



No. 60 Series Trap

DRIP AND HEAVY DUTY TRAPS

Where large amounts of condensation are encountered, it is recommended to use one of the float and thermostatic traps, which are available with or without the thermostatic element. These traps are available in large capacities and are mainly used for venting and dripping risers, steam mains, unit heaters, blast coils, etc. These traps are made in four different pressure ranges 15 lb, 30 lb, 60 lb, and 125 lb.

VACUUM AND CONDENSATION PUMPS

The Hoffman-Economy line of Vacuum and Condensation Pumps offers a dependable method of economically returning the condensation from larger heating systems to the boiler. These pumps are made in single and duplex units, for varying capacities and pressures.

HOFFMAN SALES AND SERVICE

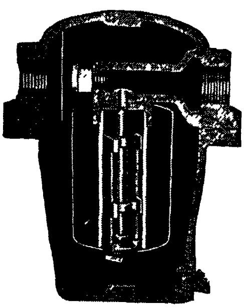
Hoffman Products are sold and stocked by leading wholesalers of heating and plumbing supplies everywhere. Hoffman representatives are available to assist in selection of suitable equipment for various services.

ILLINOIS ENGINEERING COMPANY

General Offices
and Factory:
Chicago

Branches and Representatives In Principal Cities

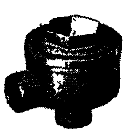
Illinois Steam Trap



Series 30

Valve and stem are separate from the bucket and operated only by the bucket at the extreme top and bottom of travel—result—*valve is always either full open or tight closed.* No wire drawing or cutting of valve and seat which are of stainless steel.

Illinois Thermostatic Traps for High Pressures

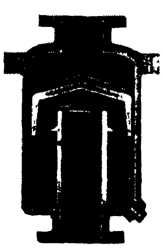


Series HG

Maximum working pressure 150 pounds. Used where neat appearance and compactness are desirable, as for trapping sterilizers or water stills in hospitals; steam jacketed kettles, coffee urns, warming tables and for process work. Also used extensively for air vents on blast type drying heaters. Multi-diaphragm of phosphor bronze. Heavy duty bronze body. Made in three sizes.

These traps are also furnished for medium pressures.

Steam and Oil Separators

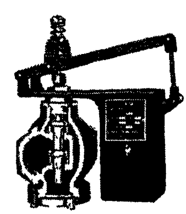


Vertical Standard Separators

Eclipse steam separators are made in both horizontal and vertical type, and also the special receiver separators for standard or extra heavy pressures.

Eclipse oil separators are furnished in the horizontal type and have a removable baffle plate to facilitate cleaning of baffle and keeping the separator's efficiency at the highest point.

Illinois Motorized Valves (on and off)



Type E

For automatic control of steam temperatures and pressures to prevent overheating and conserve steam; to control fluid levels; and to regulate flow in hot water heating systems. May be operated by any automatic contact device or by manual switches.

Furnished in three types.

Spring Controlled Regulating Valve

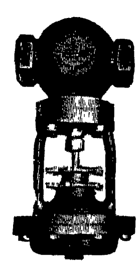


Fig. 121

Furnished in either single seated or double seated type as the service conditions require, for the control of steam, air or gas. Controlling spring is completely enclosed, protecting it from dirt and rust. Valves are furnished with the proper size diaphragm and the proper length spring to give satisfactory service under all operating conditions. Furnished also in weight loaded type, Fig. 71.

Master Type Pressure Regulator

Fig. 142

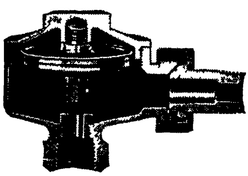
Used wherever high pressure steam must be accurately reduced in varying amount to any steady lower pressure, in service such as hospital, laundry, cooking, process, dry kilns and railway steam control. It will reduce initial pressure up to 250 pounds down to any lower pressure. Does not build up pressure on a closed or dead end line. Made of bronze with monel metal valves and seats.

ILLINOIS ENGINEERING COMPANY

General Offices
and Factory:
Chicago

**Branches and
Representatives
In Principal Cities**

Illinois Thermo Radiator Traps



Series G

Illinois Thermo Radiator Traps for vacuum, vapor and low pressure heating systems. Has cone type valve. Flushes thoroughly and seats perfectly at all times. Valve and seat are of Nitralloy. The duplex diaphragm is of special phosphor bronze. Scientific design and rugged construction assure flexibility and long life. These diaphragms have withstood over three million strokes on a breakdown test.

Boiler Return Trap

Vapair Vent Trap

Vapor Gauge

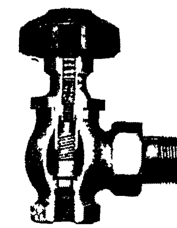
Damper Regulator

Illinois Selective Pressure Control Systems

Illinois Selective Controller

An entirely new and unique method of Steam Circulation Control . . . Heating Systems that set new standards in comfort, economy, simplicity and convenience of operation. Each system is individually engineered to meet the exact requirements. Recorded fuel savings, without sacrifice of comfort, warrant your investigation. Ask for Bulletin 16.

Illinois Modulating Supply Valve



Quick-opening, packless. Steam tight on 50 lb pressure. Large diameter of thread spool and machine cut threads make valve operation easy. Furnished in a complete line of sizes and patterns.

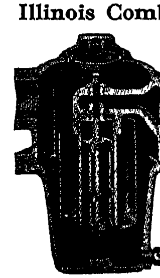
Illinois Vapor System

A two pipe low pressure steam circulating system which may be installed in any type of building, where the condensate can return to the boiler by gravity.

A sensitive damper regulator or other means of automatic control is used to control initial steam pressure above, at or below atmospheric pressure. Steam is regulated at the radiators by Illinois Modulating Supply Valves. Condensate and air are discharged from the radiator through Illinois Thermostatic Radiator Traps. In the boiler room a Vapair Vent Trap and Boiler Return Trap are installed near the boiler. The vent trap eliminates air from the system and the Return Trap insures return of condensate to the boiler.

The system and the piping arrangement are simple. No metering orifices or vacuum pumps are needed. This system will be found suitable for many installations where low first cost and low operating cost are of prime importance. May be used with unit heaters or any type of radiation.

Illinois Combination F & T Traps



Series 7G

Unsurpassed for draining ventilating units, unit heaters, and for dripping mains and risers—wherever it is desirable quickly to vent air from the main as well as handle the water of condensation in quantity, whether hot or cold.

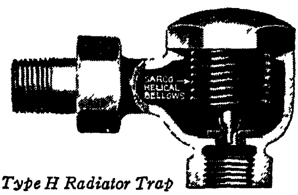
Sarco Company, Inc.

475 Fifth Ave., **New York, N. Y.**

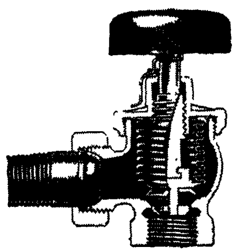
Branches in Principal Cities

SARCO CANADA LIMITED, 85 RICHMOND ST., W., TORONTO, ONT.

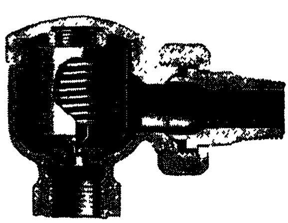
PRODUCTS—A complete line of Specialities for Vapor, Vacuum and Gravity Steam Heating Systems and Control combined with a competent Engineering Service to architects and heating engineers to assist them in providing modern heating.



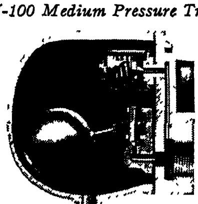
Type H Radiator Trap



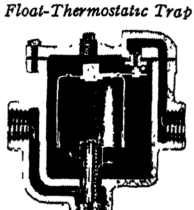
Bellows-Packless Valve



N-100 Medium Pressure Trap



Float-Thermostatic Trap



Inverted Bucket Trap

SARCO RADIATOR TRAPS

Sarco Heating Systems are "prestige Systems." The traps and valves *are* the system as far as maintenance and cost are concerned.

Sarco Type H Traps—Are available in angle, straightway, and corner patterns. The Sarco Thermostatic Bellows—made by special machinery, has not been duplicated or even imitated with success. It works efficiently, repeatedly and persistently. It has worked that way for a quarter of a century. Sizes ½ in. to 1 in. Catalog HV-45.

SARCO RADIATOR VALVES

Sarco Packless Valves—Used for one and two pipe heating systems and are truly packless. Steam leaks are impossible. Furnished with round or lever handles or lock shield in angle, straightway, or corner patterns. Sizes ½ in. to 1½ in. Catalog HV-45. For Hot Water Heating Systems, Catalog HV-175.

SARCO N-100 TRAP

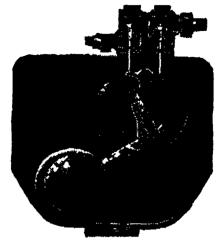
For high pressure radiators and heating coils in stationary and marine service, and for hospital and kitchen equipment. Has full length protecting shield and stainless steel valve head and seat. Sizes ⅜ in. to 1 in. Catalog HV-46.

SARCO FLOAT-THERMOSTATIC TRAPS

For dripping ends of mains and risers, and for stack or blast heaters, large unit heaters and hot water generators. Automatic thermostatic air vents built in. Available in six sizes with connections ¾ in. to 2 in. Catalog HV-38.

SARCO INVERTED BUCKET TRAPS

Are recommended for high pressure unit heaters and sometimes preferred for kitchen and laundry equipment. Strainers are built right into these sturdy traps. Seats and valves are stainless steel and renewable. Thermostatic air vents can be furnished on the larger sizes. Available in sizes ½ in. to 2 in. for pressures up to 900 lb. Catalog HV-350.



Alternating Receiver

SARCO ALTERNATING RECEIVER

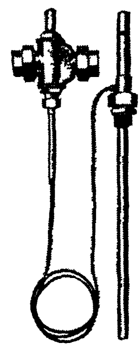
A complete line of boiler return traps for vapor systems.

Returns water of condensation to boiler automatically, thereby assuring positive return of water under all pressure conditions.

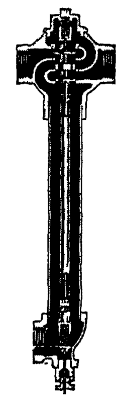
Made in six sizes for from 1,500 to 25,000 sq ft of radiation. Catalog HV-45.

SARCO AIR ELIMINATORS

For venting air from vapor systems at one central point in the basement. Available in two sizes: No. 6 for systems up to 3,000 sq ft and No. 12-A for 15,000 sq ft. Both are equipped with float valves to stop water escaping through the vent and with check valves to prevent ingress of air when system is under vacuum. Catalog HV-45.



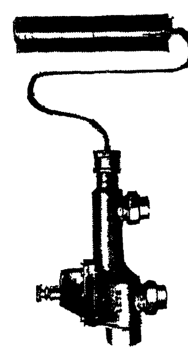
Type TR-21 Standard for hot water storage tanks, fan units, etc.



Water Blender Type MB

SARCO SELF-CONTAINED TEMPERATURE REGULATORS

Sarco Temperature Regulators are simple, self-operated valves—the only self-contained units that use the irresistible force of liquid expansion. No stuffing boxes to leak, no auxiliary "power" required; all moving parts are *inside* the equipment. Here again—a type and size for every purpose—for steam, gas, oil, water or brine for temperatures ranging from 0 to 400° F. Catalog HV-52.



Sarcotherm Weather Control Valve

SARCO WATER BLENDERS AND TEMPERING VALVES

For mixing hot and cold water to deliver automatically water at any desired temperature. Two models are available, type MB for showers, wash basins, etc., and type DB, a tempering valve for use with submerged heating coils or tankless heaters. Catalog HV-800.

SARCOTHERM HOT WATER HEATING SYSTEM

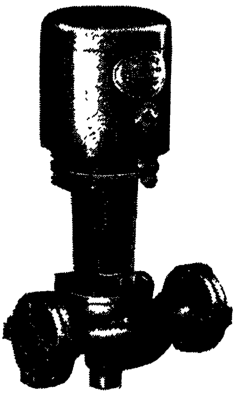
A simple, all-mechanical system for the control of radiator temperatures in direct relation to outside temperatures. Radiation is balanced by Sarcoflow fittings in the radiator outlets.

The Sarcotherm three-way valve recirculates a varying proportion of water around the boiler and back to the system as dictated by the thermostatic bulb outside the building. Catalog No. HV-175.

Balancing Fitting

Triplex Heating Specialty Co., Inc.

242-268 Grant St. Peru, Indiana



Direct Drive

HOT WATER SYSTEMS AND SPECIALTIES

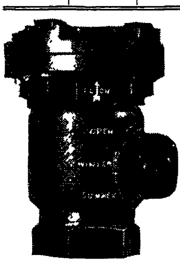
DIRECT DRIVE AQUALATOR

A most powerful Circulator. The design of the pump proper is the result of years of experiments. It will deliver water against an 11 ft head—about twice as powerful as direct driven pumps. The trouble-free coupler combined with the unique stuffing box assures lasting service. The Circulator is lubricated, protecting the bearings and stainless steel shaft. Every motor is protected with an automatic overload switch and meets all states, electrical codes.

Motors of all Voltages and Cycles carried in stock.

SIZES AND CAPACITIES

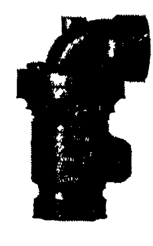
No.	Size	60 Cyc. 110V Motor Size	Circulator R. P. M.	Rad. Cap. Sq Ft	Storage Tank Cap.	Gals. Per Min	Max. Head	End to End	Ship. Wt.
22D	¾"	1/8 H. P.	1750	300	500 gal.	20	11½'	8¾"	46
23D	1"	1/8 H. P.	1750	500	1000 gal.	25	11½'	8¾"	46
24D	1¼"	1/6 H. P.	1750	800	1500 gal.	35	11½'	8¾"	46
25D	1½"	1/6 H. P.	1750	1200	2000 gal.	50	11½'	8¾"	46
26D	2"	1/6 H. P.	1750	2000	2000 gal.	75	11½'	8¾"	56
26HD	2'	1/4 H. P.	1750	2000	3000 gal.	90	12'	11¼"	65
27HD	2½"	1/3 H. P.	1750	3000	4000 gal.	115	14'	11¼"	85



Straight Type Flow Control Valve

THERMOLATORS

Only one Flow Control Valve is required in a properly balanced Flow Control System. For best operation, the air should be eliminated from the boiler and carried directly to the expansion tank. The Anti Gurgle Fitting shown in the bottom of the Flow Control Valve positively accomplishes this. TRIPLEX Flow Control Valves have a convenient lever handle to set the valve for emergency or seasonal operation. The Anti Gurgle Fitting is not inlcuded with the price of the Thermolator.



Angle Type Flow Control Valve

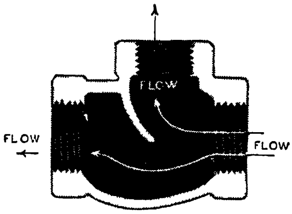
No.	Size	Width	Height	Sq Ft Rad.	Ship. Wt.
123	1"	4"	4"	500	4
124	1¼"	5"	7½"	800	9
125	1½"	5½"	8½"	1200	11
126	2"	6"	9¾"	2000	15
127	3"	6¾"	9½"	5000	28

No. 127 is not an angle type. When ordering specify straight or angle type.

FLOW CONTROL SYSTEMS COMPLETE

SIZE INCHES	¾" P. 1" FCV	1" P. 1" FCV	1¼" P. 1¼" FCV	1½" P. 1½" FCV	2" P. 2" FCV	3" P. 3" FCV
No. of System	22-3-501	23-3-502	24-4-503	25-5-503	26-6-504	27-7-5044
Sq Ft Rad.	300	500	800	1200	2000	5000

TRIPLEX DISTRIBUTOR and AIR ELIMINATOR FITTINGS



CUT OPEN VIEW OF TRIPLEX DISTRIBUTOR

COPPER PIPE DISTRIBUTORS

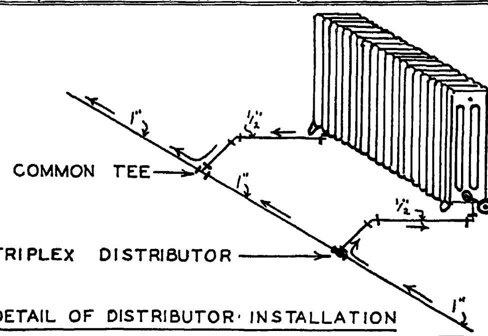
Main Pipe Size	Run	Outlet	Ship. Wt.
3/4"	3/4" x 3/4"	1/2"	1 lb.
1"	1" x 1"	1/2" or 3/4"	1 1/2 lbs.
1 1/4"	1 1/4" x 1 1/4"	1/2" or 3/4"	2 lbs.
1 1/2"	1 1/2" x 1 1/2"	1/2" or 3/4"	3 lbs.

IRON PIPE DISTRIBUTORS

Size	Tapping	No. per Box	Wt. per Box
3/4"	3/4" x 3/4" x 1/2"	6	9
1"	1" x 1" x 1/2" -1" x 1" x 3/4"	6	11
1 1/4"	1 1/4" x 1 1/4" x 1/2" -3/4" -1"	6	13
1 1/2"	1 1/2" x 1 1/2" x 1/2" -3/4" -1"	6	18
2"	2" x 2" x 1/2" -3/4" -1"	6	24

TRIPLEX DISTRIBUTOR

Circulating fittings for one pipe systems. Efficiency comparable to a two pipe system. Only one fitting per radiator, except where basement radiators are installed which require two Distributors. By closing a radiator valve the full area of the main is bypassed through the fitting to the balance of system.



DETAIL OF DISTRIBUTOR INSTALLATION

SAFE-T BOILER PLUG

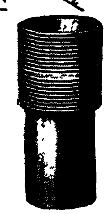
No. 36

Positive relief from dangerous pressures. Included with Flow Control Systems (complete), Tank-in-Basement Systems, No. 7 and 8 Control Units. Breakages of boiler from excessive pressure eliminated and protection assured.

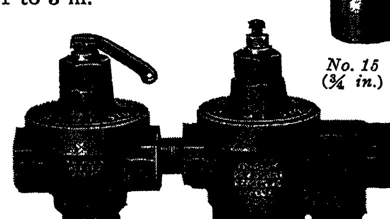
No.	Break. Pres.	Breaking Bars	Diaphragms	Dimensions Wd.	Ht.	Dia.	Ship. Wt.
36	40 lbs.	Wht. Iron	Lead	3 1/2	4 1/2	3 1/2	3 lbs.
37	75 lbs.	Wht. Iron	Lead	3 1/2	4 1/2	3 1/2	3 lbs.
38	100 lbs.	Wht. Iron	Lead	3 1/2	4 1/2	3 1/2	3 lbs
39	125 lbs.	Wht. Iron	Lead	3 1/2	4 1/2	3 1/2	3 lbs.

AIR ELIMINATOR

A positive method of permanently eliminating air from concealed copper or cast iron radiator after the system is vented the first time. Sizes 1 to 3 in.



No. 15 (3/4 in.)



TRIPLEX CONTROL UNITS

Easily installed. All water strained as it enters system. Strainer cast integral with Pressure Reducing Valve. Proper amount of water maintained in the system and the city water pressure reduced with the Pressure Reducing Valve. **The fast-filling feature will save the cost of the unit in time saved filling the system.** The Relief Differential Type Valve, seating under water and has lever for testing and flushing. Check is between Relief and Pressure Reducing Valves.

No. 7 CONTROL UNIT is a positive protection against boiler breakages from pressure. It consists of a No. 15 unit and No. 36 (40 lb) Safe-T Boiler Plug.

No.	Tapped	Body	Diaphragms	Springs	Dimensions Length	Ht.	Dia.	Ship. Wt.
15	3/4"	R. P. Iron	Bronze	Bz. & R. P. Stl.	10 3/8"	6 3/4"	3 1/4"	9 1/4 lbs.
7	3/4"	R. P. Iron	Bronze	Bz. & R. P. Stl.	10 3/8"	6 3/4"	3 1/4"	12 1/4 lbs.
14	1/2"	R. P. Iron	Bronze	Bz. & R. P. Stl.	9 3/4"	6 3/8"	3 1/4"	7 3/4 lbs.
14B	1/2"	R. Brass	Bronze	Bz. & R. P. Stl.	9 3/4"	6 3/8"	3 1/4"	7 lbs.
8	1/2"	R. P. Iron	Bronze	Bz. & R. P. Stl.	9 3/4"	6 3/8"	3 1/4"	10 3/4 lbs.
13	1/2"	R. P. Iron	Bronze	Bz. & R. P. Stl.	9 3/4"	6"	3 1/4"	7 1/2 lbs.
45	3/4"	R. P. Iron	Comp.	Bz. & R. P. Stl.	12"	7"	5"	10 1/2 lbs.

WARREN WEBSTER & COMPANY

Pioneers of the Vacuum System of Steam Heating

-since 1888 Webster Systems of Steam Heating

Main Office and Factory:
Camden, New Jersey

Representatives in Principal Cities—
Consult Your Local Phone Directory

Webster Nesbitt UNIT HEATERS

PRODUCTS AND SERVICES

Webster Systems of Steam Heating including Vacuum and Type "R" (vapor).

Webster Central Control Systems including HYLO and MODERATOR.

Modernization of Obsolete and Faulty Heating Systems.

Webster System Equipment including Light-Weight Concealed Radiation (Gravity Convection Heaters), Radiator Supply Valves, Metering Orifices, Thermostatic Traps, Drip Traps, Heavy Duty Traps, Dirt Strainers, Dirt Pockets, Boiler Return Traps, Vent Traps, Damper Regulators, Boiler Protectors, Lift Fittings, Expansion Joints, Separating Tanks, Steam and Oil Separators, Steam Vacuum Pump Governors, Air Separating and Receiving Tanks, Gages, Water Accumulators.

Webster Series "78" and Series "79" Traps for use at process pressures (10 to 150 lb per sq in.)

Webster-Nesbitt Unit Heaters and Residential Conditioners.

WEBSTER SYSTEMS

Webster Systems are low pressure, two-pipe systems of steam circulation *with the addition* of accurately-sized metering orifices at radiator supply connections and, when required, intermediate metering orifices at points in branch mains. Metering orifices effect even distribution of steam to all parts of the heating system and permit the successful application of a centralized control. Webster Valves are used at supply of radiators. Webster Thermostatic Traps prevent flow of steam into return mains when radiators are filled. Webster Drip Traps and Dirt Strainers are used where needed on steam mains. Webster Systems are available for vacuum, open return or "vapor" operation. The Type "R" System corresponds to the so-called Vapor type. Fig. 1 illustrates a typical arrangement of Boiler Return Trap, Vent Trap, etc., when low pressure boiler is the source of steam.

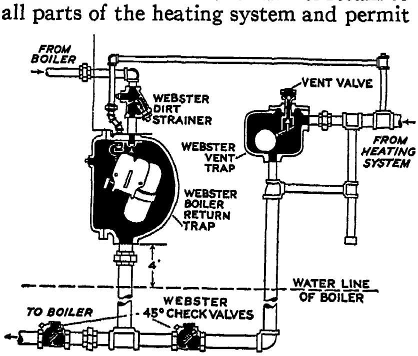


Fig. 1. Conventional arrangement of piping around Webster Basement Equipment for the Webster Type "R" System

WEBSTER CENTRAL CONTROLS

These are patented systems for varying the amount of steam to all radiators according to outside temperature. They provide *continuous heat delivery* with effective *fractional filling of radiators.* The Hylo Systems may be provided for manual control, or if desired, may be semi-automatic by incorporation of inside thermostat or thermostat and schedule clock. The Moderator Systems employ an automatic Outdoor Thermostat supplemented by a manual Variator.

The latter is used for quick heating-up, night load, and unusual weather or occupancy conditions. Use of Webster Central Control Systems results in (1) increased comfort because over-heating and underheating are minimized and (2) lower fuel or steam costs.

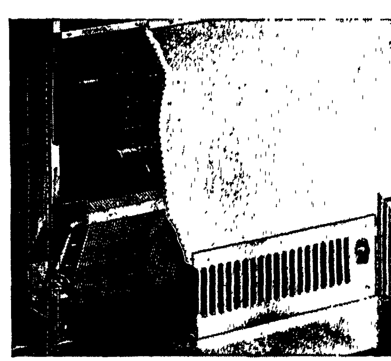


Fig 2. Webster System Radiation

WEBSTER SYSTEM RADIATION

Concealed, non-ferrous type for use exclusively with Improved Webster Systems. Is unique in that it combines *in a single unit*, a light-weight heating element of high efficiency with an orificed radiator supply valve, a radiator trap and supply and return piping connections. Metal enclosures for installation within the wall and exposed metal cabinets are available. Webster System Radiation and enclosures are so designed that the entire heating element can be quickly removed without damage to plaster or paint. Space requirements reduced to a minimum and installation greatly simplified.

RADIATOR SUPPLY VALVES

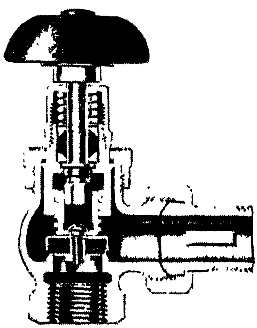


Fig. 3. Webster Type "WB-P" Valve

The new Webster Series 600 and Series 600S are supply valves of highest quality designed to eliminate the many sources of annoyances caused by valves of inferior design and quality. Webster Supply Valves open quickly and easily in less than a turn of the handle. They have non-rising stems. Steam can actually be shut off from radiators because they seat positively.

Type "WB-P" Valve (Series 600 P)—Uses a heavy Stainless Steel spring to automatically maintain pressure on the packing. Modulation sleeve furnished on special order at slight added cost. This type entirely suitable for hot water heating service; furnished with or without leak hole as desired.

Type "WB" Valve (Series 600)—Has high quality features of Type "WB-P" valves but differs in that it uses a screwed packing gland.

Sylphon Packless Valve (Series 600S)—Same features as Type "WB" except for genuine seamless Sylphon Bellows to completely encase valve stem. Meets fully "bellows packless" specifications. Modulation sleeve is standard equipment for ½, ¾, and 1 inch sizes. Not suitable for hot water heating service.

Bodies and Handles—Angle Body is made in ½, ¾, 1, 1¼ and 1½ inch sizes; right corner, left corner, straightway (both single and double union) bodies in ¾ and 1 inch sizes.

The ¾ in. size is available with ½ in. spud also. Choice of wheel, lever, lockshield, chain wheel, and extended stem handles.

Pressures—For low pressure vapor and vacuum steam heating service. Maximum pressure for Series 600S Sylphon Packless Valve, 20 lb per sq in.; for Series 600P Type "WB-P" and Series 600 Type "WB" 75 lb per sq in. Other Webster Valves are available on order for higher maximum pressures.

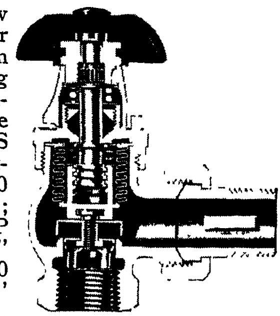


Fig. 4. Webster Sylphon Packless Valve

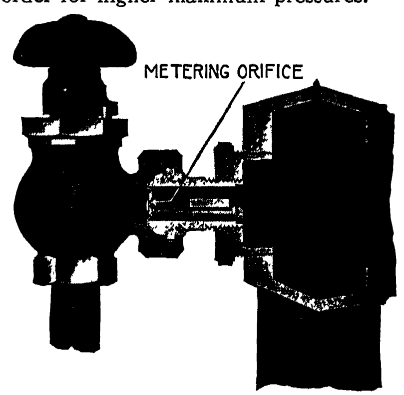


Fig. 5 Metering Orifice Inserted in Union Connection of a Webster Supply Valve. Other types are available.

Metering Orifices—Accurately sized and made of heavy gage Monel Metal to resist erosion and corrosion, amply thick to be free from vibration and shaped for quiet operation.

RETURN TRAPS

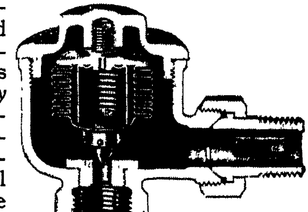


Fig. 6. Webster 502 Sylphon Trap

Sylphon—Perfected thermostatic bellows trap, *fully compensated for pressure*. Stainless steel valve piece and renewable seat. Factory adjusted. Made in angle, right-corner, left-corner, vertical and straightway bodies. Sizes: ½, ¾ and 1 in. For low pressure

vapor and vacuum steam heating service. Maximum pressure 25 lb per sq in.

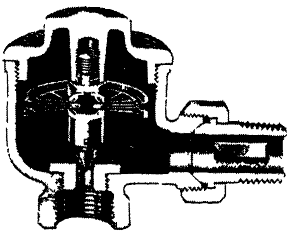


Fig. 7. Webster Size 702-M Trap

Series "7M"—Perfected diaphragm-type thermostatic trap, *fully compensated for pressure.* Uses Monel Metal diaphragm, Stainless Steel valve piece and seat insert. Renewable seat. Factory adjusted. Made in angle, right-corner, left-corner, vertical and straightway bodies. Sizes: ½, ¾ and 1 in. For low pressure vapor and vacuum steam heating service. Maximum pressure 25 lb per sq in.

Series 7 with phosphor-bronze diaphragm, brass valve piece and seat is also available.

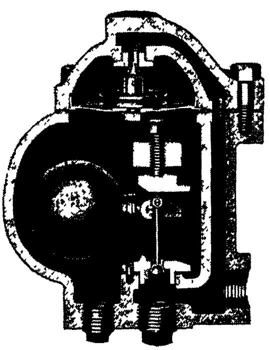


Fig. 8. Webster Size 0026-T Drip Trap will handle 1100 lb Water per Hour at 2 lb Pressure Difference

Series "26"—A heavy duty trap for drips of mains, blast radiation, unit heaters, hot water generators and similar applications. A rugged float-type trap available with and without thermostatic air vent. Made in six sizes: 400, 1100, 1600, 3000, 5000 and 11,700 lb water per hour at 2 lb pressure difference. Maximum working pressure is 15 lb per sq in.

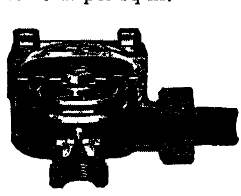


Fig. 9. Webster Size 782 Trap

Series "78" — thermostatic trap *built for* process steam pressures (10 to 150 lb per sq in.). Monel Metal diaphragm. Stainless Steel valve piece and seat insert. Angle model only. Sizes: ⅜, ½, ¾ and 1 in. Extensively used with laundry, cooking, sterilizing and other process-steam uses.

Series "79"—For use where large volumes of very hot condensate form more quickly than can be discharged by thermostatic traps alone. Float and thermostatic traps designed for normal working pressures between 15 and 150 lb per sq in. Water of condensation is passed through a float-controlled seat opening while air is discharged into the return piping by a thermostatically controlled vent. Compact and light in weight. Can be readily mounted in a pipe line without other support. Available with either ¾ in. or 1 in. inlet and outlet.

Cast iron body, copper asbestos gasket and cover bolted together with steel cap screws. Monel Metal valve piece and stem. Stainless steel seat. Air vent unit is Monel Metal diaphragm with Stainless Steel valve piece and brass seat with Stainless Steel insert.

DIRT STRAINERS AND POCKETS

Placed in return lines of steam heating systems to prevent dirt, rust and scale from impairing tightness of traps.

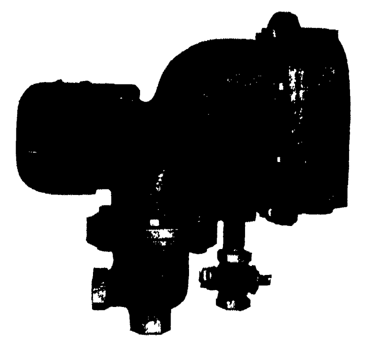


Fig. 10. Size 34C-1 Webster Boiler Protector with Low Water Electrical Cut-out Switch. Size 34 has no Cut-out Switch

BOILER PROTECTOR

Prevents breakage in low pressure heating boilers when water level becomes inadequate. Automatically supplies raw water to boiler when water level drops to 1 in. above bottom of gage glass.

For maximum boiler pressure of 15 lb per sq in. Maximum cold water main pressure should not exceed 150 lb per sq in.: minimum must not be less than 25 lb per sq in.

Made with ¾ in. connections, with or without electrical cut-out switch.

WEBSTER-NESBITT UNIT HEATERS

Are manufactured by John J. Nesbitt, Inc., Holmesburg, Philadelphia, Pa., and are distributed solely through Warren Webster & Company, Camden, New Jersey. Designed to circulate large volumes of air at comparatively low temperatures, assuring quick heating.

Ratings of Webster-Nesbitt Unit Heaters are based on tests made in accordance with standard test code of Industrial Unit Heater Association and A.S.H.V.E.

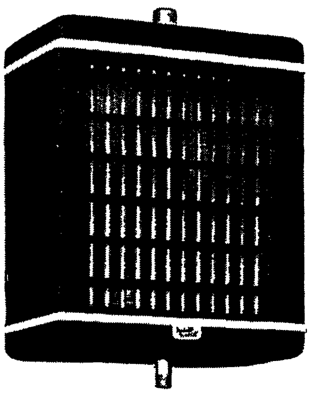


Fig. 11. Standard Propeller-Fan Type

PROPELLER-FAN UNIT HEATERS

Newly designed twenty-one unit models now available giving a wide selective range. Quiet large area blade fans. Rubber-isolated motors; single or multispeed. Compact suspended type. Sturdy casings. Modern styling. Catalog W-N 111.

GIANT UNIT HEATERS

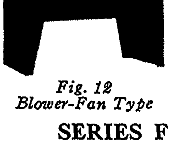


Fig. 12 Blower-Fan Type

Blower-fan type for economical heating of large areas. Floor-mounted, wall-mounted, ceiling-suspended, from 109,000 Btu, 2620 cfm to 1,008,000 Btu, 16,000 cfm with 2 lb steam, 60 degrees entering air. With or without Thermadjust Temperature Control Damper. Catalog W-N 104.

SERIES F UNIT HEATERS

Fig. 13 Series F Unit Heater

Centrifugal fan type for lobbies, showrooms, offices where quietness and appearance count. Four casing sizes with two radiator sizes available for each casing. Floor or ceiling mounting. Publication W-N 105.

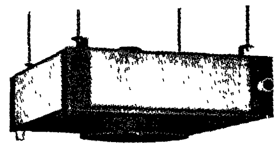


Fig. 14 "Little Giant" Down Blow Type

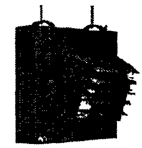


Fig. 15. Horizontal Blow Type

LITTLE GIANT UNIT HEATERS

New, light, compact, draw - through, high-velocity units available in down blow and horizontal blow models. 39,200 Btu, 710 cfm to 625,000 Btu, 9150 cfm.

Down Blow Type—In general, indicated when the presence of cranes and other machinery requires that the unit and piping be located well above the floor level.

Horizontal Blow Type—Application follows principles of heat distribution regularly employed in suspended blower fan type heater. Units are located closer to working zone than in the case of the Down Blow type. See Publication W-N 109.

RESIDENTIAL CONDITIONERS

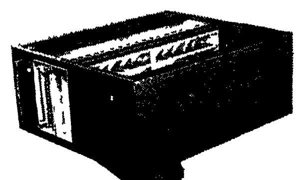


Fig. 16 *Series R*

Series R—For large and small residences. In two-section combinations for winter heating only with steam or hot water systems; or for heating, air cleaning and humidification (with a trouble-free cascade-type of humidifier) for summer cooling and dehumidification with cold water. Eight basic sizes: 750 to 4000 cfm.

Fig. 17 Series D

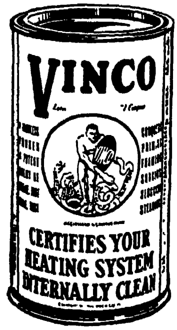
Fig. 18 Series A

Series D—Simplified compact unit heating systems for apartments and multiple residences. Available with steam or hot water heating element. Three sizes: 300, 450, 600 cfm.

Series A—For apartments. All the newly designed features of Series D Units plus (in a little added space) air cleaning; winter humidification, and summer cooling with cold water. Four sizes: 350 to 1000 cfm. Send for Publication W-N 107.

The Vinco Company, Inc.

305 East 45th Street — **New York, N. Y.**



Boiler Cleanser
3, 5 and 10 lb cans

VINCO BOILER CLEANSER

A positively harmless insoluble powder cleaner for new, remodeled and old heating systems. A unique, scientifically processed compound on a special formula not to be confused with other powder boiler cleaners.

What Vinco Boiler Cleanser Does

VINCO removes oil, grease, scale, rust and dirt from the internal surfaces and from the boiler water *without the labor, expense, and uncertain results of blowing boilers over the top or of wasting returns.*

By this thorough cleaning Vinco prevents or cures foaming, priming, surging, and slow steaming.

How Vinco Boiler Cleanser Works

Each minute grain of VINCO powder adsorbs several times its own weight of oil, rust and dirt. These larger grains of adsorbed impurities then settle and are drained through the bottom according to directions on each can.

Our Guarantees

1. VINCO contains no potash, lye, soda of any kind, oil, acid, or other harmful ingredients.
2. Purchase price is refunded if results are not as claimed when VINCO has been used according to directions.

Rust Preventer
1 qt cans only

VINCO RUST PREVENTER

When used after VINCO Boiler Cleanser has removed oil, grease, rust, scale and dirt, it will keep the rust inhibiting factors at the optimal constant for a year or more. (Testing kit below has complete instructions and chart.)

Vinco Testing
Kit No. 10
(Patent applied for)

VINCO TESTING KIT No. 10

for Testing Heating Boiler Waters

The kit enables the layman to make simple, rapid tests to diagnose and prescribe correct treatment of boiler waters right on the job.

A new time saving method that permits valid conclusions heretofore requiring complicated and often lengthy laboratory analysis and technique.

Each kit has sufficient material for complete tests on 100 jobs.

Refills cost about 2 cents per test.

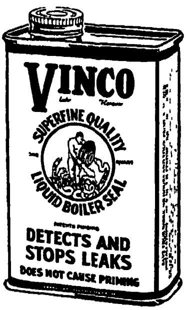
Soot-Off—1 lb cans 50 and 100 lb drums

VINCO SOOT-OFF

Safely and thoroughly removes the insulating blanket of soot on fire pot, flues and chimney. It also insures against external corrosion (caused by dampness and soot forming sulphuric acid during summer layoff.) No dangerous chemicals.

VINCO SUPERFINE LIQUID BOILER SEAL

A different liquid seal. Unique in that it does not induce priming and foaming. It has no unpleasant smell. Makes speedy and permanent repairs of boiler and heating system leaks. Fine to tighten up new jobs. Directions simple.



Liquid Boiler Seal 1 qt. cans only

Quantities

Steam and Vapor Systems—Use 1 quart VINCO Liquid Boiler Seal to each 6 sq. ft. *grate* area.

Hot Water Systems—Use 2 quarts VINCO Liquid Boiler Seal to each 6 sq ft *grate* area.

SPECIFICATIONS FOR COMPLETE VINCO
Treatment of New or Remodeled Steam or Vapor Heating Systems

Do not use as a cleaning agent soda or any alkali, vinegar or any acid. Use Vinco.

1. After the system is tested and tight, use the proper quantity of Vinco listed. After this first clean-out of any new or remodeled heating system, Vinco Boiler Cleaner need be used only if more piping, radiation, or another boiler is added to the original installation.

2. After using Vinco Boiler Cleaner, Vinco Field Test Kit should be used to determine and apply the proper quantity of Vinco Rust Preventer. Vinco Rust Preventer should be applied annually or whenever the boiler water is drained for necessary repairs to the system.

QUANTITIES OF VINCO (IN POUNDS) REQUIRED FOR HEATING SYSTEMS

(Note that quantities are based on actual installed radiation, not on boiler capacity.)

Sq Ft of Radiation	For Steam or Vapor Systems, to prevent or cure priming or foaming. Also for Hot Water Heating Systems maintained at approx. 200 F or above.	Annually, to remove rust, scale, dirt and for Hot Water Systems below 200 F.
up to 350	3	1½
351 " 600	5	2½
601 " 1100	8	4
1101 " 1400	10	5
1401 " 1800	13	6½
1801 " 2100	15	7½
2101 " 2700	18	9
2701 " 3100	20	10
3101 " 3700	23	11½
3701 " 4200	26	13
4201 " 4600	28	14
4601 " 5000	30	15
5001 " 5300	31	15½
5301 " 5600	32	16
5601 " 5900	33	16½
5901 " 6200	34	17
6201 " 6500	35	17½
6501 " 6800	36	18
6801 " 7100	37	18½
7101 " 7400	38	19
7401 " 7700	39	19½
7701 " 8000	40	20
8001 " 8300	41	20½
8301 " 8600	42	21
8601 " 8900	43	21½
8901 " 9200	44	22
9201 " 9500	45	22½
9501 " 9800	46	23
9801 " 10100*	47	23½

*Above 10100 sq ft use an additional pound Vinco for each additional 300 sq ft of actual installed radiation.

SPECIFICATION FOR OLD HEATING SYSTEMS THAT DO NOT PERFORM PROPERLY

Diagnose and treat according to Vinco Field Test Kit.

SPECIFICATION FOR HOT WATER SYSTEMS

Use half quantities listed for treatment of steam systems to remove impurities. Then use test kit to determine proper quantity of Vinco Rust Preventer.

REMOVE SOOT WITH VINCO SOOT-OFF SEVERAL TIMES A YEAR

McDONNELL & MILLER

Safety Devices for Steam and Hot Water Boilers and Liquid Level Controls

General Offices: Wrigley Building, **Chicago, Ill.**

"Doing one thing well"

PRODUCTS:

Boiler Water Feeders; Feeder-Cut-off Combinations; Low Water Fuel Cut-offs; Pump Controls, Low Water Alarms; Humidifier Water Level Controls; Safety Relief Valves for hot water heating boilers and storage tanks; Liquid Level Controls for a wide range of services.

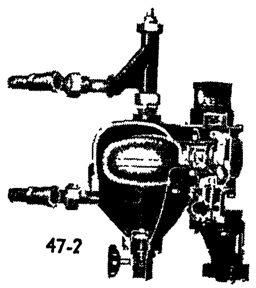
Boiler Water Feeders—McDonnell boiler feeders protect steam boilers from low water by automatically supplying water to the boiler when and as needed. See illustrations and service data opposite.

Feeder Cut-off Combinations—For automatically fired boilers the No. 2 Low Water Cut-off Switch is added to form a feeder-cut-off combination, like the No. 47-2, No. 51-2, etc. In such a combination the feeder takes care of the normal water requirements. In case of an emergency, such as excessive priming and foaming or failure of the pump, the low water cut-off switch stops the burner until emergency is passed. Electrical ratings of No. 2 Cut-off Switch: A.C.—¾ hp, 110-220 V.; D.C.—10 amp, 125 V.

Low Water Fuel Cut-offs—If the feeder-cut-off combination is not desired, the No. 67 alone can be installed to dependably stop the burner when low water threatens. Has two switches—one operates alarm or controls No. 101 Electric Feeder, other acts as low water cut-off. Rating (each switch): A.C.—¾ hp, 115-230 V; D.C.—¼ hp, 115 V.

For high pressure jobs the No. 150 will serve not only as a low water fuel cut-off but also as a pump control and low water alarm—for pressures as high as 150 lbs. Electrical ratings, Cut-off and Pump Control: A.C.—1 hp, 110-220 V; D.C.—½ hp, 115-230 V. Alarm: A.C. or D.C.—1 amp, 110 V. Specify 150-M for manual reset low water cut-off.

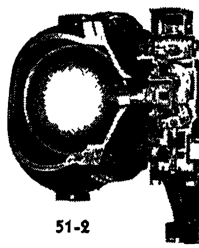
Advanced Features—A notable advance in No. 47 and 67 is the deep sediment chamber, with large capacity, straight-through (A.S.M.E. Standard) blow-off valve. Other features of feeders and cut-offs include: Quick-Hook-Up; Cool feed valve; finer stainless steel valves; large area built-in strainers; double switch construction of the No. 67; electric boiler water feeders; self-cleaning built-ins.



47-2

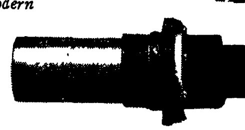
No. 47-2 Feeder Cut-off Combination. For automatically fired boilers up to 5000 sq ft capacity—maximum steam pressure 25 lbs. No. 47 is for hand fired boilers—same service range, without switch. For process boilers up to 5000 sq ft, with pressures up to 35 lbs use No. 147 (Hand Firing) or No. 147-2 (Automatic Firing).

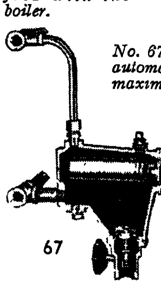
No. 51-2 Feeder-Cut-off Combination For automatically fired boilers above 5000 sq ft—maximum steam pressure 35 lbs. No. 51 (without switch) for hand fired boilers. For pressures from 35 to 75 lbs use the No. 53 (Hand Firing) No. 53-2 (Automatic Firing).



51-2

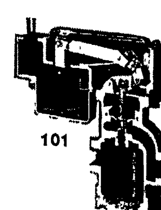
"Built-in" Low Water Fuel Cut-offs—chosen as standard equipment on modern jacketed boilers. Self cleaning to insure dependable operation. Should be specified with the boiler.





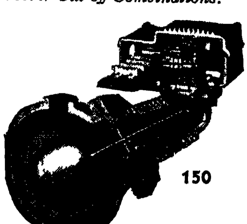
67

No. 67 Low Water Fuel Cut-off. For automatically fired boilers of any size; maximum steam pressure 25 lbs.



101

No. 101 Electric Water Feeder for use with No. 67 or Built-in cut-offs—converts into Feeder-Cut-off Combinations.



150

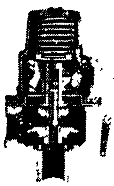
No. 150 Combination Pump Control, Low Water Fuel Cut-off and Alarm. for steam pressures up to 150 lbs. Has two switches: one controls pump—other stops burner and completes alarm circuit when water level falls to danger zone.

McDonnell Snap Action Safety Relief Valves

McDonnell Safety Relief Valves are the first to be rated in Btu capacity—in ability to dissipate heat at their set relief pressure. Their "snap action" design was inspired by exhaustive research which proved that the only proper solution to the problem of preventing explosions and losses of hot water boilers, domestic hot water heaters, and hot water tanks was to be found in a valve that would have sufficient discharge capacity at relief pressure to prevent further pressure rise when the boiler or domestic water heater is operated at maximum gross Btu output.

The series includes Nos. 29 and 129 for hot water heating boilers and the Nos. 229, 329 and 429 for domestic hot water heaters and tanks. The snap action mechanism (Pat. No. 2,248,807) provides, for the first time, a precision-built means of opening the valve wide at set pressure. Revolutionary features are hardened stainless steel cone instead of composition disc; long lived bellows diaphragm; remarkable ease of testing; complete protection of working parts; many other refinements.

No. 29 and No. 129 Safety Relief Valves for Hot Water Heating Boilers

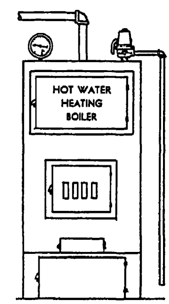


No. 29-129

—are rated in Btu capacity so that they can be matched to the gross Btu output of the boilers on which they are used:

No. 29 for heating boilers with gross heat output up to 156,000 per hour.
No. 129 for heating boilers with gross heat output up to 350,000 per hour.

Set relief pressure of both No. 29 and 129 is 29 lbs. When used in accordance with their ratings they will prevent pressures over 29 lbs under *all conditions*—even such an emergency as a bottled up system with all temperature-limiting devices inoperative and heat input at maximum. This is "safety the McDonnell Way."



Installation No. 29-129

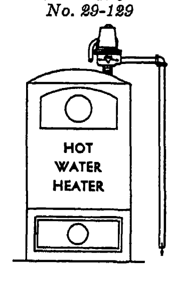
No. 229-329-429 Safety Relief Valves for Domestic Hot Water Heaters and Tanks

No. 229 329-429

Engineering fundamentals, confirmed by practical tests prove that there is just one simple rule to observe in protecting domestic hot water heaters and tanks: Keep the pressure below the maximum allowed by the manufacturer of the tank or heater and there will be no failures or explosions.

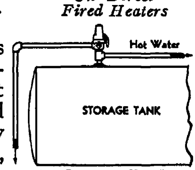
To accomplish this, a relief valve must have capacity to dissipate the maximum heat to which the tank can be subjected. This means that it must be Btu-rated, just as for a hot water boiler, so that it can be matched to the service condition. Nos. 229, 329 and 429 are so rated:

No. 229 is for water supply pressure up to 50 lbs; handles approximate Btu output of 316,720. Inlet tapping 1 in.; outlet tapping ¾ in.
No. 329 is for supply pressure up to 75 lbs; handles Btu output of 380,896. Inlet tapping 1 in.; outlet tapping ¾ in.
No. 429 is for supply pressures up to 100 lbs; handles Btu output of 432,590. Inlet tapping 1 in.; outlet tapping ¾ in.



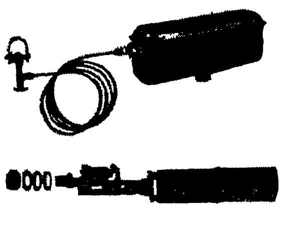
On Direct Fired Heaters

Never forget the pressure control—not temperature control—is the fundamental safety measure. You can have pressure and breakage without excess temperature, but you can't have an explosion at any temperature, unless you have excess ***pressure***. No. 229, 329 and 429 are the first valves to be built and rated in such a way that they will prevent excess pressure under *all* conditions—assuming, of course, that their Btu rating is properly observed.



On Storage Tank

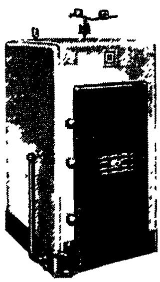
McDonnell No. 217 Humidifier Water Control



• *No. 217. Complete with float chamber, tubing and saddle valve.*
• *No. 117. Same as 217, omitting tubing and saddle valve.*
• *No. 17. Float and valve only.*

—automatically maintains proper water levels in evaporation pans of warm air furnaces. New snap action eliminates tendency of former controls to become stopped up by foreign matter or to stick and become inoperative. This valve has only two positions—tight closed and ***wide open***. When water falls ¼ in. in pan it ***snaps*** wide open feeding a full stream that flushes orifice. Closes leak-tight against any water pressure up to 150 lbs.

AMERICAN RADIATOR & Standard Sanitary
New York CORPORATION *Pittsburgh*

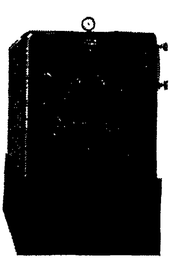


SEVERN BOILER FOR COAL (stoker or hand-fired), OIL OR GAS

An exceptionally efficient Boiler with many new features for convenience and economy. Ratings: Steam—350 to 780 sq ft, Water—560 to 1250 sq ft, installed radiation.

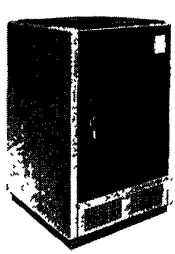
OAKMONT OIL BOILER

A highly efficient moderate priced Boiler for small homes. Also supplied as complete boiler-burner unit with Arcoflame Burner. Ratings: Steam—390 to 810 sq ft, Water—625 to 1295 sq ft, installed radiation.



IDEAL ARCOFIRE STOKER-BOILER

Extra efficient, extra economical—especially designed for automatic stoker operation only. Ratings: Steam—900 to 1,775 sq ft, Water—1,440 to 2,840 sq ft, installed radiation.



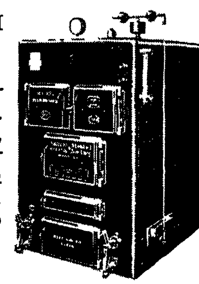
"EMPIRE" GAS BOILER

Designed by experts to burn gas efficiently, economically. All controls concealed. Ratings: Steam—163 to 1097 sq ft, Water—135 to 1755 sq ft, installed radiation.

ONE FAMOUS NAME

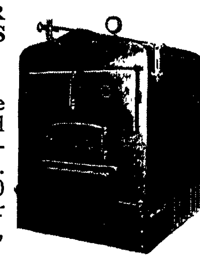
One name and one responsibility backs the complete line of American Heating Equipment . . . the line that is complete for every heating need. It includes Boilers of every type—for hand or stoker-fired coal, oil or

IDEAL REDFLASH BOILERS (All Fuels)



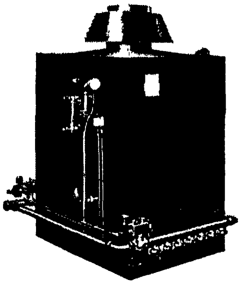
Economical heat for any size building. Attractive red jacket, fully insulated. Four sizes. Ratings: Steam—770 to 9,900 sq ft, Water—1230 to 15,840 sq ft, installed radiation.

IDEAL WATER TUBE BOILERS (Oil or Stoker Fired)



For medium to large size buildings. Noted for efficient performance. Three sizes. Ratings: Steam 650 to 4600 sq ft, Water—1040 to 7360 sq ft, installed radiation.

STANDARD GAS BOILER



Basically the same as the "Empire" Gas Boiler shown at left, but without jacket. Ratings: Steam — 400 to 11,905 sq ft, Water — 135 to 19,050 sq ft, installed radiation.

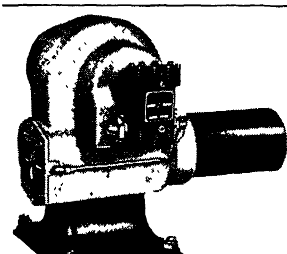
AMERICAN RADIATOR & STANDARD SANITARY

New York **CORPORATION** *Pittsburgh*

FILLS EVERY HEATING NEED

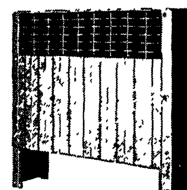
gas. It also includes Radiators, Convectors and Enclosures for every type of structure, as well as Accessories, domestic Water Heaters, Oil Burners and Oil Heating Units . . . all products of the world's largest Heating and Plumbing organization.

AMERICAN HEATING EQUIPMENT
COSTS NO MORE THAN OTHERS



ARCOFLAME OIL BURNERS

The Model "C" Arcoflame has a capacity of up to 3 gallons per hour. The Model "L" (not shown) from 3 to 7 gallons per hour. Both embody unusual and highly efficient features.



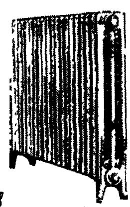
Sunrad Radiator

SUNRAD RADIATOR

Streamlined and modern, the Sunrad is one integral unit and supplies both radiant and convected heat.

ARCO RADIATOR

Slim, space-saving and highly efficient, the Arco Radiator comes in four narrow widths and in five heights.



Arco Radiator

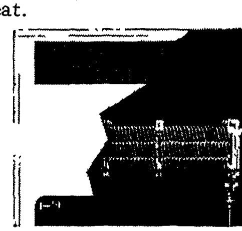
Arco Multifin Convector

ARCO MULTIFIN CONVECTOR

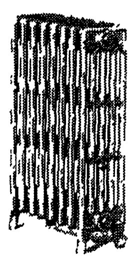
Non-ferrous. Highly efficient. For all systems except one-pipe steam. Available in five widths.

ARCO CONVECTOR

For convection heating at its best. Available in four widths and in virtually any desired length.



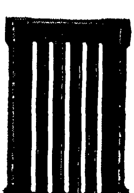
Arco Convector



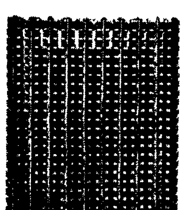
CORTO RADIATOR

The Corto is the original thin tube radiator. It is available in six heights and four widths.

Corto Radiator

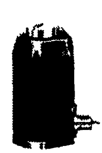


Peerless Wall Radiator

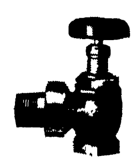


Vento Cast Iron Blast Heater (for fan and blower work)

No. 861 Arco Detroit Hurivent Valve (for main)



No. 300 Arco-Detroit Multiport Valve (for radiators)



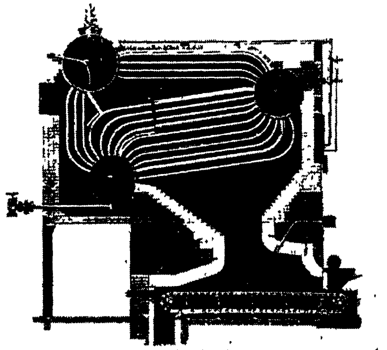
No. 999 Arco Packless Steam Radiator Valve

Type H Stirling Boiler

The Babcock & Wilcox Type H Stirling Boiler is a highly efficient unit built for moderate pressures at moderate prices.... and is designed to occupy minimum floor space and head room for the heating surface required.

This boiler is built in four classes and 36 sizes ranging from 691 to 6225 sq ft of heating surface, and can be designed for operation with any fuel and every method of firing.

The moderate price is due only to the simplicity of design, efficient production methods and superior shop equipment.



Type H Stirling Boiler with Babcock & Wilcox Chain-Grate Stoker

Advantages of the Babcock & Wilcox Type H Stirling Boiler:

Unusual steaming capacity for the floor space and head-room required.

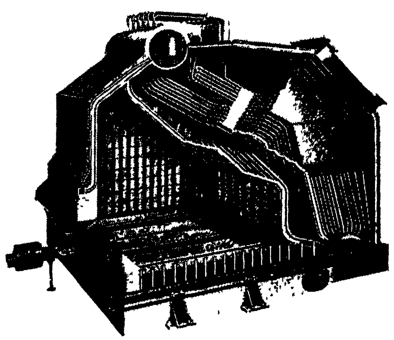
The choice of three locations for gas exit reduces cost of flues and breeching.

Distribution baffles make effective all of the heating surface.

Tube renewal is facilitated by correct tube spacing, and a tube removal door.

The boiler is supported by a structural-steel framework entirely independent of the brickwork.

A complete table of sizes and dimensions will be sent upon request. Simply ask for Bulletin G-8-C.



B&W Integral-Furnace Boiler, Type FF

Many of the advantageous features incorporated in large B&W central-station boilers are now available for the first time in the B&W Integral-Furnace Boiler, Type FF, which is offered in sizes ranging from 1353 to 6506 sq ft heating surface.

Distinguishing features include:

A completely water-cooled furnace. The construction provides water cooling for front and rear (or bridge) walls, as well as side walls and roof.

A furnace arrangement in which the primary combustion zone is followed by an open pass, thus making use of a principle of combustion that was first developed and used successfully in the B&W Open-Pass Boiler for central stations. This design insures mixing of the gases while at high temperatures, thereby aiding efficient and smokeless combustion.

Cyclone Steam Separators, which provide dry steam at high boiler-water concentrations independently of normal variations in water level, and increase circulation by eliminating steam from the water.

These, with related features, result in a boiler that is outstanding for economy of fuel and maintenance and for ease of operation. Write for Bulletin G-34.

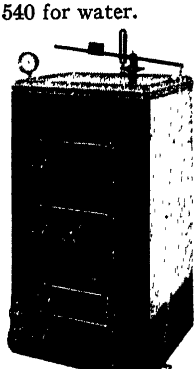
Burnham Boiler Corporation

Irvington-on-Hudson, N. Y.—Zanesville, Ohio

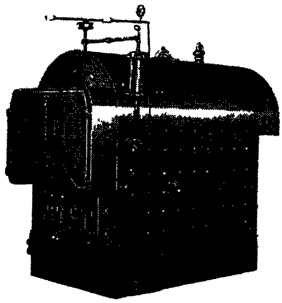
***There's a Burnham for Every Purpose*—Catalogs Sent on Request**

Yellow Jacket Boiler—All Fuel Convertible. 305 to 935 sq ft for steam and 490 to 1495 for water.

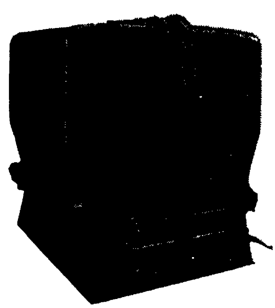
DeLuxe Gas or Oil Boiler—250 to 960 sq ft steam and 415 to 1540 for water.



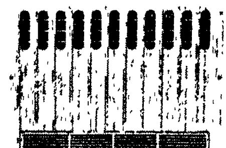
No. 1, 2, 3 and 36 in. Series—All Fuel. 230 to 4920 sq ft steam and 370 to 7880 for water.



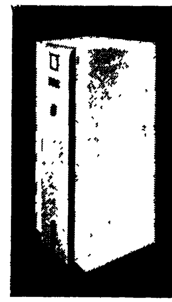
Welded Steel Boiler — Residence type with jackets. Capacities from 400 to 1750 sq ft steam and 640 to 2800 for water. Commercial type. Capacities from 1800 to 42,500.



50 Inch Twin Section—4500 to 14,600 sq ft steam and 7200 to 23,360 for water.

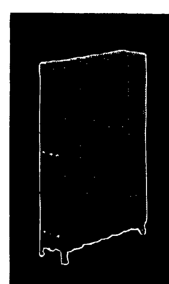


Cabinet Type Radiant Radiator — Two heights. 20 and 23 inches.



Junior Yello-Jacket—For Oil Only. 360 sq ft steam and 580 sq ft for water.

Round Sectional, All Fuel—275 to 830 sq ft steam and 440 to 1330 for water.



Burnham Slenderized Radiator—Made in 3 to 7 tubes in heights of 14 to 32 inches.

Crane Co.

BOILERS, RADIATORS, VALVES, FITTINGS, PIPE, STEAM SPECIALTIES, PLUMBING AND HEATING MATERIALS

General Offices: 836 South Michigan Avenue, **Chicago, Illinois**

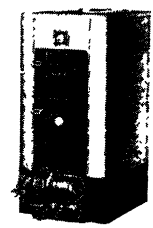
Nation-Wide Service Through Branches, Wholesalers, Plumbing and Heating Contractors

A complete line of heating equipment—boilers and furnaces for coal, coke, oil, or gas burning—for steam, hot water, or warm air systems. Full descriptions and specifications are given in your Crane Catalog—or supplied on request.

BOILERS FOR SMALL HOMES

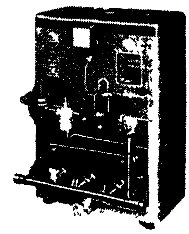
SERIES FOURTEEN

Wet base; low return inlet. Patented controlled water travel. Large ceiling heating surface. Internal heater and jacket optional. For steam or hot water. Capacities: manual firing, up to 90,000 Btu., oil or stoker up to 119,000 Btu. (IBR).



CONSERVOIL UNIT

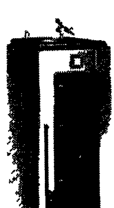
Low-priced boiler-burner unit in 4 sizes up to 131,000 Btu. (IBR) Controlled water travel, large ceiling surface, and flue inserts assure fuel economy. Includes burner, draft regulator and 3 controls. For steam or hot water.



No. 2WG BASMOR GAS BOILER

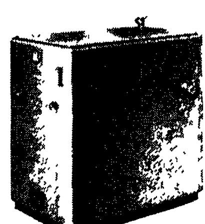
New hot water boiler for smallest homes. Sections are cast-iron with water-jacketed combustion chamber. Fully automatic. Shipped completely assembled; housing, controls in position. Up to 110,800 Btu. net capacity.

BOILERS FOR AVERAGE-SIZE HOMES



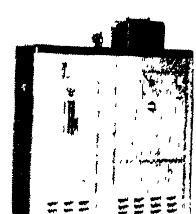
No. 10 ALL-FUEL BOILER

Can be installed for manual firing—easily converted for stoker, oil or gas firing. High base and removable grate lugs give ample space for stoker or oil burner. Provision for internal heater. For steam or hot water. Net capacity up to 207,000 Btu. (IBR).



No. 16 SUSTAINED HEAT BOILER-BURNER UNIT

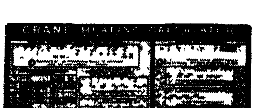
Application of Crane sustained heat principle extracts more heat from fuel. Down-draft flue construction prevents escape of combustion gases before heat has been absorbed. Net capacity up to 216,000 Btu. (IBR). Steam or water.



No. 25 BASMOR GAS BOILER

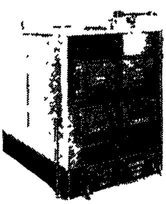
Unusual efficiency obtained with staggered fin construction and improved Bunsen-type burners. Safe, can't back-fire. Simple controls. Many sizes; for manufactured and natural gas. Net capacity to 177,400 Btu. Steam or hot water.

CRANE HEATING CALCULATOR FREE



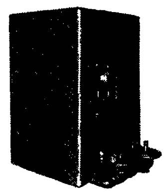
With this accurate calculator, employing the A.S.H.&V.E. method of determining heat losses, you can quickly select the right boiler and radiator requirements for any job. Easy to use—slide rule type. Free on request. Please write on your letterhead to address at top of this page.

BOILERS FOR LARGER BUILDINGS



No. 4 SECTIONAL BOILER

For manual, oil, and stoker firing. Up to 1,756,800 Btu. net capacity—steam or hot water.

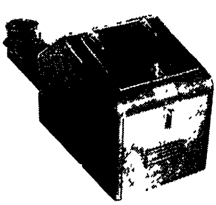


SERIES 60 BASMOR GAS BOILER

Has built-in steam header with Hartford return loop. Up to 2,857,200 Btu. net capacity—steam or hot water.

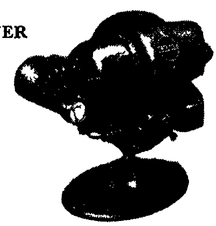
AUTOMATIC HEAT-CONVERSION UNITS

AUTOCOAL STOKER



For even, controlled room temperature with minimum attention. Hopper models: 20 to 350 lb. per hour capacity. 35 and 50 lb. bin-feed models.

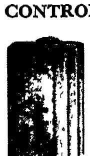
CONSERVOIL BURNER



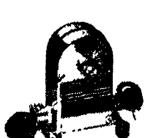
Will burn lower grades of fuel oil. Only one moving part. Quiet; cannot foul. Models up to 25 gal. per hour capacity.

CONTROLS

Low Voltage Relay-Transformer



Room Thermostat



Draft Tender

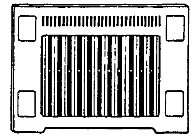
A full line of precision-built Crane controls including room thermostats, night set-back clocks, oil and stoker controls, limit switches for steam, hot water, and furnace systems.

The Crane line includes valves, fittings, and pipe for all boiler and radiator systems; a selection of furnaces for coal, oil, and gas; also split-system equipment and well-water cooling for year 'round air conditioning.

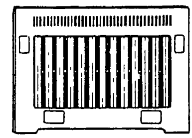
HEATING ELEMENTS—ALL TYPES

COMPAC SLIM-TUBE RADIATORS

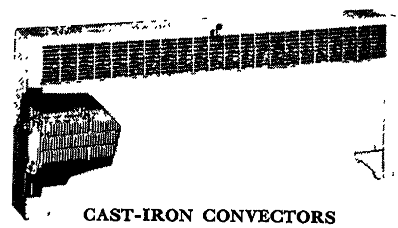
Cast-iron; space-saving. Modern slender design. For free-standing or recessed installation—with or without attractive front panel. Maximum delivery of radiant, infra-red ray heat. Further space-saving with bottom connections.



End Connection



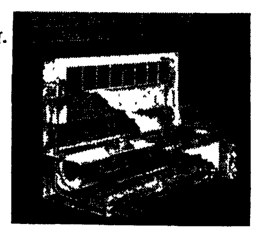
Bottom Connection



CAST-IRON CONVECTORS

Enclosures of heavy steel, smartly styled. Models for fully or partially recessed, free-standing, wall-hung, and plaster-front installations. Convectors of sturdy cast-iron with large integral fins designed to stimulate air flow. For all systems.

DUCTLESS WINTER AIR-CONDITIONING UNIT

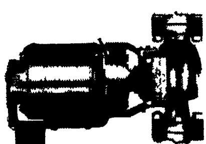


Recessed in wall and floor; no sheet metal work. Provides heat, humidification, filtering and circulation.

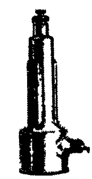
FOR LARGE SPACE HEATING REQUIREMENTS, SPECIFY CRANE SPEED HEATERS.

Made in a Complete Line.

HEATING SPECIALTIES



Circulating Pump



Steam Venting Valves

Crane supplies a complete range of hot water specialties including circulators, flow controls, monoflo fittings, pressure tank systems, indirect heaters. Also, air valves, traps, condensation and vacuum pumps, low-water cut-offs, and other steam specialties.

Fitzgibbons Boiler Company, Inc.

Established 1886

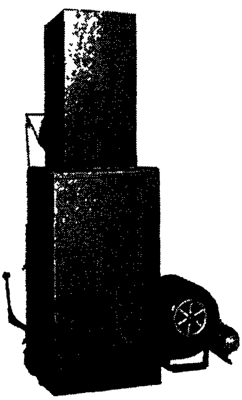
General Offices: Architects Bldg., 101 Park Avenue

New York, N. Y.

Works: OSWEGO, N.Y.

Branches and Representatives in Principal Cities

PRODUCTS—STEEL HEATING and POWER BOILERS for all fuels and all heating systems. Capacities to meet requirements of any building. Built and rated according to *S. H. B. I.* Code. —AIR CONDITIONERS for "Split-Systems" and for Direct-Fired installations in residences of all sizes.



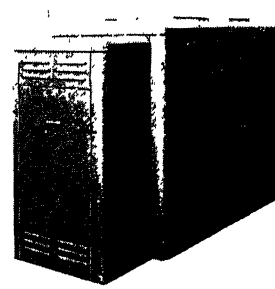
WARM AIR FURNACE 80 FWA

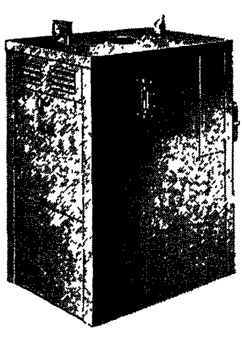
For hand firing with coal. Automatically controlled blower provides forced circulation of warmed air. Designed in accordance with the specifications of Procurement Division of the U. S. Treasury Department and of FWA, USHA, PBA and FSA for Defense housing. Fitzgibbons "Weldseal" construction positively insures against leakage of flue gases.

Bonnet capacity, 80,000 Btu based on Standard Code of National Warm and Air Conditioning Asso.

DIRECT-FIRED AIR CONDITIONERS

The DIRECTAIRE—The conditioner that has broken the shackles of traditional "hot air furnace" design, providing far greater Efficiency, Ruggedness, Quietness, Fuel Economy, Cleanability. Streamlined jacket in two types. Nine sizes—65,000 to 600,000 Btu at the bonnet.





FITZGIBBONS 400 SERIES STEEL BOILERS

The choice of architects and builders wherever low cost heating in small homes is needed. Beautifully adapted to defense housing using radiator heat with oil, gas or stoker firing, or with coal hand firing. Built-in coil provides domestic hot water. All the advantages of Fitzgibbons steel boiler construction in an attractively jacketed unit, priced for the field it serves. Five sizes—260 sq ft (steam or vapor, hand fired) to 1440 sq ft (Hot water system, mechanically fired.)

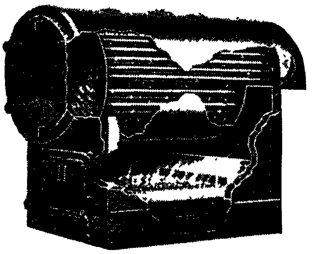
FITZGIBBONS "EIGHTY" SERIES STEEL HEATING BOILERS

The OIL-EIGHTY AUTOMATIC*—An outstanding residential steel boiler that teams up with any good rotary or gun type burner to form a highly efficient unit. Provides room for burner inside the jacket. Year-'round tankless domestic hot water optional. Ratings, Steam—12 sizes—425 to 2680 sq ft.

The GAS-EIGHTY—For gas. Jacketed. Ratings, Steam—12 sizes—425 to 2680 sq ft.

FITZGIBBONS R-Z-U- JUNIOR—For oil, stoker, coal hand firing. Auxiliary grate (optional) for refuse disposal and stand-by service. Tanksaver or Tankheater (optional) provides year-'round domestic hot water supply with or without storage tank. Ratings, Steam, hand fired type, 900 to 3200 sq ft. Oil or stoker fired, 1100 to 3900 sq ft.

*Reg. U. S. Pat. Office.

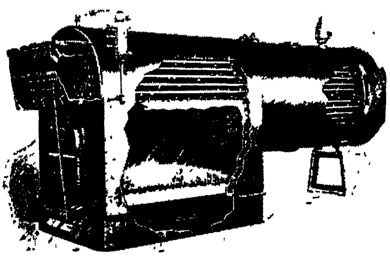


R-Z-U

FITZGIBBONS R-Z-U

Steel Firebox Boilers

The R-Z-U arranged for rear smoke outlet.
Built for 15 lb w.s.p.—*A.S.M.E.* Code.
Up-Draft Type............1800 to 35,000 sq ft steam
Oil, Gas, Stoker..........2190 to 42,500 sq ft steam
Smokeless Type..........1800 to 35,000 sq ft steam

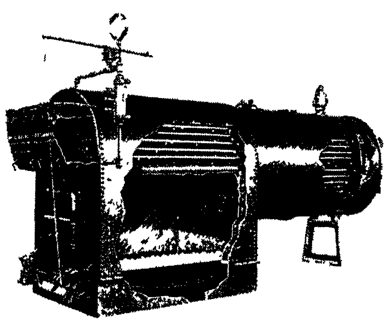


500 Series

FITZGIBBONS 500 SERIES

Portable Welded Firebox Boilers—Return Tubular

Built for 15 lb s.s.p.—*A.S.M.E.* Code.
Ratings, steam....3500 to 35,000 sq ft hand fired.
4250 to 42,500 sq ft mech. fired.
Oil, Gas, Stoker, and hand-fired types.

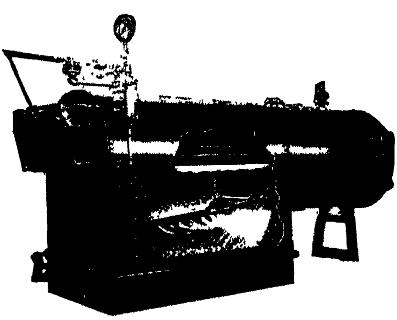


700 and "P" Series

FITZGIBBONS 700 AND "P" SERIES

Portable Riveted Firebox Boilers

700 Series for 15 lb w.s.p.—*A.S.M.E.* Code.
Ratings, steam—3500 to 35,000 sq ft hand fired.
4250 to 42,500 sq ft mech. fired.
"P" Series for 100 and 125 lb w.s.p.—*A.S.M.E.* Code.
Ratings, horsepower—25 to 250 hand fired.
30 to 261 mech. fired.
Oil, Gas, Stoker, and hand-fired types.



600 and 800 Series

FITZGIBBONS 600 AND 800 SERIES

Smokeless Down-Draft Riveted Firebox Boilers

600 Series for 15 lb w.s.p.—*A.S.M.E.* Code.
Ratings, steam—3500 to 35,000 sq ft hand fired.
4250 to 42,500 sq ft mech. fired.
800 Series for 100 and 125 lb w.s.p.—*A.S.M.E.* Code.
Oil, Gas, Stoker and hand-fired types.
Ratings, horsepower—25 to 250 hand fired.
30 to 261 mech. fired.

When this catalog went to press, all products and accessories described herein were available for sale. Government priorities or other circumstances beyond our control may now affect delivery. Consult the nearest Fitzgibbons Sales Engineer for up-to-date information.

Descriptive Bulletins on any or all of above boilers will be mailed on request.

Farrar & Trefts

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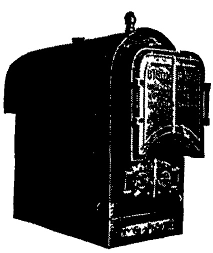
Seattle, Wash.
Toledo, Ohio
Washington, D. C.

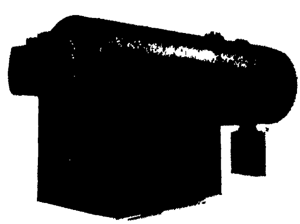


The Bison Compact

The **F&T Bison Compact Welded Heating Boiler** is more than just another boiler. It has been designed carefully so as to have a large furnace volume, the proper volume of water, just the right amount of steam liberating surface, the correct volume for steam storage and a balanced circulation. The result is a remarkably steady water line—**A Balanced Boiler.**

This boiler requires a minimum amount of floor space and is easy and inexpensive to install. It is reasonable as to first cost and economical in operation. Construction is in accordance with the *A.S.M.E.* Code for 15 lb working pressure and boilers are designed for hand firing with anthracite or bituminous coal or for mechanical firing with oil, gas or stoker. There are various sizes available from 1800 to 35,000 sq ft of steam radiation, all ratings as required by the *Steel Heating Boiler Institute.*

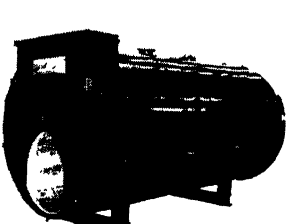
The **Bisonette Compact Boiler** has the same characteristics as the larger Bison Compact Boiler. It has been designed for installation in large residences and small business establishments where the advantages inherent in a **Steel** boiler are desired.



Firebox Return Tubular Boiler

Firebox Return Tubular Heating Boilers are **Quality Boilers.** They are constructed to measure up to the high standards set by Heating Engineers and will give unfailing service under all conditions. Being economical to install and operate, they are highly favored by Architects and Engineers for heating Schools, Hospitals, etc.

There are two types of Firebox Boilers, the Up-Draft Type and the Down-Draft Type. Both types are made of welded or riveted construction for heating purposes at 15 lb working pressure and riveted, or, Class 1 fusion welded x-rayed and stress-relieved for power purposes at 100, 125 and 150 lb working pressure in accordance with the *A.S.M.E.* Code. Sizes from 1800 to 35,000 sq ft of steam radiation, as rated by the *Steel Heating Boiler Institute,* are designed for hand firing with coal or for mechanical firing with oil, gas or stoker.



The **Bison Low Pressure Scotch Wet-Back Top Boilers** are *carefully proportioned* and *balanced.* They are designed for hand, oil, gas or stoker firing, for ratings from 15 to 250 hp. These boilers operate efficiently and carry sustained overloads. The *Front Smokebox Door Open Sideways* giving easy access to the tubes.

The *Wet-Back Top* increases the heating surface and steam disengaging area, thus adding to the capacity of these boilers. **F & T** boilers are designed so that the round furnace is always longer than the tube length which increases the furnace volume. This gives a large combustion volume in proportion to horsepower rating which makes the boilers very *economical to operate* and exceedingly "*Quick Steamers.*"

The International Boiler Works Company

East Stroudsburg, Pa.

"Fuel Saver" Water Tube Steel Heating Boilers

SALES OFFICES

Alaska	Jackson, Miss.		Scranton, Pa.
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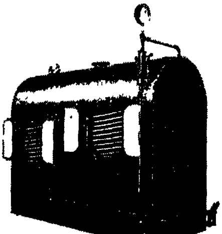
Detroit, Mich.	New Haven, Conn.	Rochester, N. Y.	Washington, D. C.
E. Stroudsburg, Pa.	New York City, N Y.	St. Paul, Minn.	White Plains, N. Y.
Harrisburg, Pa.	North Olmsted, O.	Salt Lake City, Utah	Wilmington, Del

International "FUEL SAVER" Water Tube Steel Heating Boilers offer the same quick steaming and economy that have long been accepted as most efficient in marine and industrial service. "FUEL SAVER" Water Tube Boilers are available for large and small heating requirements in a wide range of types and capacities.



TYPE C "FUEL-SAVER" WATER TUBE STEEL HEATING BOILERS

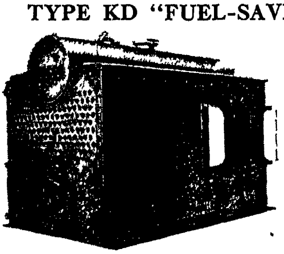
For Office and Apartment Buildings, Schools, Hotels, Theaters, Institutions and Industrial Plants

Built in a complete range of standardized sizes and provide highly efficient performance for heating large buildings.

Up-to-date water tube design permits absorbing the intense heat released by modern methods of firing and they will operate efficiently under loads considerably in excess of ratings.

18 sizes { from 2680 to 42,500 sq ft mechanically fired rating
from 2200 to 35,000 sq ft hand fired rating.

TYPE KD "FUEL-SAVER" WATER TUBE STEEL HEATING BOILERS



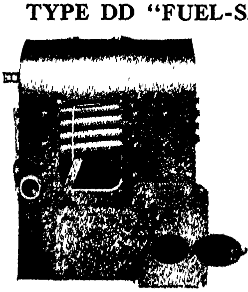
For Replacement Installations in Large Buildings Eliminates Costly Cutting and Patching

Especially designed for renovation and replacement work. Shipped knocked down in standardized parts that can be taken through existing doors or openings to basement and boiler room.

International erects or assumes full responsibility for erection work of knocked down boilers.

15 sizes { 5850 to 56,470 sq ft mechanically fired rating.
4810 to 46,510 sq ft hand fired rating.

TYPE DD "FUEL-SAVER" WATER TUBE STEEL HEATING BOILERS

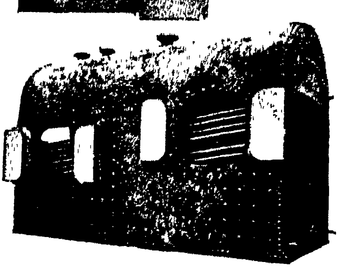


For Residences, Small Apartments and Other Buildings

Highly economical. Superior performance of water tube design usually found only in larger commercial and industrial installations. Fully utilize intense heat generated by modern automatic firing devices. Stoker-fired coal and oil most commonly used, with savings often exceeding 20 per cent over previous installations. Stoker-fired boiler tested and approved by Anthracite Industries Laboratory.

A copper coil submerged above the crown furnishes ample hot water for domestic needs. Insulated steel jackets in two tones of gray enamel are included.

10 sizes { 510 to 2550 sq ft net steam rating.
816 to 4080 sq ft net hot water rating



TYPE CR "FUEL-SAVER" WATER TUBE STEEL POWER BOILERS

For High Pressure Steam Service

Designed for pressure of 100, 125 and 150 lbs, the Type CR is especially suitable for hospitals, hotels, laundries, dairies, institutions and manufacturing plants requiring process steam.

Sizes range from 5 to 300 hp.

Data and catalogs on "Fuel-Saver" Boilers will be furnished on request.

KEWANEE BOILER CORPORATION

Kewanee, Illinois

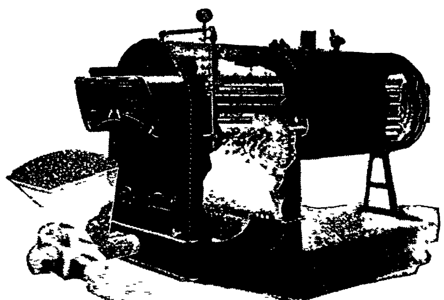
BRANCHES IN 64 PRINCIPAL CITIES

Steel Heating and Power Boilers, Water Heating Garbage Burners, Tabasco Heaters and Tanks.

KEWANEE STEEL HEATING BOILERS

Kewanee offers a dependable line of Steel Boilers built for heating every size building, with high efficiency, burning any kind of fuel. There are 380 standard sizes and 33 types of Kewanee Boilers most of which are kept in stock, ready for immediate delivery.

Seventy-four years of intensive study and effort are back of Kewanee Boiler designs. They are all constructed in our extensively equipped factory at Kewanee, Illinois, in conformity with these Codes: *American Society of Mechanical Engineers* for construction, and for rating with the *Steel Heating Boiler Institute* Simplified Practice.



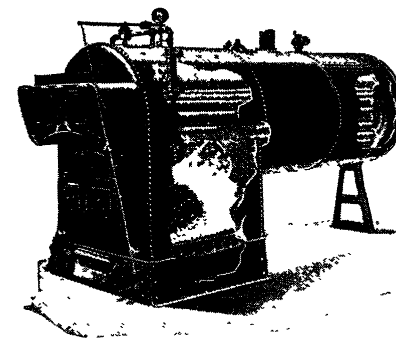
Firebox Boiler for Stoker "400" and "500" Series

The Kewanee series include:

Heavy Duty Riveted Firebox Types: 1,240 ft to 42,500 ft. Brickset and portable settings, Updraft and Downdraft Smokeless Furnace, Single-pass tubes for rear smoke outlet; Two-pass tubes for front smoke outlet.

Welded Boilers: 2,200 ft to 42,500 ft. Direct Draft or Smokeless Arch with Corrugated Crown Sheet. Rear Smoke outlet and Weld + Rivet for front Smoke outlet.

Residence Steel Boilers: 790 ft to 2,924 ft. Square Type "R" with and without Jackets and Hot Water Heating Coils for Storage Tank or Instantaneous flow.



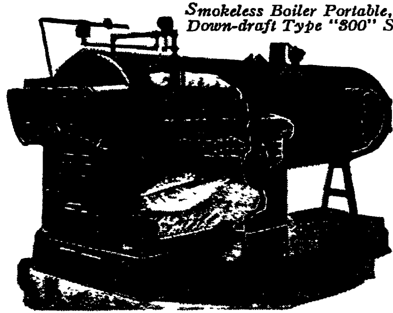
Firebox Boiler Portable Up-draft Type "400" and "500" Series

SPECIFICATIONS—PORTABLE UP-DRAFT BOILER

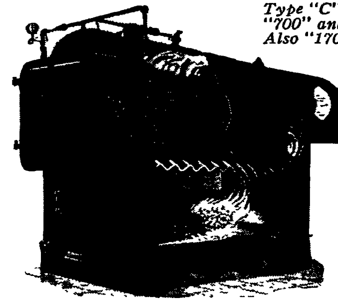
Boiler No.	576	577	578	579	580 480	581 481	582 482	583 483	584 484	585 485	586 486	587 487	588 488	589 489	590 490
Rated Steam Capacity:															
Coal.....Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker.....Sq Ft	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width and Length In. x Ft In.	42x8-7	42x9-6½	48x8-10	48x9-6½	54x11-5	54x13-1½	60x13-4½	60x15-5½	66x15-6½	66x18-0½	72x17-0	78x17-7¼	78x21-3½	84x20-7	84x23-4
Overall Height Shell.....In.	80	80	86	86	94	94	101	101	107	107	113	115	115	125	125
Height of Water Line.....In.	70	70	73	73	79½	79½	84½	84½	89½	89½	94	96½	96½	105	105
Approximate Weight: Coal .Lb	6100	6700	7300	7900	10200	11400	13300	14800	17300	19700	22000	24200	28400	32300	36100
Oil . Lb	5500	6100	6600	7100	9300	10400	12200	13600	15800	18100	20300	22300	26400	30200	33800

Rated Capacity for Water Boiler is 60 per cent greater than Capacity for Steam Boiler.

Table for two series of Boilers lists maximum dimensions only.

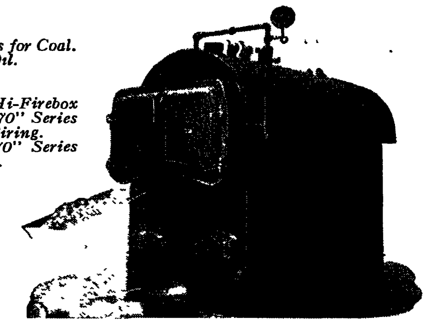


Smokeless Boiler Portable, Down-draft Type "300" Series



Type "C" Boiler "700" and "2700" Series for Coal. Also "1700" Series for Oil.

At Right: Type "C" Hi-Firebox Boiler, "7L70" Series for Stoker Firing. Also "27L70" Series Hand Fired.



SPECIFICATIONS—SMOKELESS DOWN-DRAFT BOILER

Boiler No.	376	377	378	379	380	381	382	383	384	385	386	387	388	389	390
Rated Steam Capacity:															
Coal . . . Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker . . . Sq Ft	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width and Length . . . In. x Ft In.	42x8-3	42x9-3	48x8-7½	48x9-5	54x11-2½	54x12-11	60x12-8½	60x14-9	66x14-11	66x17-4	72x16-5	78x17-0½	78x20-7½	84x19-11	84x22-8
Overall Height Shell . . . In.	80	80	86	86	94	94	101	101	107	107	113	115	115	125	125
Height of Water Line . . . In.	70	70	73	73	78	78	85	85	88½	88½	94½	95	95	107½	107½
Approximate Weight: Coal . . . Lb	6800	7500	8100	8800	10000	11100	13600	15300	18000	20500	22900	25100	29500	33600	37500
Oil . . . Lb	6200	6900	7400	8000	9100	10100	12500	14100	16500	18900	21200	23200	27500	31500	35200

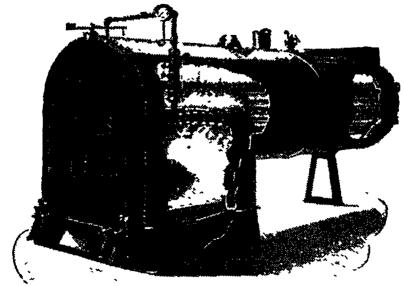
SPECIFICATIONS—TYPE "C" WELDED BOILER

*Boiler No.	2773	774 2774	775 2775	776 2776	777 2777	778 2778	779 2779	780 2780	781 2781	782 2782	783 2783	784 2784	785 2785	786 2786	787 2787	788 2788	789 2789	790 2790
Rated Steam Capacity:																		
Coal . . . Sq Ft	2200	2600	3000	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker . Sq Ft	2680	3160	3650	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width & Length, In.xFt in.	36x5-10	36x6-4	36x6-10	36x7-9	42x7-10½	42x8-6½	42x9-2	48x9-4½	48x10-7	54x9-11	54x11-2½	60x11-6½	66x12-3½	72x12-1½	72x13-4½	78x14-9½	84x14-2½	84x15-11½
Overall Height Shell . . . In.	77½	77½	77½	77½	83½	83½	83½	86½	86½	99	99	108½	112	118	118	122	135	135
Height of Water Line . . In.	69	69	69	69	72	72	72	73	73	85	85	94	95	101	101	103	114	114
Approximate Weight:																		
700 Series, Coal . . Lb	. . .	3900	4400	5000	5500	6000	6500	7500	8400	9700	11000	12900	14900	16600	18400	22000	25200	28400
2700 Series, Coal . . . Lb	3400	3800	4300	4800	5300	5800	6300	7200	8100	9400	10600	12500	14400	16100	17900	21200	24400	27500
Oil . Lb	2900	3300	3700	4100	4600	5000	5400	6100	6900	8000	9100	10700	12300	13800	15400	18100	20900	23400

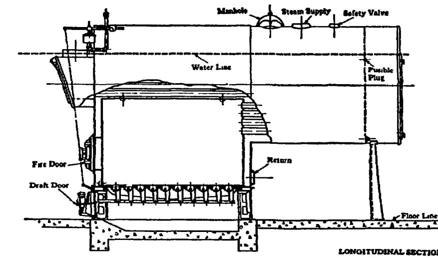
SPECIFICATIONS—TYPE "C" HI-FIREBOX WELDED BOILER *Boiler Series 1773-1790 for Oil, Gas, Stoker; Series 2773-2790 for Anthracite.

Boiler No.	7L73	7L74	7L75	7L76	7L77	7L78	7L79	7L80	7L81	7L82	7L83	7L84	7L85	7L86	7L87	7L88	7L89	7L90
Rated steam capacity,																		
Stoker . . . Sq Ft	2680	3160	3650	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width x Length, In.xFt In.	36x5-10	36x6-4	36x6-10	36x7-9	42x7-10½	42x8-6½	42x9-2	48x9-4½	48x10-7	54x9-11	54x11-2½	60x11-6½	66x12-3½	72x12-1½	72x13-4½	78x14-9½	84x14-2½	84x15-11½
Overall Height . . . In.	82½	82½	82½	82½	88½	88½	88½	94½	94½	108	108	119½	123	131	134	140	157	161
Height of Water Line . . In.	74	74	74	74	77	77	77	81	81	94	94	105	106	114	117	121	136	140
Approximate Weight . Lb	3100	3500	3900	4400	4900	5400	5800	6700	7600	8800	10000	11800	13600	15300	17000	20100	23200	26100

Brickset Boiler Up-draft Type



Type "K" Up-draft Boiler



Weld + Rivet Firebox Boiler "5000" Series Up-draft.
Also "6000" Series Smokeless Down-draft

SPECIFICATIONS—BRICK-SET AND TYPE "K" PORTABLE UP-DRAFT BOILERS

Boiler No.	3 3K	4 4K	5 5K	6 6K	8K	8 9K	9 10K	10 11K	11 12K	12 13K	13 14K	14 15K	15 16K	17 18K	18 20K	19	20
Rated Steam Capacity:																	
Coal .... Sq Ft	1240	1380	1800	2200	3000	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000
Oil, Gas or Stoker ...... Sq Ft	1770	2020	2190	2680	3650	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290
Width and Length .. In. x Ft In.	30x9-10	36x8-10	36x10-4	36x11-10	42x11-4	42x12-10	48x11-10	48x13-4	48x14-10	54x15-3	54x17-9	60x17-0	60x19-6	66x19-6	72x19-6	72x16-0	72x18-0
Overall Height Shell .... In.	59	65	65	65	71	71	77	77	77	83	83	93	93	99	105	107	107
Height of Water Line .... In.	52	55	55	55	58½	58½	61	61	61	66	66	75	75	80	85½	85½	85½
Approximate Weight: Coal ... Lb	3500	3900	4200	4700	5700	6400	7000	7600	8200	9500	10700	12600	14500	17600	20700	19400	21900
Oil. ... Lb	3130	3550	3800	4200	5100	5800	6400	6900	7400	8600	9700	11500	13300	16100	19100	17700	20000

SPECIFICATIONS—WELD + RIVET FIREBOX BOILER "5000" SERIES

Boiler No.	5076	5077	5078	5079	5080	5081	5082	5083	5084	5085	5086	5087	5088	5089	5090
Rated Steam Cap: Coal ... Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, gas or stoker .... Sq Ft	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Diam. x Length ..... In. x Ft In.	42x7-3	42x8-0	42x8-11	42x9-10	48x9-9	48x11-1¼	54x11-1	54x12-9	60x13-2½	60x15-3	66x14-5½	66x16-0½	72x16-8½	78x17-5½	78x19-8½
Overall Height ...... In.	81	81	81	81	86½	86½	92½	92½	100½	100½	105	105	112	117	117
Height of Water Line ..... In.	70	70	70	70	73½	73½	78	78	84	84	86	86	94½	96	96
Approx. Weight: Coal ..... Lb	5100	5600	6100	6600	7500	8400	9600	10800	12600	14400	16100	17900	21200	24400	27500
Oil ..... Lb	4500	5000	5400	5800	6600	7400	8500	9600	11100	12800	14400	16000	19200	22300	25200

SPECIFICATIONS—WELD + RIVET SMOKELESS BOILER—"6000" SERIES

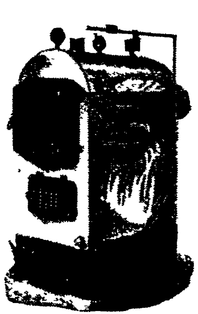
Boiler No.	6077	6078	6079	6080	6081	6082	6083	6084	6085	6086	6087	6088	6089	6090
Rated Steam Capacity: Coal ...... Sq Ft	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker .... Sq Ft	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Diameter x Length .... In. x Ft In.	42x8-2	42x8-8	42x9-6	48x9-7	48x11-0	54x10-8½	54x12-4½	60x12-9½	60x14-9½	66x14-0½	66x15-7½	72x16-4½	78x17-1	78x19-4
Overall Height .... In.	81	81	81	86½	86½	92½	92½	100½	100½	105	105	112	117	117
Height of Water Line .... In.	70	70	70	73½	73½	78	78	84	84	86	86	94½	96	96
Approximate Weight: Coal .... Lb	5800	6300	6800	7700	8600	9900	11200	13100	14900	16600	18300	21500	24500	27300
Oil .... Lb	5100	5600	6100	7000	7800	8900	10100	11900	13500	15100	16700	19800	22700	25300

KEWANEE TYPE "R" RESIDENCE BOILERS

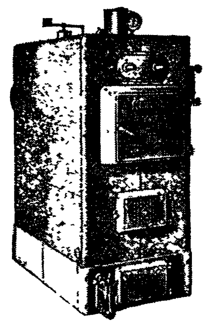
Kewanee Type "R" Boilers are especially designed and constructed to meet all heating and hot water requirements for homes and small buildings. Every kind of solid fuel, coke, all grades of hard or soft coals and their briquette or treated forms are burned with excellent results. Also, any liquid fuel, oil, and natural or commercial gas can be used with high efficiency.

Standard snug fitting jackets, or Regal style for completely enclosing burners are available for 83R.

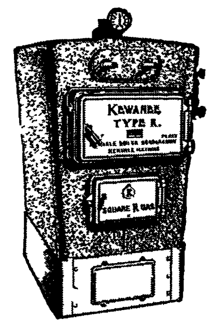
Hot Water Copper Coil. Capacities up to 720 may be ordered for Square and 83R Boilers.



Square Type "R" Residence Boiler



Square Type "R' Jacketed Boiler



83R Oil or Gas Jacketed Boiler

SPECIFICATIONS—RESIDENCE SQUARE TYPE "R" BOILER

*Boiler No.		742	743	745	746	747	748
Rated Steam Capacity: Coal	Sq Ft	790	1000	1350	1600	1780	1960
Oil, Gas or Stoker	Sq Ft	840	1120	1470	1900	2160	2380
Width and Length	In. x In.	32x39½	32x45½	32x45½	32x51½	32x57½	32x63½
Overall Height Shell Top	In.	59½	59½	70¼	70¼	70¼	70¼
Height of Water Line	In.	48	48	58½	58½	58½	58½
Approximate Weight: Coal	Lb	2150	2360	2800	3050	3300	3550
Oil	Lb	1900	2060	2500	2730	2920	3125
Standard Jacket, Crated	Lb	205	225	175	190	200	225

Boiler No., Square "R" Oil or Gas	83R1	83R2	83R3	83R4	83R6	83R7	83R8	83R9
Rated Steam Capacity, Sq Ft	901	1105	1326	1513	2091	2363	2652	2924
Approximate Weight with Jacket, Lb	1600	1800	2000	2200	2800	2900	3100	3300

*Boiler Series 1742-1748 for Oil, Gas or Stoker; 2742-2748 for Anthracite. Kewanee Indirect Hot Water Heating Coils for Type C and Square "R" Boilers; 55 sizes, 90 to 1520 Gal

KEWANEE STORAGE WATER HEATERS

Kewanee Storage Heaters—use exhaust or live steam. 15 Standard Coil Elements in 29 standard size storage tanks. Capacities 95 to 2240 gals.

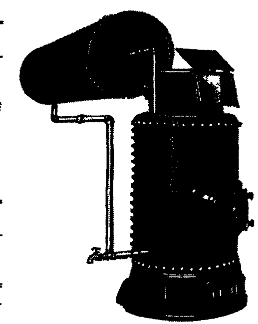
KEWANEE SCOTCH MARINE BOILERS

Kewanee Scottie Junior for High Pressure Steam in small industrial usage. 5 sizes, 9.9 to 30 hp at 100 lbs steam pressure.

Kewanee Welded Scotch Marine, Low Pressure Steam. 18 sizes, 2680 to 42500 sq ft, mechanically fired.

KEWANEE HI-TEST BOILER

Kewanee Hi-Test Fusion Welded Series for High Pressure Steam in power or industrial process. 6 stock sizes, 50 to 150 hp, 125 and 150 lbs steam working pressure. All A.S.M.E. Code



Tabasco Water Heater, Magazine Feed

KEWANEE WATER HEATERS AND TANKS

Tabasco Water Heaters, steel riveted, 17 sizes, heat 130-700 gal 50 deg per hour. Standard 60 lb: Extra Heavy 100 lb W. P.

Water Heating Garbage Burners, 3 types, 16 sizes, heat 200-2600 gal 50 deg per hour. Working Pressure up to 125 lbs.

Spencer Heater Division

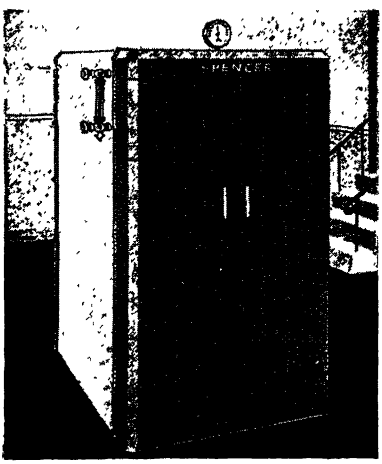
The Aviation Corporation

Williamsport, Pa.

Sales Representatives in Principal Cities

Spencer Automatic Magazine Feed Heaters are furnished in cast iron sectional types—and steel tubular types for larger buildings—for steam, vapor and hot water heating. There is a size and capacity for every type of building, to provide economical and convenient heat—safe, dependable, sure.

COMFORTABLE HEAT AT LOW COST



Spencer Jacketed Heater L-1 Series

Why Spencer Heaters perform so satisfactorily can best be explained by an inspection of their design and construction. The Spencer principle, illustrated in the cross-sectional view, is simple:

Once a day fuel (No. 1 Buckwheat Anthracite or small size by-product coke) is put into the magazine. It fills the sloping grate to the level of the magazine mouth. The fire bed always stays at the proper level, for as fast as fuel burns to ash, it shrinks and settles on the sloping grate; and more fuel rolls down automatically over the top of the fire bed. Fuel feed is by gravity alone, in just the right amount to keep the fire always burning at its most efficient combustion point.

This explains why a Spencer Automatic Magazine Feed Heater always gives the same uniform, satisfying heat, and burns less fuel. These exclusive Spencer advantages are available in all types of the magazine feed heaters and boilers.

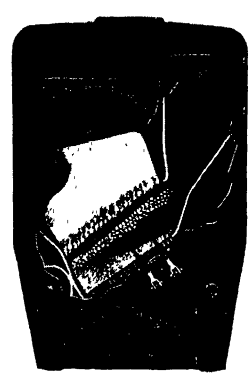
Coal — Coke — Gas — Oil — Spencer J and L series heaters and M series boilers are primarily designed to burn low cost No. 1 Buckwheat Anthracite or small size coke.

If at any time a property owner desires to burn more expensive fuels—oil or gas—his Spencer Heater can be readily converted and will show a high efficiency.

Thermostats—Thermostats and electric damper motors are furnished as optional equipment.

Jacketed Covering—Attractive metallic jackets of the deluxe enclosing type, as illustrated, are available for Spencer Cast Iron Heaters, either with or without the enclosing jacket doors.

Spencer Heavy Duty Tank Heaters—With the automatic magazine feed construction, they provide ample domestic hot water at lowest cost, and with a minimum of tank heater attention.



Cutaway sectional view Spencer Cast Iron Heater

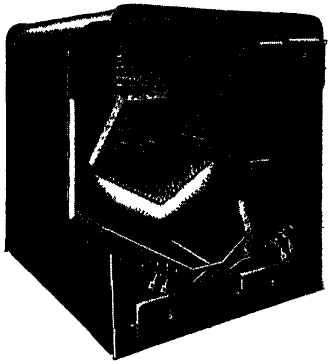
SPENCER ALL YEAR SYSTEM

In addition to the excellent heating facilities afforded by Spencer Magazine Feed Heaters, Year Round Domestic Hot Water Service can also be provided and assures at all times an ample supply of domestic hot water at lowest cost. Complete data for installation and operation upon request.

SPENCER STEEL TUBULAR MAGAZINE FEED BOILERS

For large buildings we recommend Spencer Steel Tubular Magazine Feed Boilers, burning low cost No. 1 Buckwheat Anthracite or coke.

In the cross-section diagram, part of the fire bed is cut away to show the sloping grates and the two magazines filled with fresh coal, ready to feed down automatically by gravity to the fire. These boilers are built in two vertical sections for ease in handling and installation—a great advantage on replacement jobs, eliminating the necessity of costly tearing out of walls or partitions. Combination water and fire tube construction; built to *A.S.M.E.* standards.



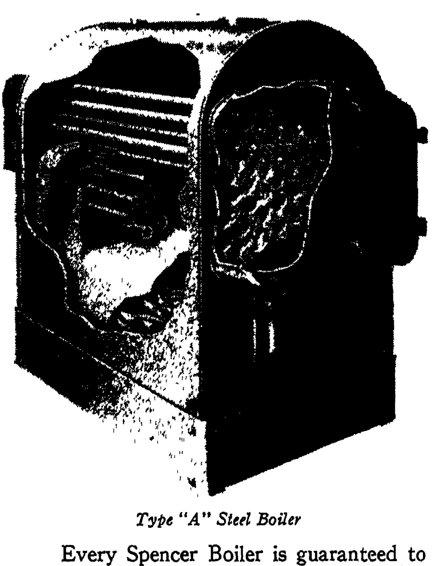
Steel Tubular Magazine Feed Boiler

SPENCER STEEL TUBULAR BOILERS
For Oil, Stoker, Gas or Hand-Firing

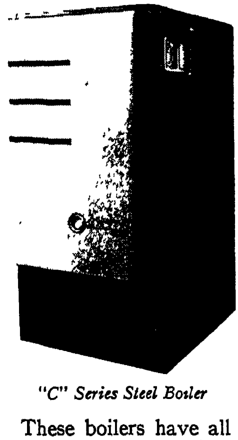
For more than 50 years, Spencer has been building, in the opinion of experts, one of the most efficient, economical and dependable automatic coal burning boilers on the market. With this background of experience, Spencer Engineers developed the Spencer Steel Tubular Boiler for oil, gas, stoker and hand-firing—the "K" and "C" series for residential use, and the Type "A" for larger buildings. They are better boilers both for the property owner and for the architect or engineer who specifies them.

The high sustained efficiency of these boilers means adequate heat for a lower fuel cost. Design is of the three pass type. Combustion chamber is amply large. Built of best quality open hearth steel boiler plate, and steel tubes. Can be furnished with domestic hot water heating coils, storage tank or instantaneous type.

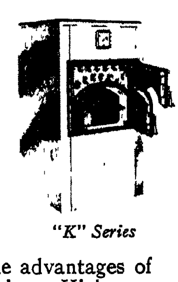
A complete range of sizes from 400 sq ft SHBI net steam rating up. They meet or exceed in every particular the requirements of the A.S.M.E. and S. H. B. I. Codes.



Type "A" Steel Boiler



"C" Series Steel Boiler



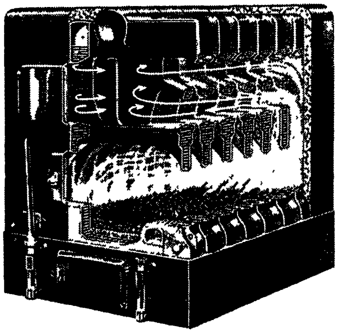
"K" Series

Every Spencer Boiler is guaranteed to carry more than its full rated load giving the installer a definite factor of safety. These boilers have all the advantages of the Spencer exclusive design. High sustained efficiency—low fuel cost.

UNITED STATES RADIATOR CORPORATION

General Offices: Detroit, Michigan • **Branches and Sales Offices in Principal Cities**

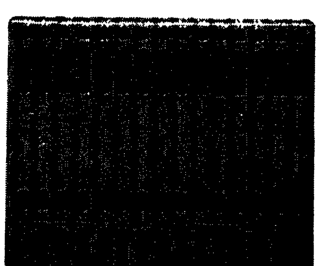
Detroit, Michigan



CAPITOL RED TOP BOILERS FOR ALL FUELS

	Capacity in Sq Ft	Direct Cast Iron Radiator Loads—Sq Ft
"A" Series	Steam— 575-1450 Water— 975-2475	240- 740 385-1185
"B" Series	Steam—1800-3600 Water—2970-5940	550-1800 880-3000
"C" Series	Steam—4700-10,500 Water—7760-17,325	2250-5800 3600-9280

Illustrated above is a Capitol Red Top Series "C" Boiler. Capitol Red Top Boilers can be furnished with extra high steel bases to provide extra setting height or desired additional furnace volume for stoker firing.



U. S. SUNRAY RADIATOR

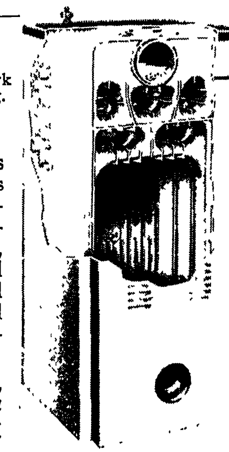
Space Saving—Can be fully or partially recessed—Also well adapted for free standing installation.

Self-contained Cabinet Radiator—designed to form its own enclosure.

U. S. SUNRAY—NO. 3 SERIES

Trade Mark U. S. Reg. Pat. Off.

The No. 3 Series Sunray Boiler is furnished in coal-burning and oil-burning types, with a standard jacket, or a special front extended jacket for the oil-burning type.



Cutaway of Sunray No. 3 Series Steam Oil Burning Boiler showing Biltin Taco Heater

Direct Cast Iron Radiator Load—Sq Ft

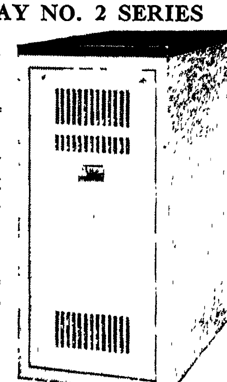
Boiler Number	Hand Fired Steam	Hand Fired Water	Stoker Fired Steam	Stoker Fired Water	Oil Fired Steam	Oil Fired Water
23 S or W	See Note	See Note		.	260	415
33 S or W	. .	. .			310	500
43 S or W	300	480	400	640	400	640
53 S or W	450	720	550	880	550	880
63 S or W	600	960	700	1120	700	1120

NOTE: Rating—150 sq ft Steam and 240 sq ft Water Direct Cast Iron Radiator Load for Anthracite Fuel only.

Biltin or external Taco Heaters for summer-winter domestic hot water hook-up available on these boilers.

U. S. SUNRAY NO. 2 SERIES

Trade Mark U. S. Reg. Pat. Off.



Direct C. I. Radiator Load—Sq Ft

Oil Fired Only

Blr. No.	Steam	Water
2-03	350	560
2-04	500	800
2-05	650	1040
2-06	800	1280

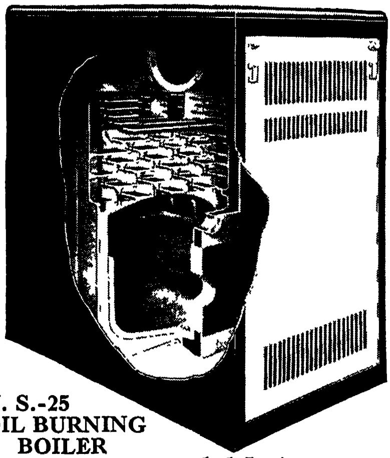
The No. 2 Series Sunray Boiler is designed to provide a highly efficient, yet moderately priced boiler for modern, automatic heating. The two-tone green jacket completely encloses the burner, controls and accessories, which are readily accessible for installation or servicing.

Literature Upon Request

UNITED STATES RADIATOR CORPORATION

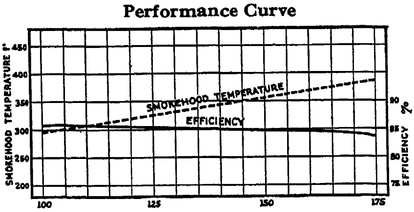
General Offices: Detroit, Michigan • **Branches and Sales Offices in Principal Cities**

Detroit, Michigan



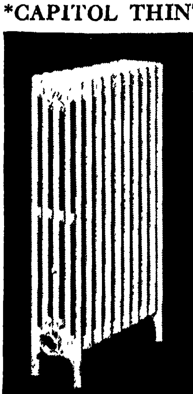
U. S.-25 OIL BURNING BOILER
Front Extended Jacket

Performance Curve



Output—% of Direct Standing Radiator Load Flue Gas Analysis CO_2—12.5%; O_2—4.1%; CO—0.0%.

*CAPITOL THINTUBE RADIATORS



Heights	Per Section Heating Surface
3-Tube	
25″	1.6 Sq Ft
4-Tube	
19″	1.6 Sq Ft
22″	1.8 Sq Ft
25″	2.0 Sq Ft
5-Tube	
22″	2.1 Sq Ft
25″	2.4 Sq Ft
6-Tube	
19″	2.3 Sq Ft
25″	3.0 Sq Ft
32″	3.7 Sq Ft

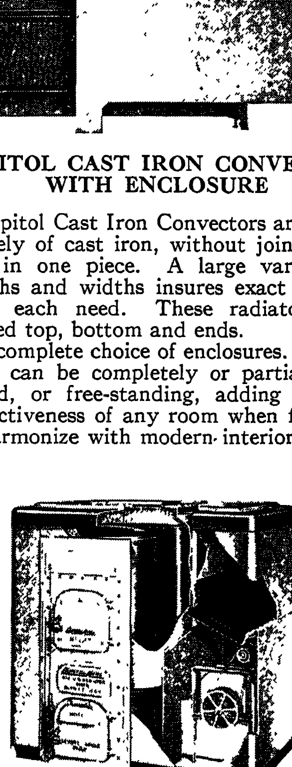
*1¾ in. Centers.

40 per cent less space needed for these graceful, efficient Capitol ThinTube Radiators.

CAPITOL CAST IRON CONVECTOR WITH ENCLOSURE

Capitol Cast Iron Convectors are made entirely of cast iron, without joints, and cast in one piece. A large variety of lengths and widths insures exact size to meet each need. These radiators are tapped top, bottom and ends.

A complete choice of enclosures. These units can be completely or partially recessed, or free-standing, adding to the attractiveness of any room when finished to harmonize with modern interiors.



THRIFT SERIES CAPITOLAIRE CONDITIONING UNIT

An air conditioning unit especially designed for all-fuel firing. Streamline flue construction in preheating and primeheating sections insures high efficiencies. The handsome enamelled casing houses the heating element, blower and filter.

The blower and filter compartment may be placed at either side of the heating element. The "Luxury" series units have blower section at rear only.

Other gravity and forced air furnaces available for gas, oil, stoker or hand coal firing.

Literature Upon Request

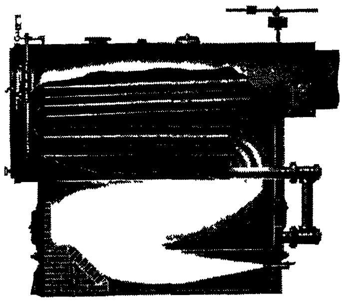
Pacific Steel Boiler Division

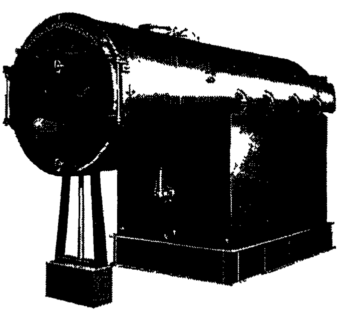
United States Radiator Corporation

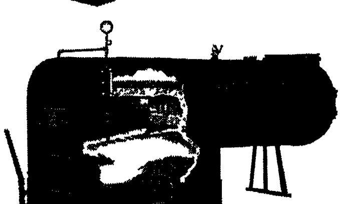
General Offices: **Detroit, Michigan**

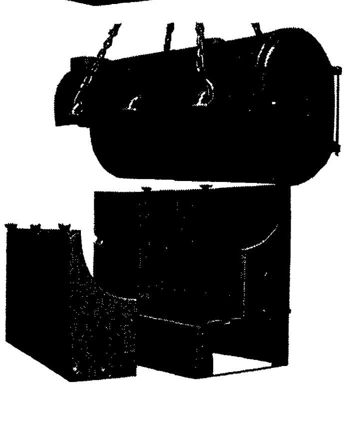
Sales Offices in Principal Cities

A Complete Line of Low Pressure Steel Heating Boilers









All Pacific Boilers are built using the *A.S.M.E.* Boiler Code Standards as minimums.

LOW WATER LINE SERIES

Built in the following capacities for steam:

Coal Burning Sizes—1800 to 35,000 sq ft.

Mechanically Fired Sizes—2680 to 42,500 sq ft.

High Fire Box for Stoker Firing—Sizes—2680 to 42,500 sq ft.

All Pacific Boilers are built, inspected, and tested under the supervision of the Hartford Steam Boiler Inspection and Insurance Company.

TWO-PASS FRONT SMOKE OUTLET

Built in the following capacities for steam:

Coal Burning Sizes—4000 to 30,000 sq ft.

Mechanically Fired Sizes—4860 to 42,500 sq ft.

All Pacific Boilers are made of steel with each joint and seam electrically arc-welded —built to last a life-time.

SINGLE-PASS REAR SMOKE OUTLET

Built in the following capacities for steam:

Coal Burning Sizes—1800 to 6000 sq ft.

Mechanically Fired Sizes—2190 to 7290 sq ft.

PACIFIC THREE-PIECE CONSTRUCTION

Made up of three parts, shell, firebox and base, Pacific Boilers are particularly adaptable to replacement work. Where necessary Pacific fireboxes can be split (as illustrated) allowing the boiler to be taken into the building in four pieces and erected without welding on the job.

Descriptive Bulletins on Pacific Steel Boilers will be mailed on request.

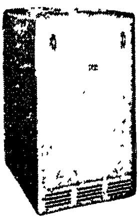
Weil-McLain Company

Manufacturing Division: **Michigan City, Ind. and Erie, Pa.**

General Offices: **641 W. Lake Street, Chicago**

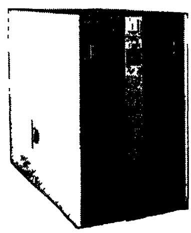
NEW YORK OFFICES: **501 Fifth Avenue**

Prompt Weil-McLain Boiler and Radiator service is made conveniently available through local stocks carried by Weil-McLain Distributors in most of the important distributing centers.



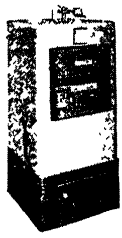
No. 68 Boiler for Automatic Firing

Boiler is completely jacketed and insulated. Has an integral front burner extension. Net I-B-R Ratings: Steam 390 to 690 sq ft, Water 625 to 1,100 sq ft.



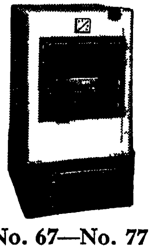
No. 78 Boiler for Automatic Firing

Boiler has insulated enameled de luxe jacket. Front or rear jacket extension available. Net I-B-R Ratings: Steam 530 to 1,130 sq ft, Water 850 to 1,810 sq ft.



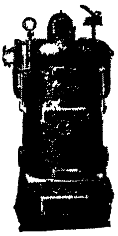
New No. 57 All-Fuel Boiler

Jacketed and insulated square boiler for small homes. Net I-B-R Ratings: Steam 210 to 640 sq ft, Water 340 to 1,030 sq ft.



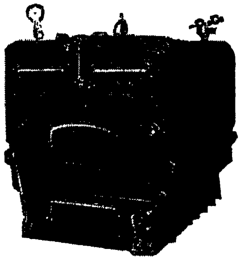
No. 67—No. 77 All-Fuel Boilers

Conversion type boilers with insulated enameled jacket. For hand or automatic firing. Net I-B-R Ratings: Steam 290 to 620 sq ft, Water 465 to 990 sq ft.



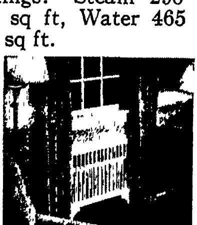
Round-Type Boiler

Unjacketed Round Boiler with corrugated heating surfaces for economical home heating. Connected Load Ratings: Steam 210 to 1,000 sq ft, Water 335 to 1,600 sq ft.



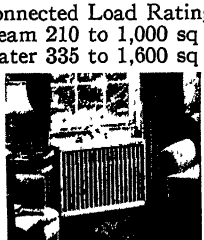
Square-Type Boilers

Sectional boilers for larger installations. Complete range of sizes. Connected Load Ratings: Steam 925 to 11,300 sq ft, Water 1,470 to 17,900 sq ft.



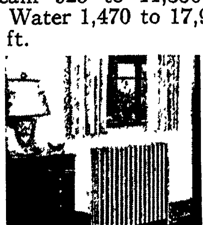
Raydiant "Concealed"

A Raydiant convector type all cast-iron Radiator. Made in "Concealed," also Partially Recessed, Cabinet and Humidifying types.



Solray Radiator

Free standing Cabinet type Radiator in a lower price range than Raydiant Cabinet Radiators. Available in three depths in 21, 24 and 27 in. heights.



Junior Radiator

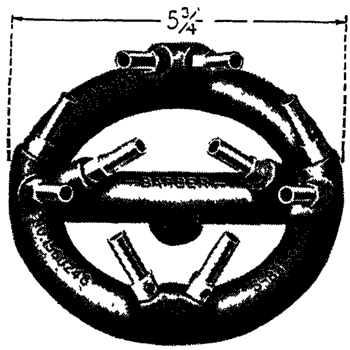
Smaller Tubular type Radiation which conserves space. Available in 1¾ in. centers in 3, 4, 5 and 6 tube widths and 14 to 33 in. heights.

The Barber Gas Burner Company

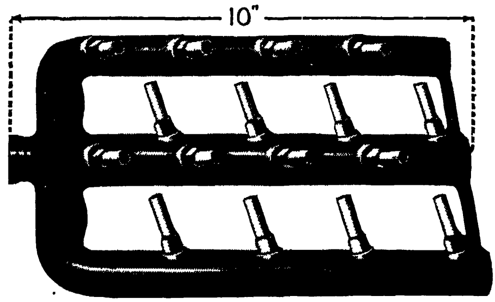
3704 Superior Ave., **Cleveland, Ohio**

Address Michigan Inquiries to The Barber Gas Burner Co., of Mich., 4475 Cass Ave , Detroit, Mich.

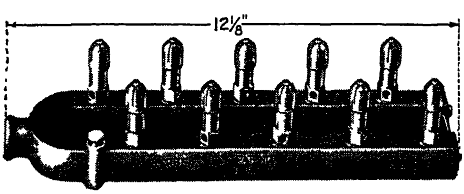
Barber Automatic Jet Gas Conversion Burners, for heating and air conditioning equipment, have a record of high efficiency, for a period of over 20 years giving continuous satisfaction in many thousands of homes and other buildings. The exclusive Barber Jet principle of combustion, attaining 1900 deg flame temperature on atmospheric pressure, and other basic advantages of design, have given Barber a permanent place in modern heating and air conditioning practice. Barber Burners for gas burning appliances have been adopted as standards by many appliance manufacturers, for Natural, Butane-Propane, or Bottled Gas. Shown here are only a few items from Barber's complete line. Illustrated No. 42 Catalog and Price List furnished on request.



No. S-85 Burner Unit



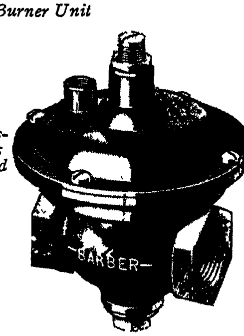
No. U-16 Burner Unit

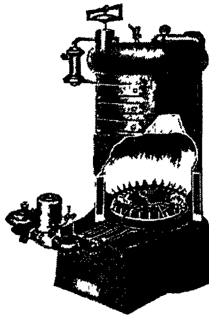


No. C. L-90 Burner Unit

Barber Gas Pressure Regulators A.G.A. Approved

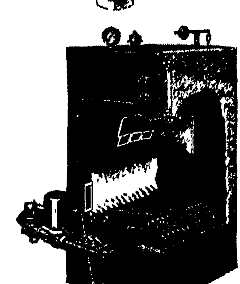
Made in sizes ¼" and up.





Conversion Burners for Furnaces or Boilers

Burners are adjustable as to diameter, on the job, to fit practically all round grate sizes. Also to fit grates of oblong furnaces and boilers. Listed in the *A.G.A.* Directory of Approved Appliances. Equipped with automatic controls with motor gas valve, with magnetic gas valve control, with quick acting gas valve control (for buildings equipped with automatic heat control), or in "M" series with manual control.



Barber Burners and Regulators are Adaptable to: Air Conditioning Equipment, High Pressure Boilers (Tubular and Tubeless), Bakery Ovens, Garage Heaters, Coffee Urns, Hair Dryers, Space Heaters, Floor Furnaces, Clothes Dryers, Water Heaters, Confectioners' Stoves, Vulcanizing Machines, Pressing Machine Boilers, Japanning Ovens, Core Ovens, Banana Room Heaters, Other Appliances.

Gas Burner Specialists offering Engineering Department and Laboratory facilities for Gas Burner problems. Consultation invited.

The Webster Engineering Co.

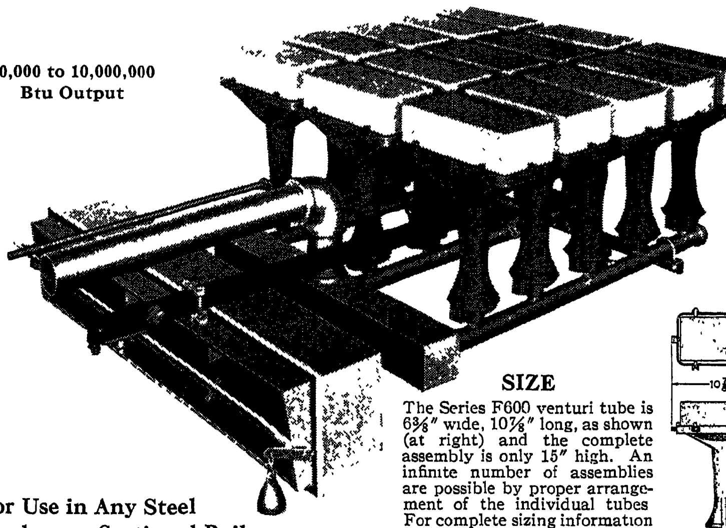
419 West 2nd St., **Tulsa, Oklahoma**

Division of

SURFACE COMBUSTION CORP., TOLEDO, OHIO

WECO-N.G.E. SERIES F600 GAS BURNERS

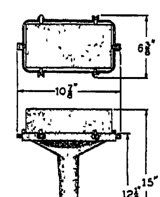
50,000 to 10,000,000 Btu Output



For Use in Any Steel Firebox or Sectional Boiler

SIZE

The Series F600 venturi tube is 6⅜″ wide, 10⅞″ long, as shown (at right) and the complete assembly is only 15″ high. An infinite number of assemblies are possible by proper arrangement of the individual tubes For complete sizing information see Bulletin F600H.



Improved venturi and greater port area insure much higher capacities at lower pressures.

Unique baffles at the outlet of the mixing tube make possible perfectly even distribution of flame completely around the baffle brick. As a result the maximum flame length is greatly reduced.

Interchangeable grills with multiple ports can be varied to suit the combustion characteristics of various gases. The proper sizing of these grills prevents any possibility of flash back.

In addition to the above major improvements the F600 possesses the same desirable features that made the 600 so popular.

1. Simple installation requiring no expensive insulated combustion chamber and having no furnace radiation loss.
2. Extreme quietness due to low rate of combustion over a large area.
3. Flexibility from infinite number of possible combinations varying both size and shape to meet load and firebox conditions at various gas pressures.
4. High radiant transmission rate due to radiant temperature of the standard firebrick baffles on the top of the burner tubes.
5. Low draft loss because of ample secondary air openings.
6. Plain gas pilots of heat resistant material and of a design that will not allow flame to pull off.
7. Safety pilot applied in a cool zone in a manner that insures perfect direct ignition of the burner yet allowing the the thermal element to cool quickly upon flame failure.
8. Guaranteed vibrationless under all conditions.

CAPACITY OF SINGLE F600 VENTURI TUBE—No. 17 MTD ORIFICE

Manifold Gas Pressure	0.5″ W.C.	1.0″ W.C.	2 0″ W.C.	3 0″ W.C.	4 0″ W.C.	5.0″ W.C.	6.0″ W.C	4 oz. W.C.	6 oz. W.C.	8 oz. W.C.
Input—Cu Ft, 1 hr	24.5	38.5	58.0	72.0	84.0	94.5	104.0	112.0	138.0	159.5
Output—Sq Ft, St. Rad.	75	117	178	222	258	2.9	318	343	423	488
Output—Boiler H.P.	.54	.84	1.23	1.59	1.85	2.07	2.28	2.46	3.04	3.4

TODD COMBUSTION EQUIPMENT, INC.

(Division of Todd Shipyards Corporation)

601 West 26th Street, New York City

NEW YORK MOBILE NEW ORLEANS GALVESTON SEATTLE BUENOS AIRES LONDON

THE TODD HEX-PRESS REGISTER in combination with the **TODD "VEE-CEE" VARIABLE CAPACITY BURNER . . .** makes possible increased combustion efficiency under almost any type of boiler of 100 H.P. capacity or larger, operating at 50 pounds steam pressure or higher.

It provides equal efficiency under either forced or natural draft conditions. The Hex-Press Register assures the most intimate mixture of oil and air as well as quicker, more complete combustion . . . with minimum draft loss at high capacity . . . effecting great economy in maintenance and materially reducing fuel costs.

Through the exclusive "variable range" feature of the "Vee-Cee" Burner, practically unlimited firing range is assured . . . without change of burner tips, oil delivery pressure or angle of spray.

Constant steam pressure can be maintained regardless of demand . . . changing load requirements are met instantly under manual or fully automatic control.

All installations of Todd Equipment are always ***individually engineered*** to fulfill specific requirements. *Send for descriptive literature.*

RECENT INSTALLATIONS OF TODD BURNERS:

Apartment House, *240 Central Park South, New York, N.Y.*
Hunter College....*New York, N. Y.*
New York Life Insurance Co. Bldg., *New York, N. Y.*
Prudential Insurance Co. Bldg., *New York, N. Y.*
American Can Co....*Portland, Maine*
Bell Telephone Laboratories, *Murray Hill, N. J.*
Consolidated Edison Co. of New York, Sherman Creek Plant....*New York, N. Y.*
Grumman Aircraft Eng. Corp., *Beth Page, Long Island, N. Y.*
Holyoke Gas & Electric Co., *Holyoke, Mass.*
Celulosa Argentina, Rosario, *(Santa Fe) Argentina, S. A.*
Queensbridge Housing Project, *Long Island City, N. Y.*
New York Municipal Airport, *(La Guardia Field) Jackson Heights, N. Y.*
Washington Municipal Airport, *Gravelly Point, Va.*
The Glenn L. Martin Co., *Middle River, Baltimore Co., Md.*
National Gypsum Co., *Port Wentworth, Ga.*
Brooklyn College....*Brooklyn, N. Y.*
Sears Roebuck & Co....*Chicago, Ill.*
Allison Engineering Plant, *Indianapolis, Ind.*
Bethlehem Steel Co....*Lackawanna, N. Y.*
Allis Chalmers Mfg. Co....*La Crosse, Wis.*
Hudson Naval Gun Plant....*Detroit, Mich.*
Pratt-Whitney Div. of United Aircraft Corp....*East Hartford, Conn.*
Peoples Gas Light & Coke Co..*Chicago, Ill.*
Wright Aeronautical Corp..*Paterson, N. J.*
Edgewood Arsenal....*Edgewood, Md.*
International Silver Co....*Meriden, Conn.*
Camp Blanding....*Starke, Fla.*
Canadian Car & Munitions, Ltd., *Quebec, Canada*
Grayslake Gelatin Corp....*Grayslake, Ill.*
Elwood Ordnance Plant....*Wilmington, Ill.*
Woodward & Lothrop Department Store, *Washington, D. C.*
Puget Sound Navy Yard, *Bremerton, Wash.*
Stone & Webster Office Bldg., *Boston, Mass.*

TODD MANUFACTURES: Mechanical Pressure Atomizing Oil Burners--VEE-CEE Variable Capacity Burners—Horizontal Rotary Oil Burners—Oil Burning Air Registers for Natural, Assisted, Induced or Forced Draft—Inside Mixing Steam Atomizing Oil Burners—Combination Gas and Oil Burners—Furnace Doors and Interior Castings for Converting Howden Type Furnace Fronts to oil firing—Oil Burning Galley Ranges—Oil Heating, Pumping and Straining Equipment.

Todd engineers are always available for consultation and analysis of combustion problems—without obligation.

The Brownell Company

ESTABLISHED 1855

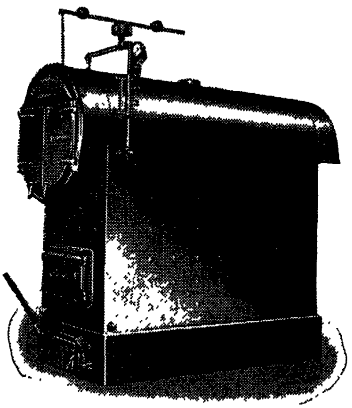
Dayton, Ohio

Manufacturers of

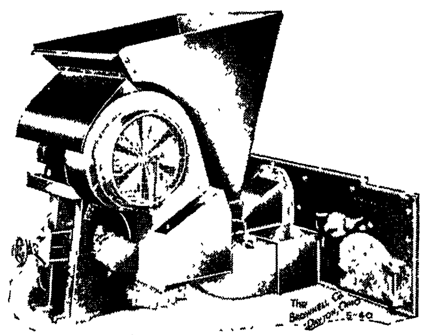
BROWNELL BOILERS AND STOKERS

Representatives in All Principal Cities

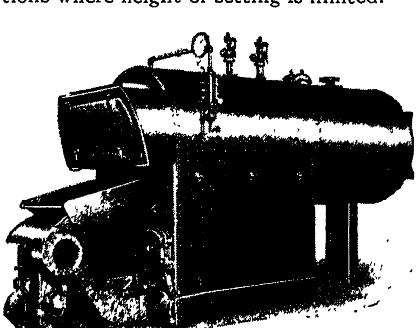
FIRE TUBE BOILERS of various types. HEATING BOILERS riveted and welded. UNDERFEED STOKERS from 5 Horse Power upwards STEEL STACKS, TANKS AND SPECIAL PLATE WORK.



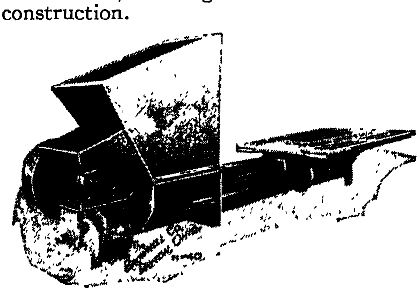
Welded Triple Pass Heating Boilers built in either high leg or low water line types. Hand fired ratings 500 to 35,500 sq. ft. steam, 800 to 56,800 sq. ft. water radiation. Stoker, Oil or Gas fired up to 43,100 sq. ft. steam or 69,000 sq. ft. water radiation. A.S.M.E. Code construction.



Type L R (Low Set) Underfeed Ram Type Stoker with automatic air volume control. Can be furnished with Brownell exclusive, fully automatic coal feed control. Sizes up to 300 horse power. An ideal stoker for firebox boiler or other installations where height of setting is limited.



High or Low Pressure Double Pass Boiler with Type L R Stoker. Designed and manufactured as a matched unit steam generating plant. Furnished in working pressures from 15 to 150 pounds and sizes up to 300 horse power. For power, heating and process steam. Steam ratings 3,600 to 42,500 sq. ft. Water rating 5,800 to 68,000 sq. ft. when used with stoker, oil or gas. A.S.M.E. Code construction.



Type C Screw Feed Stoker, proved by years of service to be sturdy, reliable and efficient. Illustration shows dead plates can also be furnished with dump plates in the larger sizes. 30-300 HP.

The illustrations above show only a part of the complete Brownell line. We shall gladly send literature describing BROWNELL BOILERS and STOKERS. Our nation wide field organization is ready to assist in problems of steam generation.

Combustion Engineering Company, Inc.

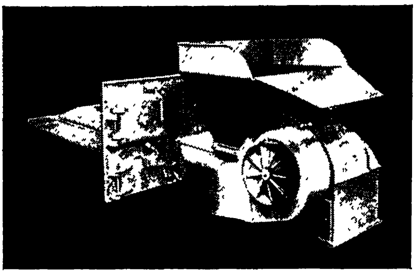
All Types of Fire Tube and Water Tube Boilers
Mechanical Stokers

Complete Steam Generating Units
Pulverized Fuel Systems

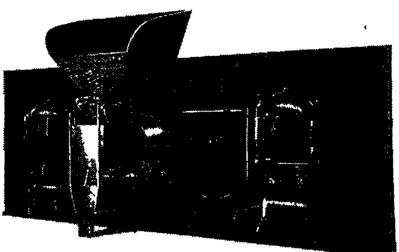
200 Madison Avenue, **New York, N. Y.**

Offices in all principal cities of the United States and Canada

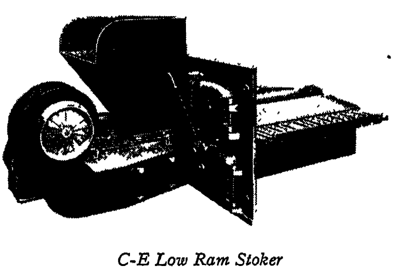
More than 16,000 C-E Stokers installed to date



C-E Skelly Stoker Unit



Type E Stoker



C-E Low Ram Stoker

C-E Spreader Stoker

C-E Skelly Stoker Unit—A compact, self-contained unit with integral forced-draft fan, adapted to burn either anthracite or bituminous coal. Alternate fixed and moving grate bars assure lateral distribution of fuel. Automatic control is standard equipment. Approximate application range—20 to 200 rated boiler hp.

Type E Stoker—A single-retort, underfeed stoker with an established reputation of many years' standing for dependable service. Designed to burn a variety of bituminous coals under boilers up to about 600 rated hp. Available with steam, electric or hydraulic drive.

C-E Low Ram Stoker—A single-retort, stationary-grate underfeed stoker for burning bituminous coals under boilers in the upper size range of the C-E Skelly Stoker.

C-E Spreader Stoker—A simple, rugged overfeed stoker designed to burn a wide variety of coals. Fines are burned in suspension and the coarser coal on a grate which may be of either stationary or dumping type. Rate of coal feed and air supply may be regulated over a wide range and are readily adaptable to automatic control. Applicable to boilers from about 100 boiler hp up.

C-E Multiple Retort Stoker—For burning bituminous and semi-bituminous coals under boilers up to the largest sizes.

C-E Traveling Grate and Chain Grate Stokers—Including both Coxe and Green types. Available with grate surfaces suitable for anthracite, coke breeze, lignite or bituminous coal, as required. Traveling grates are all forced-draft types; chain grates are either forced or natural draft types.

C-E Boilers—All fire tube and water tube types in sizes ranging from 25 hp up to the largest. Standard and special designs to suit all conditions of fuel, load and space. Included are all types formerly known by the trade names "Heine," "Walsh & Weidner," "Casey-Hedges," "Ladd" and "Nuway".

Separate Catalogs describing each of these stokers are available. A-531-A

Hershey Machine & Foundry Co.

Factory and Home Office
Manheim, Pa.

MOTOR STOKOR

Installation and Service by factory-trained dealers in all anthracite burning areas.

DEFINITION

A complete stoker for automatic combustion of buckwheat or rice anthracite. Applicable to coal, gas or oil furnaces or boilers, in anthracite burning areas. Especially designed for automatically heating buildings and providing year-round hot water.

RANGE OF TYPES

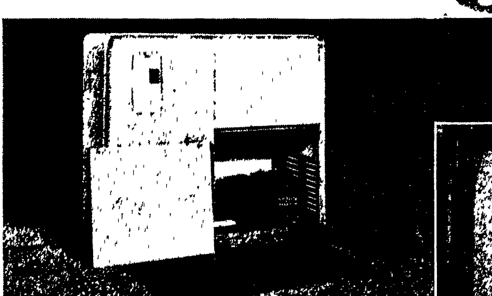
Ten different models include direct-from-bin feed with ash removal, direct-from-bin feed with pit collection, hopper feed with ash removal, and hopper feed with pit collection of ashes.

RANGE OF SIZES

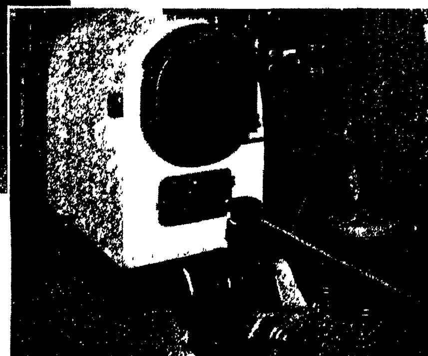
Domestic models available in sizes from 12 lb coal per hour to 65 lb coal per hour. See illustrations of models 5AF and 10AF. Commercial models available up to 100 lb coal per hour. See illustration of 2AF.

Model 10AF Motorstoker for the average home. Feeds coal from bin and deposits ashes in sealed containers. Available also for pit storage of ashes (11AF).

Model 5AF Motorstoker bin feed type with gravity ash removal into can under floor or pit of desired depth. Also available with hopper feed (5AH).



Model 5AF Motorstoker contained in a single unit with the boiler. Deposits ashes in a deep pit with ash storage capacity to last several months



Model 2AF bin feed ash removal type for large homes or commercial buildings. Also available with hopper and pit-type ash removal.

Since 1898

Detroit Stoker Company

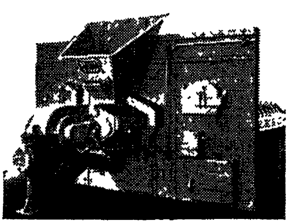
Sales and Engineering Offices
General Motors Bldg., Detroit, Mich.

Main Offices and Works at
Monroe, Mich.

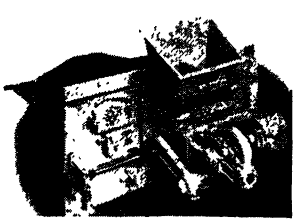
District Offices in
Principal Cities

Built in
Canada at London, Ont.

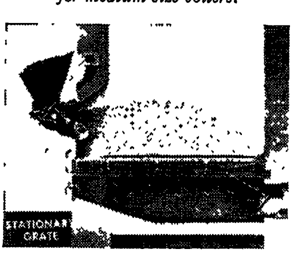
Detroit Stokers are unsurpassed for economy and dependability. They include Underfeed and Overfeed Stokers of many sizes and capacities for all types of boilers, 30 Horse Power and upwards. All grades of Bituminous Coal successfully burned. Operating costs are low. Substantial, heavy designs represent over forty years' experience in Stoker manufacture. Catalogs of various types, furnished on request.



Detroit C-D Stoker is a Single Retort, Moving Grate Stoker with Continuous Ash Discharge.



Detroit Double Retort Stoker, a multiple retort side cleaning stoker for medium size boilers.



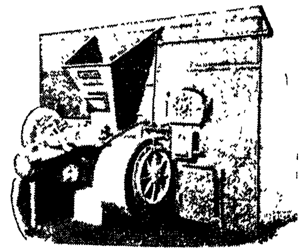
Detroit RotoStoker, (Stationary Grate Type). Ash removed through doors at grate level successfully burns a wide range of fuels.

Detroit UniStoker with Detroit Adjustable Feed (Coal Feed Control) insures accurate fuel and air supply for best economy. Single Retort, Side Cleaning, for boilers approximately 125 to 250 horsepower.

C-D Stoker—Single Retort, Moving Grate Stoker. Continuous Ash Discharge Sections at each side have a rocking movement. Rate of ash discharge, controlled at the front. Ash pit losses are low. Motor or steam ram driven. For boilers of approximately 300 to 500 Horse Power.

Detroit Double Retort Stoker, a Multiple Retort Stoker having two retorts with the side cleaning feature. For medium sized boilers having wide furnaces. Used to advantage where limited space conditions prevent the use of the rear cleaning Multiple Retort Stoker.

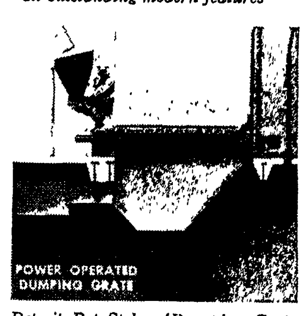
Detroit Roto-Stokers are Overfeed Spreader Type Stokers, having an *Overthrow Rotor* action, which insures uniform fuel distribution over the entire area. Offers advantages over other firing methods for burning inferior fuels and efficiently handling extremely fluctuating loads.



Detroit UniStoker with Detroit Adjustable Feed provides a wide range of coal feed control.

Detroit C-D Stokers C-D stands for Continuous Discharge of Ashes.

Detroit Multiple Retort Stoker for large boilers and high capacities. An inclined fuel bed Stoker, possessing all outstanding modern features



Detroit RotoStoker (Dumping Grate Type) (Either Power or Hand Operated) for large boilers. Particularly suited to fluctuating loads.

DETROIT LOSTOKER

Detroit LoStoker is a complete mechanical firing unit in many grate area sizes and capacities for application to all types of boilers from approximately 30 to 150 hp. Burns various grades of Bituminous Coal with high efficiency. Fuel is fed only when needed—none wasted. Single Retort, Side Cleaning, Adjustable Plunger Feed Type, mechanically driven from electric motor, requires little power for operation. Automatically controlled from steam pressure, water temperature or room thermostat. Compact, easily installed, responsive and automatic. A great coal saver.

DETROIT LOSTOKER ADVANTAGES:

Continuous Adjustable Plunger Feed with control of the quantity of coal fed and its distribution

Heavy Mechanical Drive of simple design, requires little power.

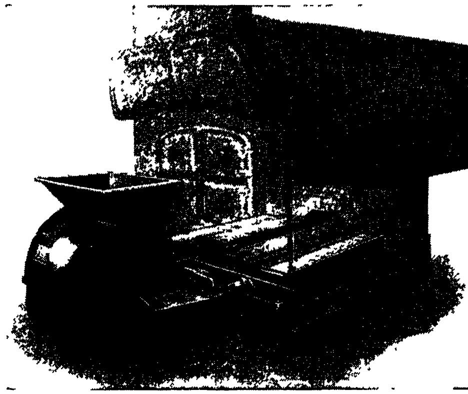
Side Cleaning with dumping grates, ashes removed through doors provided in the Stoker front. No hand cleaning.

Agitator in coal hopper for continuous coal feed, cannot stick or jam with wet coal.

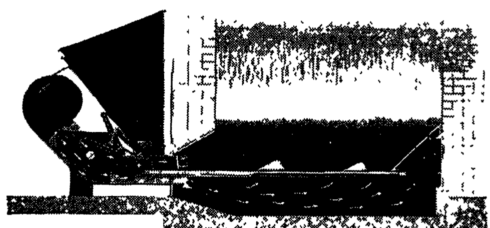
Automatically Controlled. Motor or steam turbine driven, controlled from steam pressure, water temperature or thermostat.

Many grate area sizes and capacities to fit the furnace and provide the proper grate area to readily handle heavy loads and also to operate efficiently under light load conditions.

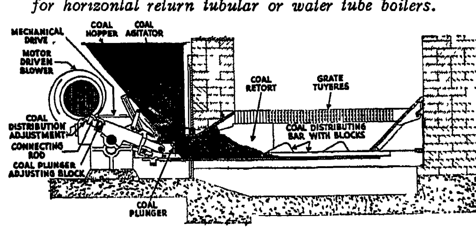
Detroit LoStoker (brickset type) for application to horizontal return tubular or water tube boilers.



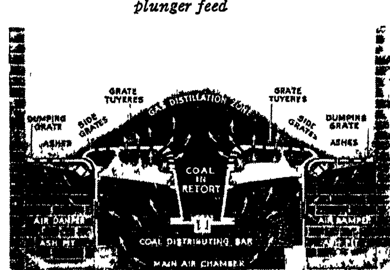
Detroit LoStoker readily applied to Firebox Boilers—built to fit the Furnace or Firebox. Coal Hopper with Agitator designed to clear Boiler Doors. Plunger Feed-side cleaning feature eliminates arduous hand cleaning of fires and corresponding losses.



Detroit LoStoker (Side elevation in brick setting) for horizontal return tubular or water tube boilers.



Detroit LoStoker side elevation showing adjustable plunger feed



Front Elevation of Detroit LoStoker (brickset type) built to fit the furnace. For use with horizontal tubular, firebox boilers on brick foundations or water tube boilers. Arrows indicate flow of air to all parts of the fuel bed.

Iron Fireman Manufacturing Company

Automatic Coal Stokers

Portland, Oregon

Factories: PORTLAND, ORE.; CLEVELAND, OHIO; TORONTO, CANADA

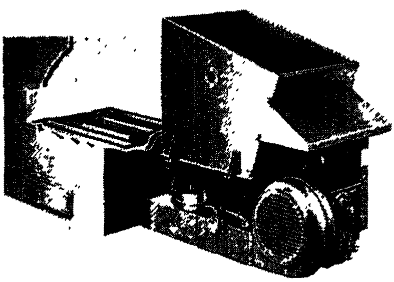
Retail Branches or Subsidiaries: CHICAGO, ILL.; MILWAUKEE, WIS ; ST. LOUIS, MO.; NEW YORK, N. Y.; BROOKLYN, N. Y ; MONTREAL, CANADA

Dealers in Principal Cities and Towns in the United States and Canada

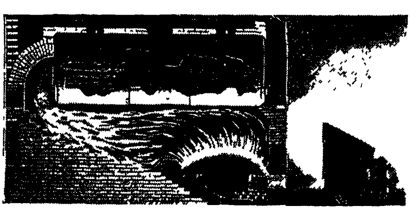
Representation in numerous foreign countries

IRON FIREMAN AUTOMATIC COAL STOKERS

COMMERCIAL HEATING MODELS



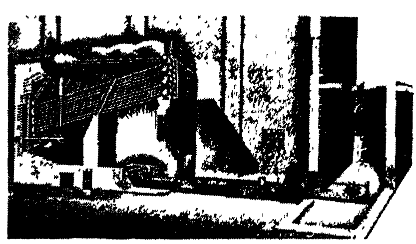
Commercial Installation—Hopper Model



Iron Fireman in Operation in Horizontal Return Tubular Boiler

Hopper Model

The Iron Fireman Commercial Heating stokers are general-purpose units, with wide application in schools, hotels, apartments, churches, office buildings, theaters and similar structures requiring a central heating plant . . . and in manufacturing plants, dairies and other establishments requiring heating and processing steam. The Commercial Heating stoker is the original type of Iron Fireman—the machine which made coal an automatic fuel.



Commercial Installation—Coal Flow model that carries coal direct from bunker to fire

MODEL	OUTPUT RANGE		
	Boiler Horsepower	Equivalent Direct Radiation	
		Steam (240 Btu)	Hot Water (150 Btu)
Standard Hopper	2 to 6	250 to 800	400 to 1,300
Deluxe Hopper	3 to 50	400 to 7,000	650 to 11,000
Coal Flow (available in all models)	3 to 500	400 to 70,000	650 to 110,000
Commercial and Industrial Standard Underfeed	20 to 350	2,500 to 50,000	4,000 to 75,000
Commercial and Industrial Poweram Underfeed	30 to 400	4,000 to 56,000	6,000 to 90,000
Commercial Anthracite	30 to 130	4,000 to 18,000	6,000 to 29,000
Pneumatic Spreader	50 to 1,000	7,000 to 140,000	11,000 to 225,000

Space Heaters—Capacities from 30,000 to 100,000 Btu per hour.

Combination Stoker—Winter Air Conditioning Units—Capacities from 60,000 to 210,000 Btu per hour.

Unit Heatmaker Room Furnace

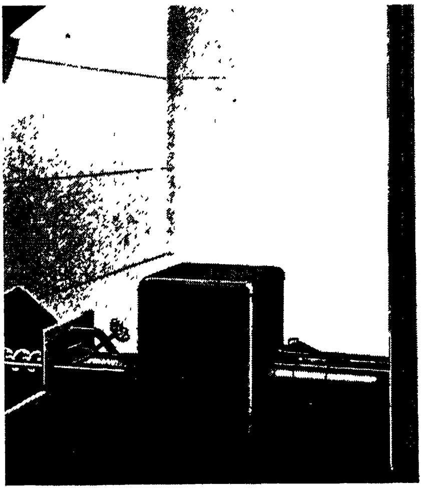
The Unit Heatmaker is a combination automatic coal stoker, room-furnace, humidifier, and forced warm-air circulator. It is made in two sizes; each size available for either bituminous or anthracite coal. This unit is particularly adaptable for small commercial and industrial heating. Forced circulation improves the distribution of heat. The Iron Fireman Unit Heatmaker is clean and quiet in operation.

Self-Firing Winter Air-conditioner

This Iron Fireman innovation produces superior winter air conditioning at extra low first cost and comparably low operating cost . . . together with the convenience of coal feeding automatically from the bin. This unit contains stoker, steel furnace, humidifier, filters and blower. May be easily adapted to existing conditions and requires a minimum amount of space. Bituminous and anthracite models.

Domestic Hopper Model

For use in warm-air, steam and hot-water systems, and also in industrial applications. Principal application is in residences. Hopper model is easily filled through a low



Domestic Coal Flow Model

opening. Quiet in operation. Mechanism is dust-tight. (Similar models for anthracite remove ash automatically). Coal Flow models feed fuel automatically from bin.

Buffalo Pumps, Inc.

450 Broadway, **Buffalo, N. Y.**

Branch Offices

ALBANY, N. Y., 1305 Standard Bldg., R. B. Taylor
ATLANTA, GA., 305 Techwood Drive, J. J O'Shea
BALTIMORE, MD., 508 St. Paul St., E. E. Thompson
BOSTON, MASS., 507 Main St., Melrose Station, E. D Johnson
CHICAGO, ILL., 20 N. Wacker Drive, L. D. Emmert
CINCINNATI, OHIO, Building Industries Bldg., F. W. Twombly
CLEVELAND, OHIO, 418 Rockefeller Bldg., T. A. Weager
DALLAS, TEXAS, 1801 Tower Petroleum Bldg., T. H. Anspacher
DAVENPORT IOWA, 305 Security Bldg, D C. Murphy Co, Inc.
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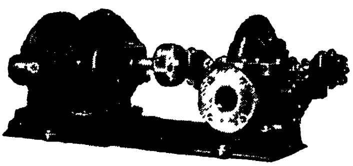
MINNEAPOLIS, MINN, 2102 Foshay Tower, E. F. Bell
NEW ORLEANS, LA., Devlin Bros., 1003 Maritime Bldg
NEW YORK, N. Y., 39 Cortlandt St., W. S Koithan
Omaha, Nebr, 106 N. 8th St, Russell Harris
PHILADELPHIA, PA., 703 Cunard Bldg., Davidson & Hunger
RICHMOND, VA., Williamson & Wilmer, Inc. Mutual Bldg.
SEATTLE, WASH., 500 First Ave., So., A. T Forsyth
ST. LOUIS, MO., 1598 Arcade Bldg., J. W. Cooper
TOLEDO, OHIO, 1922 Linwood Ave., C. M. Eyster
WASHINGTON, D. C. 512 Woodward Bldg, G. S. Franke
COMPLETE LINE MANUFACTURED IN CANADA BY CANADA PUMPS, LTD, KITCHENER, ONT.

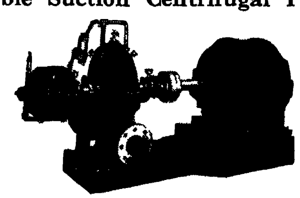
PRODUCTS—A complete line of Single and Multi-stage Centrifugal Pumps and Special Pumps for use in all types of heating and air conditioning installations.

Buffalo Double Suction Single Stage Centrifugal Pumps



For general service where clear water is handled you will get top performance with these pumps. They embody all of the accepted modern features of centrifugal pump design. Capacities range from 10 to 50 thousand U.S. gallons per minute.

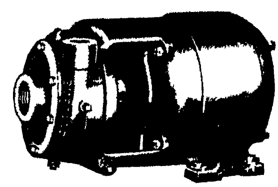
Buffalo Self-Priming Single and Double Suction Centrifugal Pumps



Now available with positive self-priming device built with the pump. This primer is built under license from the Nash Engineering Company and is fully covered by patent.

Buffalo Self-Priming Pumps offer these advantages: (1) All working parts are above the liquid to be pumped. (2) There is complete access to all parts of installation. (3) Rotors are balanced—vibrationless. (4) Buffalo Self-Priming Pumps are very quiet—no long shafts to vibrate and fewer bearings. (5) Constant positive prime obtained without foot valves.

Buffalo Single Suction Closed-Coupled Pumps



This pump is close-coupled to electric motor, eliminating the necessity for bearings. The impeller is overhung on the motor shaft, providing a compact, easily-serviced unit. Permanent alignment is assured and the pump mounted in this manner requires very little space.

Buffalo Close-Coupled Pumps are suitable for handling hot water with low submergence on suction, or for operating with suction lift as high as 25 ft.

These pumps are also available in special alloys.

Buffalo Automatic Sump Pumps

Buffalo Sump Pumps are self-contained and have unusually high efficiencies thus permitting the use of small motors. Ball bearing thrust and enclosed shaft especially adapt these pumps for their service.

Chicago Pump Company

2330 Wolfram Street **BRUnswick 4110** **Chicago**

PRODUCTS—Return Line Vacuum Heating and Boiler Feed Pumps, Condensation, House, Booster, Fire Pumps, Circulating, Brine, Sewage, Bilge, Sludge, Pneumatic and Tankless Water Supply Systems and Automatic Alternator for Duplex Sets of Pumps.

"CONDO-VAC"

Return Line Vacuum Heating and Boiler Feed Pump

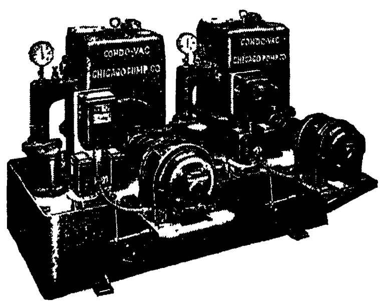


Fig. 2102—Duplex "Condo-Vacs" with Duplex Double Automatic Control

No vacuum on stuffing boxes, ample clearance in rotating member. It costs less to operate a "Condo-Vac." "Condo-Vac" reduces corrosion in piping and boiler to minimum—because pump does not take in air from atmosphere and entirely eliminates all air coming back from system. "Condo-Vac" is quiet, has a low inlet, entirely automatic, fool-proof, easy to maintain. *Ask for bulletin 270.*

Close-Coupled Pumps

Boiler Feed, Circulating, Tank Filling, Water Supply

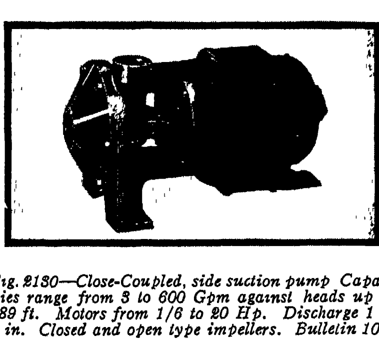


Fig. 2130—Close-Coupled, side suction pump Capacities range from 3 to 600 Gpm against heads up to 189 ft. Motors from 1/6 to 20 Hp. Discharge 1 to 3 in. Closed and open type impellers. Bulletin 108.

"Sure-Return" Condensation Pump

for Low and Medium Pressure, and Systems up to 35,000 Sq Ft Radiation

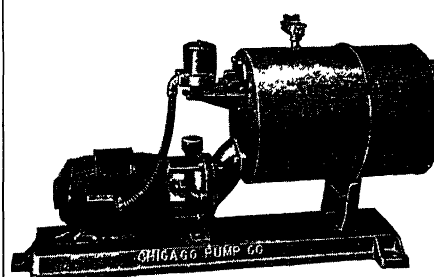


Fig. 1946

"Sure Return" Condensation Pumps and Receivers are built for systems up to 35,000 sq ft of direct radiation and for low and medium pressures. Built in either single or duplex units. Duplex units are alternated in their operation by the Automatic Alternator. *Complete data in Bulletin 250.*

Vertical Condensation Pumps

for Low and Medium Pressure for Systems from 500 to 100,000 Sq Ft Radiation

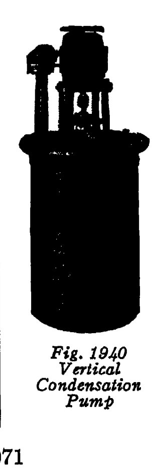


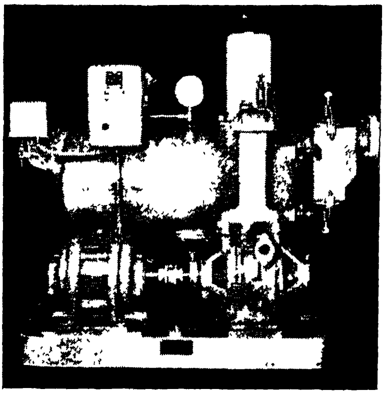
Fig. 1940 Vertical Condensation Pump

The vertical condensation pump is designed to receive returns from lowest radiation. The receiver is placed underground—an ordinary hole sufficing if necessary — and requires very little floor space. Unit is shipped complete, easy to install, assembled so as to prevent steam leaks. Special bearings will stand up under hot water for several years. A special float mechanism is guaranteed not to leak or stick in stuffing box. *Complete data and description in Bulletins 245, 253 and 255.*

The Nash Engineering Company

234 Wilson Road **South Norwalk, Conn., U. S. A.**

Sales and Service Offices in all Principal Cities

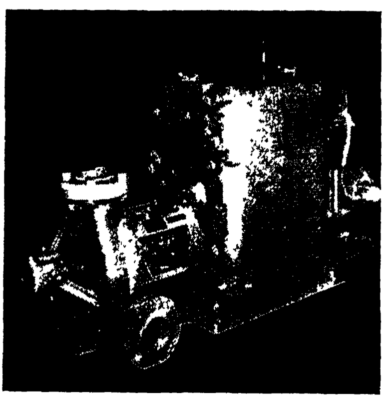


Return Line Vacuum Heating Pump

Standard with the heating industry for over seventeen years. Removes air and condensation from return lines of vacuum steam heating systems, discharging air to atmosphere and returning water to the boiler.

Two independent units are combined in a single casing—an air unit and a water unit. Impellers of both are mounted on the same shaft. Pump is bronze fitted throughout.

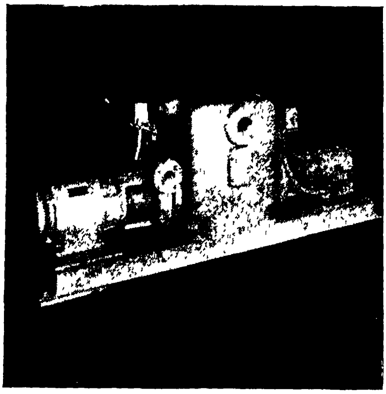
Supplied direct connected to standard electric motors, for belt drive, or for steam turbine drive. For continuous or automatic operation. Standard in capacities up to 300,000 sq ft E.D.R. Larger units special. Bulletins Nos. 307, 308, 309, and 310 on request.



Vapor Turbine Vacuum Heating Pump

Jennings Vapor Turbine Heating Pumps combine all advantages of the standard return line heating pump with a new type of drive, a specially designed low pressure turbine which operates directly on steam from the heating mains on any system, requiring a differential of only 5 in. of mercury, and returns that steam to the heating system with practically no heat loss.

This pump affords the safety and economy which goes with continuous condensation return and steady vacuum, and at no cost for electric current. Furnished standard in capacities up to 65,000 sq ft E.D.R. Larger units special. Bulletin No. 290 on request.



Condensation Pump and Receiver

Removes the condensation from radiators in return line steam heating systems, particularly radiators set below the boiler water line level, and pumps the condensation back to the boiler. Pump is bronze fitted with enclosed centrifugal impeller of improved design. By making the pump casing a part of the return tank, and bolting the motor base to the tank, floor space is conserved. The rectangular construction permits installation in a corner against the wall.

These pumps are furnished in standard sizes with capacities ranging from 1½ to 225 gpm of water. For serving up to 150,000 sq ft of equivalent direct radiation. Bulletin No. 319 on request

The Nash Engineering Company

234 Wilson Road **South Norwalk, Conn., U. S. A.**

Sales and Service Offices in all Principal Cities

Centrifugal Pump

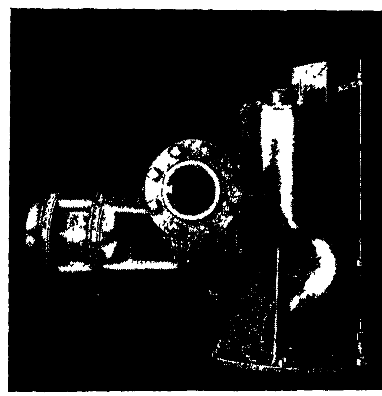
Made in standard and suction (self-priming) types. For circulating hot and cold water; boosting city water pressure; handling water in air washing and conditioning; handling ash sluicing water, etc.

Compact—motor armature and pump impeller are mounted on the same shaft. Simplified—no bearings in pump casing, one stuffing box. Accessible—impeller removable without disturbing piping or shaft alignment.

Self-priming types will handle air or gas continuously with liquid being pumped, and can be operated intermittently without foot valve.

Supplied in 1, 1¼, 1½, 2, 3, 4, 6, and 8 in. sizes, with capacity up to 2000 gpm. Heads up to 300 ft. Bulletin No. 322 on request.

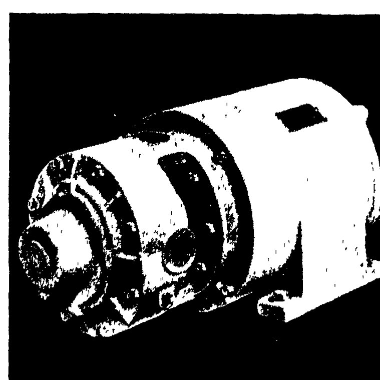
Suction Sump and Sewage Pumps



Jennings Sump Pumps are self-priming centrifugals for handling seepage water and liquids reasonably free from solids. Sewage Pumps are equipped with non-clog type impeller for liquids containing solids. Suction piping only is submerged. Centrifugal impeller and vacuum priming rotor are mounted on same shaft that carries rotor of the driving motor, forming a single moving element, rotating without metallic contact.

Will handle air or gas with liquid being pumped, and because of self-priming feature are installed entirely outside of pit, affording perfect accessibility for inspection or cleaning. Capacities to meet all requirements. Bulletins Nos. 159, 161, and 338 on request.

Air Compressor and Vacuum Pump



Nash Air Compressors operate on a unique and different principle. The one moving part rotates in casing without metallic contact. There is nothing to wear, and no internal lubrication.

Nash Compressors deliver absolutely clean air; ideal for agitation of liquids, pressure displacement, and handling gases. Vacuum pumps ideal for priming pumps, blood sucking pumps in hospitals, and wherever non-pulsating vacuum is required.

Pressure 75 lb or vacuum 27 in. of mercury. Furnished for any capacity; special for higher vacuums and pressures. Bulletins Nos. 282, 325, 331 and 337 on request.

ADSCO
PRODUCTS for STEAM SERVICE

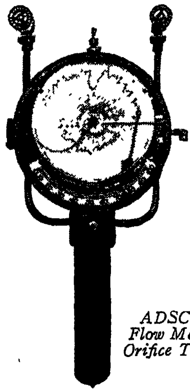
AMERICAN DISTRICT STEAM COMPANY

NORTH TONAWANDA, N Y.

IN BUSINESS OVER SIXTY YEARS

Branches and Agents in Principal Cities

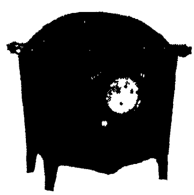
For Data on ADSCO Expansion Joints, refer to Insulation, Underground, page 1117.



ADSCO Flow Meter Orifice Type

ADSCO FLOW METER—ORIFICE TYPE

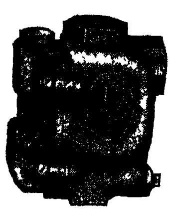
Exceptionally accurate at all rates of flow and will meter steam, water, gas or air. It is a compact unit for indicating, recording and integrating the flow and can be furnished in other combinations of these three devices. Easily installed and maintained by the purchaser. Frictionless meter mechanism, records on evenly-divided, direct-reading chart, giving a daily record from which to determine heating or processing costs. Write for Bulletin No. 35-83G.



Rotary Condensation Meter

ROTARY CONDENSATION METER

Measures steam consumption by metering condensate from heating systems or industrial equipment. Accurate within 1 per cent and factory tested to 150 per cent of rated capacity. Compact, easily cleaned, tamper-proof and equipped with non-fogging counter mechanism. Counter reads directly in pounds. Suitable for vacuum or gravity service. Available in 7 sizes from 250-12,000 lb per hour capacity. Write for Bulletin No. 35-80AG.



ADSCO Vertical Steam Trap

ADSCO VERTICAL STEAM TRAP

A float type steam trap with or without thermostatic air by-pass for vacuum service to 15 lb pressure and gravity service to 125 lb pressure. The cover with all working parts can be removed without disturbing the piping connections. The trap is equipped with a reversible valve and reversible seat of stainless alloy steel. Write for Bulletin No. 35-86G.

ADSCO Instantaneous Water Heater

ADSCO HEAT EXCHANGERS

Made in various sizes and capacities to heat or cool water, oils, other liquids or gases according to expert engineering specifications. Simple in design, sturdy in construction, dependable and economical in operation. Available in U-tube or straight tube types of heaters, economizers, condensate coolers or special units. Write for Bulletin No. 35-75BG, 35-76G.

E. B. Badger & Sons Co.

General Office: 75 Pitts Street, **Boston, Mass.**

Representatives

ATLANTA, GA.....140 Edgewood Ave.	LONDON, ENGLAND.....153, Moorgate
BIRMINGHAM, ALA.....444 Brown-Marx Bldg.	LOS ANGELES, CALIF.....609 S. Anderson St.
BUFFALO, N. Y.....361 Delaware Ave.	MINNEAPOLIS, MINN.....732 Builders Exchange
CHARLOTTE, N. C.....1408 Independence Bldg.	MONTREAL, QUEBEC.....1411 Crescent St.
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CINCINNATI, OHIO.....831 Temple Bar Bldg.	NEW YORK, N. Y.....271 Madison Ave.
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INDIANAPOLIS, IND.....825 Occidental Bldg.	ST. LOUIS, MO.....4060 West Pine Blvd.
KANSAS CITY, MO.....1332 Oak St.	VANCOUVER, B. C.....1396 Richards St.

ENGINEERS AND MANUFACTURERS

Manufacturers of Copper and Stainless Steel Badger Corrugated Expansion Joints; Engineers and Manufacturers of Chemical Apparatus; Engineers on Process Work; Designers of Complete Plants.

More than forty years' experience in design, manufacture and application are back of BADGER EXPANSION JOINTS. Most recent developments emphasize the constant study Badger engineers are giving to expansion joint development:

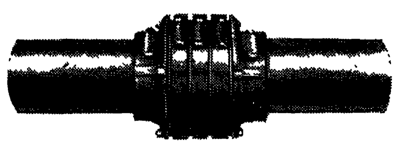
1 . . . **Application of Heat Treatment** . . . scientific heat treatment is applied throughout the fabrication of Badger Expansion Joints with the result that the buyer gets all the benefits of this important metallurgical step.

2 . . . **Directed Flexing** . . . involving a new design corrugation and equalizing ring, resulting in much longer joint life. The all-curve Directed Flexing corrugation distributes flexing stresses which, with straight-sided corrugations, tend to localize.

3 . . . **Stainless Steel Joints** . . . perfected after years of study and testing with this useful metal . . . now practicable to use the packless type of joint for high temperatures and high pressure conditions.

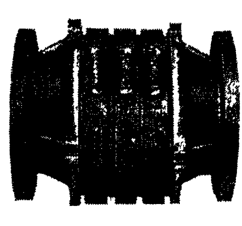
The BADGER Expansion Joint is the packless type. Requires no servicing throughout its long life. Ideal particularly for underground use or in cramped quarters. Wide range of traverse.

BADGER
Self-Equalizing, Directed Flexing, Expansion Joint



Designed for traverses ranging from fractions to 6 inches single and 12 inches double; for pressures ranging from high vacuum to 200 pounds (copper) and 300 pounds (stainless steel); and for temperatures ranging from sub-zero to 500 F (copper) and 900 F (stainless steel). List prices, installation and other data in **Bulletin 100.**

Welding End and Flanged End, Directed Flexing, Self-Equalizing Expansion Joints.



BADGER
Non-Equalizing Expansion Joint

Designed principally for traverses up to ½ inch and for pressures up to 25 pounds; also good as the connecting element between adjacent equipment to absorb vibrations or limited lateral displacements; standard shapes: round, oval, square or rectangular; special shapes to order. **Bulletin No. 200.**

BADGER Flexible Pipe Line Seal

Designed to be used on pipe passing through walls, foundations or bulkheads, the purpose being to allow expansion and contraction but to seal the opening against seepage of ground or other waters.

Bulletin No. 300.

Armstrong Machine Works

Representatives in All Principal Cities

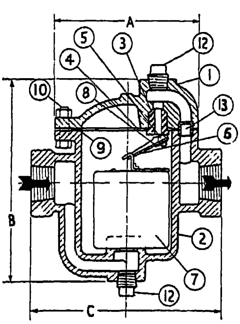
851 Maple Street **Three Rivers, Mich.**

Armstrong offers two types of traps for heating, air conditioning, and steam distribution service.

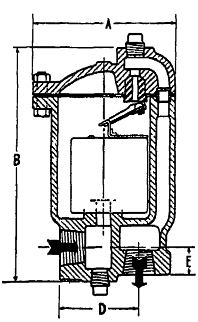
Standard Inverted Bucket Traps, the type originated by Armstrong, are *non-airbinding* and *self-scrubbing*. They are used for low, medium, and high pressure service where relatively little air must be handled along with the condensate. Their free-floating lever design makes it possible to open very large discharge orifices compared with the size of the trap itself.

Armstrong Blast Traps are used where large amounts of air must be vented quickly when steam is first turned on. They have several advantages over the conventional float and thermostatic trap.

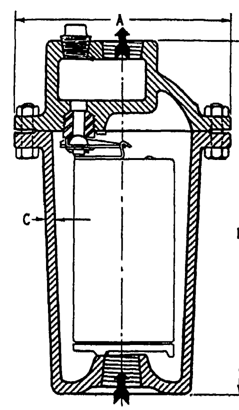
1. The Armstrong Blast Trap has but a *single orifice* to be maintained tight against the full pressure differential.
2. *Positive action.* The discharge valve in an Armstrong Blast Trap is either wide open or tight shut. Fast opening and fast closing prevent wire-drawing.
3. *Handles dirt.* There are no dead spots in an Armstrong Trap in which dirt can settle and interfere with the operation of the trap.



Cross-section of No. 800, 811, 812 and 813 traps for straight-through pipe connections.



Cross-section of No. 801 trap for standard angle pipe connections.

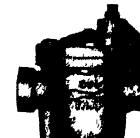


No. 211-216, Bottom inlet Type

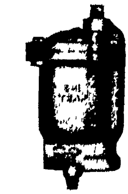
Side Inlet Traps

Trap Size	No. 800	No. 811	No. 812	No. 813	No. 801
Pipe Connections	1/2" or 3/4"	1/2" or 3/4"	1/2" or 3/4"	3/4" or 1"	1/2" or 3/4"
List Price (Regular)	$7.00	$10.00	$16.00	$22.00	$7.00
List Price (Blast Trap)	$8.50	$11.50	$18.00	$24.00	$8.50
Telegraph Code (Regular)	Aloe	Brown	Cherry	Dawn	Arrow
Telegraph Code (Blast Trap)	Aloette	Brownette	Cherette	Dawnette	Arrowette
Dimension A	3 3/4"	3 3/4"	5 5/8"	7"	3 3/4"
" B	5 1/8"	6 9/16"	8 11/16"	11 1/4"	6"
" C	5"	5"	6 1/2"	7 3/4"	..
" D	. . .	. .	.	.	2 1/16"
" E	. . .	. .			11/16"
Number of Bolts	6	6	6	6	6
Diameter of Bolts	1/4"	1/4"	3/8"	1/2"	1/4"
Weight	4 1/2 lbs.	5 1/2 lbs.	13 1/2 lbs.	25 lbs.	4 1/2 lbs.
Maximum Pressure, lbs.	125	250	250	250	125

Continuous discharge capacity in lb of water per hour at pressure indicated. For more complete information see the Capacity Chart in Armstrong Steam Trap Book. — Lb Pressure	No. 800	No. 811	No. 812	No. 813	No. 801
5	450	830	1600	2900	450
10	560	950	1900	3500	560
15	640	1060	2100	3900	640
20	690	880	1800	3500	690
30	500	1000	2050	4000	500
50	580	840	1900	4100	580
70	660	950	2200	3800	660
100	640	860	1800	3600	640
125	680	950	2000	3900	680
150	. .	810	1500	3500	. .
200	. . .	720	1200	3200	. . .
250	.	760	1300	3500	. .



No. 800



No. 801

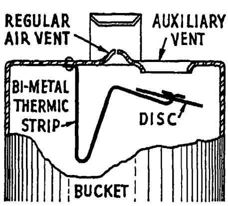
MATERIALS USED IN ARMSTRONG TRAPS

Part No.	Name	Material
1 & 2	Cap and Body	Cast Semi-Steel, 33,000 lb tensile strength
3	*Seat	Chrome Steel
4	Guide Plate Pins	Stainless Steel
5	*Valve	Chrome Steel
6	Lever	Stainless Steel
7	Bucket (drawn in one piece for No. 800, 801, 211 and 212)	Stainless Steel
8	Retainer	Stainless Steel
9	Gasket	Compressed Asbestos 1/32"
10	Bolts and Nuts	Steel 90,000 lb minimum tensile
11	Thermic Vent	Stainless Bi-metal and Stainless Steel
12	Test Plug	Steel

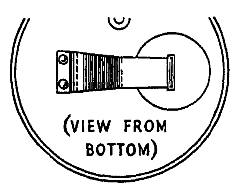
*Valve and Seat are heat treated after machining.

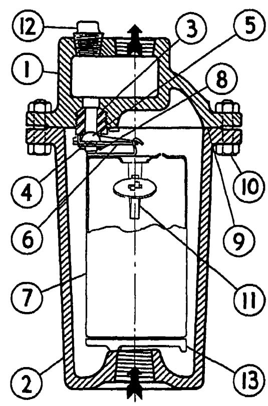
4. The wearing parts in all Armstrong Traps are identical in design, material, and precision workmanship with parts used in Armstrong Forged Steel Traps for pressures up to 1500 lb gage and total temperatures of 850 F.

Armstrong Steam Trap Book. This 36 page book gives complete information on all sizes and types of Armstrong Traps. It also contains 17 pages of data on the subject of trap selection, installation, and maintenance. A free copy will be mailed on request.



FOR BLAST TRAP JOBS





No. 211-216, Blast Type

ALL Armstrong traps are readily convertible into "Blast" type traps merely by using buckets equipped with the patented auxiliary thermic air vent. As shown in the above sketches, the mechanism for this vent consists of a stainless steel disc slotted to receive the end of a bi-metal strip. Different coefficients of expansion in the bi-metal cause it to bend down when cold and up when hot. Normally, it is set to close at 212 deg, but it can be set to close at higher temperatures. Capacity, 50 to 100 times the air-venting capacity of a standard trap.

Bottom Inlet Traps

Trap Size		No. 211	No. 212	No. 213	No. 214	No. 215	No. 216
Pipe Connections		1/2"	1/2" or 3/4"	1/2" or 3/4"	1"	1" or 1 1/4"	1 1/2" or 2"
List Price (Regular)		$ 9.25	$15.00	$20.75	$29.00	$38.00	$55.00
List Price (Blast Trap)		$10.75	$17.00	$22.75	$31.50	$40.50	$60.00
Telegraph Code (Regular)		Aspen	Birch	Walnut	Hemlock	Larch	Tamarack
Telegraph Code (Blast Trap)		Aspette	Birette	Walette	Hemlette	Larette	Tamrette
Height ... Dimension B		6 3/8"	8"	10 1/4"	12 1/2"	14"	16 3/4"
Diameter " A		4 1/8"	5"	6 3/8"	7 1/2"	8 1/2"	10 5/16"
Wall Thickness " C		5/16"	1/4"	9/32"	3/8"	3/8"	7/16"
Diameter of Bolts		1/4"	1/4"	3/8"	3/8"	1/2"	1/2"
Number of Bolts		6	8	6	8	8	12
Weight		5 1/2 lb	10 1/2 lb	19 lb	32 lb	47 lb	80 lb
Maximum Pressure		250	250	250	250	250	250
Continuous discharge capacity in lb of water per hour at pressure indicated. For more complete information, see the Capacity Chart in the Armstrong Steam Trap Book.	Lb Pressure 5	830	1600	2900	4800	7600	14500
	10	950	1900	3500	5800	9000	17300
	15	1060	2100	3900	6500	10000	19200
	20	880	1800	3500	6000	8500	18500
	30	1000	2050	4000	6800	9800	18000
	50	840	1900	4100	6300	9000	18200
	70	950	2200	3800	6000	9200	18300
	100	860	1800	3600	6200	10400	18000
	125	950	2000	3900	6700	10900	20000
	150	810	1500	3500	5700	9500	18500
	200	860	1600	3200	5300	9200	17500
	250	760	1300	3500	5700	7000	19000

Cochrane Corporation

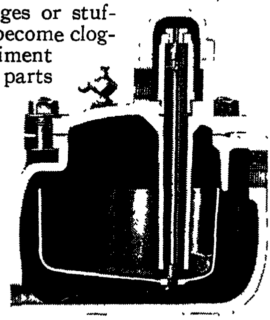
3130 North 17th Street, **Philadelphia, Pa.**

Branch Offices in 40 Principal Cities

COCHRANE HEAVY-DUTY STEAM TRAPS

A high pressure unit for condensate drainage of steam lines, separators, coils, evaporators, etc., and for conditions involving relatively high drainage rates. Recommended for pressures up to 400 lb.

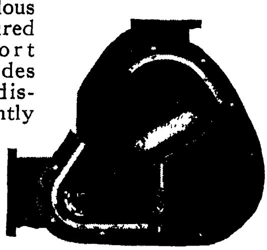
Simple construction. No levers, constricted passages or stuffing boxes to become clogged with sediment or scale. All parts are readily accessible. Action is quick and positive, avoiding wire drawing and erosion.



Write for publication No. 2850.

MULTIPORT DRAINERS

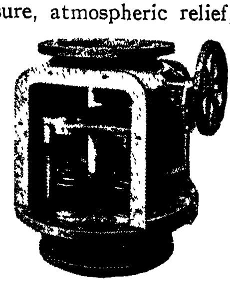
Of the multiport type, they afford unusual capacity for removing condensate or drips from purifiers, separators, jackets, radiators, pressure heating or drying coils, etc. Eliminating condensate delivers maximum heat from steam production at lower cost. Tremendous capacity assured by large port areas. Provides continuous discharge. Instantly responsive. Compact and light in weight. For pressures up to 150 lb.



Multiport Drainer

COCHRANE MULTIPORT RELIEF VALVES

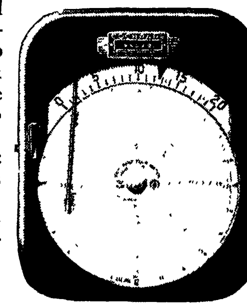
For back pressure, atmospheric relief, flow or check valve service on air, gas, steam or water lines. Positive protection against stuck, jammed or "frozen" valves as a number of small disks are used instead of one large disk. Write for publication No. 2870.



Multiport Back Pressure Valve

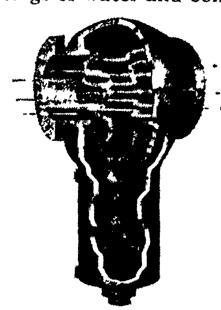
COCHRANE FLOW METERS

Flow meters of both mechanical and electrical types for measurement of steam, liquids and gases. Mechanical meter uses no working parts in the pressure chambers and no stuffing boxes. The electric meter measures flow by the extremely accurate galvanometer null principle. The new "Linameter" measures corrosive or viscous fluids. Publication 3010.



ALL-SERVICE SEPARATORS

Cochrane Separators purify steam by separating out oil, slugs of water and condensate. Complete removal of entrainment is accomplished by vertical baffle ribs which guide it into a direct unrestricted fall, and a baffle area which extends far beyond the flow from the inlet pipe. Ports at the sides of the baffle prevent the purified steam from passing over the drip area and coming into contact with the entrainment. The steam flow is uninterrupted and pressure loss is minimized.



All-Service Separator

COCHRANE-BECKER HIGH PRESSURE CONDENSATE RETURN SYSTEM

In unit heaters, coil radiation, blast heaters, etc., this sytem will reduce fuel costs by return of condensate direct to boiler at temperatures comparable to that of the pressure of steam utilized.

Based on a jet pumping principle, it accomplishes complete removal of air and non-condensible gases together with the condensate from the equipment drained, thus improving heat transfer rates.

Operates in a closed circuit from which all air is automatically vented. Made in sizes to handle from 50 to 1500 B. H. P. and at pressures up to 150 pounds. Write for Bulletin 3025.

GRINNELL COMPANY, INC

Heating, Industrial and Power Plant Piping, Fittings, Hangers, Valves, Pipe Bending, Welding, Piping Supplies, Etc.

Executive Offices: **Providence, R. I.**

National Distributors of Thermoflex Traps and Heating Specialties

For data on other Grinnell Products, see pages 1020-1022

Thermoflex Specialties

The heart of all Thermoflex Traps is the Hydron Bellows.

The Hydron Bellows is formed under hydraulic pressure. This powerful internal pressure locates any weakness of any nature in the tubing. Such hydraulic pressure is many times more severe than any pressure the Trap will ever be called upon to control. Every Thermoflex Trap, therefore, is practically indestructible.

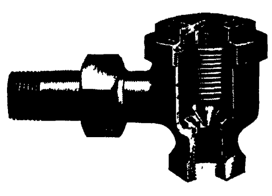
Thermoflex Traps have an exceptionally large orifice. This large orifice combined with high lift, insures fast action and freedom from clogging.

We supply Thermoflex Traps guaranteed for steam pressures of 25 lb, to 50 lb and to 125 lb. Complete information and details of typical installations will be gladly sent on your request. Ask for Catalogue on Thermoflex Heating Specialties.

Valves, Traps, Gauges, Etc.

The Thermoflex line includes: Radiator Traps, Offset Traps, Blast Traps, Drip Traps, High Pressure Traps, Vent Traps, High-grade Packless Inlet Valves, and the Thermoflex Alternator, Thermoflex Compound Gauge, Thermoflex Damper Regulator.

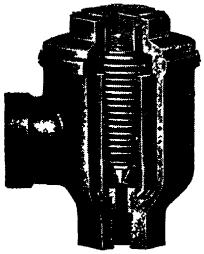
No. 12 Thermoflex Radiator Trap



The full eight-fold Thermoflex-Hydron Bellows is guaranteed because of the Hydron-forming process. Body is heavy bronze construction throughout, with renewable seat.

Fully nickel-plated with highly polished trimmings. The No. 12 is made in angle and in corner patterns, with ½ in. inlet and ½ in. outlet tappings. The inlet neck is double thick to allow for expansion strains. Guaranteed for steam pressures up to 25 lb.

Thermoflex High Pressure Traps



The No. 100A Thermoflex Trap is guaranteed for steam pressures from 50-125 lb. Must not be used where the steam temperature exceeds 400 F.

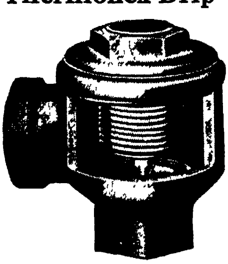
For use with all types of process work, Laundry Machinery, Kitchen Equipment, Hospital Sterilizers, Vulcanizers, Dry Kilns, Unit Heaters, Street Steam Service, etc., in fact any place that a trap is desired for service at the above pressures.

Small, compact and inexpensive.

Extra heavy body. Renewable nickel steel seat and disc. Bellows made from special bronze tubing and encased in brass sleeve to prevent distortion due to pressure.

Regularly furnished without unions, plain nickel finish. Can be furnished with unions, polished nickel or chromium plated at extra cost.

No. 4 Thermoflex Drip Traps



Used for dripping mains, risers, coils and unit heaters. Semi-steel body, bronze cap and inserted renewable bronze seat, angle pattern only, without unions. Can be used for any general purpose where a finished, nickel-plated trap is not necessary, and at a lower cost. Guaranteed for steam pressures up to 25 lb.

Kieley & Mueller, Inc.

Established 1879

Engineering Specialties for Pressure and Flow Control

40 West 13th Street, **New York, N. Y.**

Factory: NEWARK, N. J.

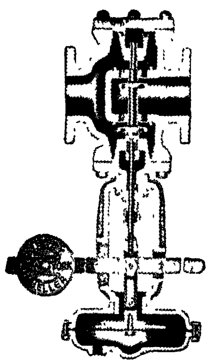
Agents in All Principal Cities

PRODUCTS—Valves: Altitude, Stop and Check, Pressure Regulating, Float, Pilot Reducing, Back Pressure, Tank Control.

Liquid Level Controllers, Water Feeders, Pump Governors, Steam Traps, Y-Type Strainers.

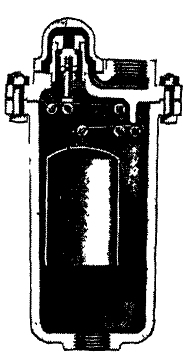
Also Damper Regulators, Hot Water Temperature Controllers, Oil Separators, Steam Separators, Return Traps, Water Columns, etc.

Catalogs sent upon request



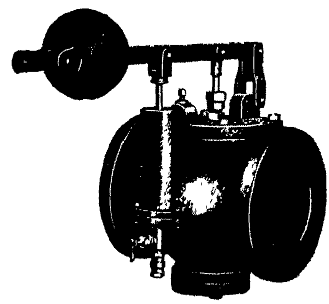
Pressure Regulating Valve

Spring and lever weighted valves for all services and for initial pressures up to 250 lb and reduced pressures from 0 to three-quarters of the initial pressure. Single or double seated in sizes ⅜ to 16 in. Suitable for steam, water, air, oil and gas. Controlled by a small feeler pipe connected from diaphragm to low pressure side.



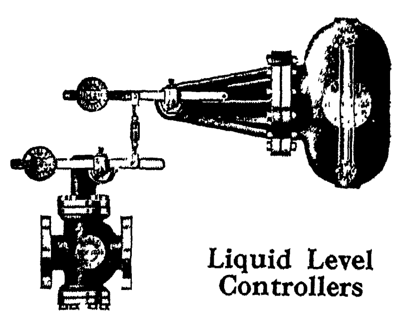
Steam Traps

Large capacity, small sized inverted bucket traps; quick-acting, self-cleaning and non-air binding. Sizes ⅜ to 2 in. Pressures up to 250 lb. Body and cover, semi - steel. Valve and seat, stainless steel. Removable cap allows inside inspection or replacement of valve parts without disturbing pipe connections. (All parts are interchangeable).



Back Pressure and Atmospheric Relief Valve

For use where plant is operated either condensing or non-condensing. Outside air dash pot insures noiseless operation. Maintains exhaust line back pressure from 0 lb to 25 lb. Made horizontal or vertical lever and weight or spring operated.



Liquid Level Controllers

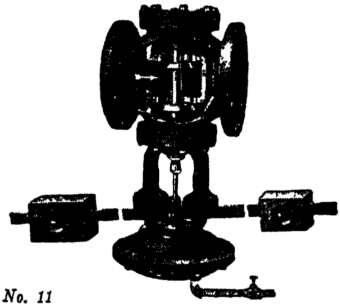
For the accurate control of liquids in tanks or other vessels; suitable for use in industrial plants, gasoline plants, refineries, etc. Direct connected or remote control; ball bearing spindle and easy-to-pack stuffing box; rotary or sliding valve. Write for special bulletin C-3.

Mueller Steam Specialty Co., Inc.

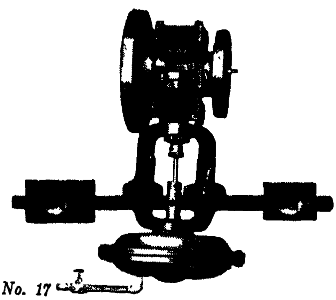
40-20 22nd Street, **Long Island City, N. Y.**

Steam, Water, Air, Oil and Gas Specialties for Heating and Power Plants

Pressure Reducing Valves—Straight Pattern and With Increased Outlet



No. 11



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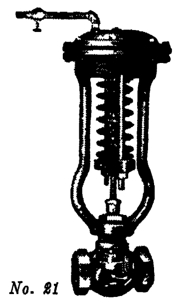


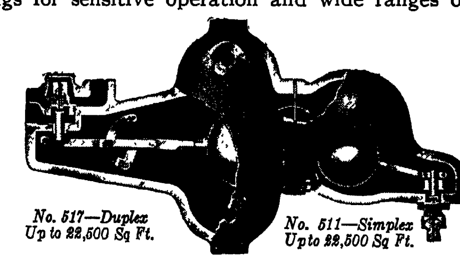
No. 21

No. 11—For Vacuum, Vapor and Low Pressure Heating Systems. Initial Pressures, up to 200 lb; Reduced Pressures, 0 to 10 lb.

No. 17 and 21—For automatic control of reduced pressures on dead-end service, requiring a tight closing valve, such as tank heaters, kitchen utensils, sterilizing apparatus, laundry equipment, kettles, cookers, driers, etc. Initial Pressures up to 200 lb. Reduced Pressures 0 to 150 lb.

Constructed with full globe bodies. Center guide eliminates the wings on discs, and increases efficiency, assures minimum noise and prolongs the life of the seats and discs. Lever and weight operates on a steel roller bolt, assuring a most sensitive valve. Spring type furnished with special long springs for sensitive operation and wide ranges of reduced pressures.

Automatic Water Feeders



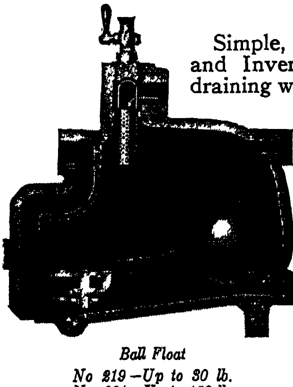
No. 517—Duplex Up to 22,500 Sq Ft.

No. 511—Simplex Up to 22,500 Sq Ft.

With a powerful leverage to control the water line in steam boilers, etc. They supply make-up water to compensate for evaporation, leaks, steam utilized in process work and condensation wasted. Where condensation held up in the system eventually returns in large quantities, our Duplex type protects the boiler against flooding. All working parts of non-corrosive metal, are accessible without breaking pipe connections. Provided with an integral strainer. For steam pressures up to 100 lb, water pressures up to 120 lb.

Equipped with single and double contact mercury Tube Switches for all services.

Steam Traps



Ball Float
No 219—Up to 30 lb.
No. 221—Up to 150 lb.
Sizes ½ to 3 in.

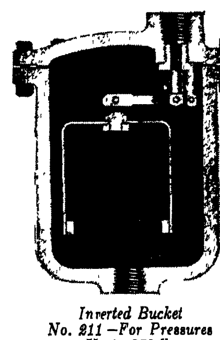
Simple, Sturdy and Compact Ball Float and Inverted Bucket Steam Traps for draining water of condensation from steam apparatus and steam mains.

Powerful leverage enables them to take care of large quantities of condensation.

Ball Float Steam Traps equipped with integral strainer, water gages, air cocks, blow-off and integral by-pass valve, when desired.

All working parts are accessible without disturbing any pipes.

Valves are sealed with several inches of water, making the escape of steam impossible.



Inverted Bucket
No. 211—For Pressures Up to 250 lb.
Sizes ½ to 2 in.

Catalogue and Bulletins covering our Complete Line gladly furnished on application.

Wright-Austin Co.

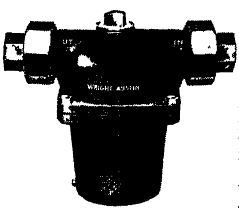
317 West Woodbridge St., **Detroit, Mich.**

PRODUCTS—Steam Traps, Strainers, Air Traps, Steam and Oil Separators, Compressed Air Purifiers, Exhaust Heads, Boiler Feeders and Controllers, Alarm Water Columns, Water Gauges, Trycocks.

"Airxpel" Bucket Type Steam Traps

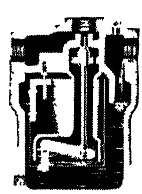
Are "double duty" traps, because they automatically discharge both air and condensate.

Union connections make them easy to connect up. Also, furnished with screw connections when desired. They save money for fittings and installation labor, by having straight through horizontal pipe connections.



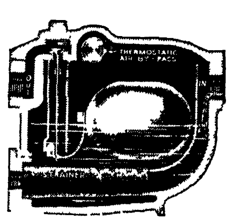
The Cub sizes are made in ½ in., ¾ in., 1 in. Especially suitable for individual unit drainage on heating and process equipment.

Also three "Master" sizes ½ in. to 2 in., for general service.

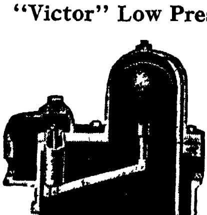


"Combination" Steam Traps

Float Type with internal thermostatic air bypass and strainer for pressures 0 to 40 lb. A modernly designed and very successful trap for vacuum and pressure heating.



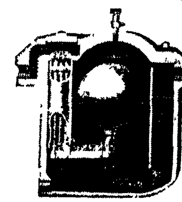
"Victor" Low Pressure Steam Traps

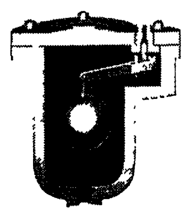


A heavy duty trap for large volumes of condensation at low pressures.

"Emergency" Float Type Steam Trap

Three valve trap with large capacity at high pressures. An exceptionally reliable trap for use in inaccessible places.



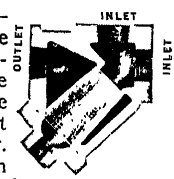


Air Relief Traps

For relieving air from forced circulation hot water heating systems, water supply lines, closed tanks, receivers, pumps, etc.

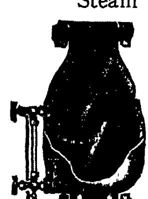
"Tuway" Strainer

May be used two ways—as a straight-way or angle strainer, in either horizontal or vertical pipe line, because it has the choice of two inlets at right angles to one another.



For cleaning, flush through blow-off connection, or remove screen by unscrewing bottom plug.

Separators—Steam and Oil

Type "A" Vertical Steam	Type "S" Horizontal Oil
	

We make separators of every type and all sizes for all pressures.

Exhaust Heads

Designed to eliminate noise and spray. Three types to select from—the "Cyclone" Heavy Duty, and Standard Galvanized Steel—also, the cast iron type, to remedy all conditions. Sizes 1 in. to 48 in.

Send for descriptive Bulletins on any of the items listed on this page.

Yarnall-Waring Company

Manufacturers of **YARWAY** Steam Specialties

7600 Queen Street, **Philadelphia, Pa.**

YARWAY IMPULSE STEAM TRAPS

Construction—The Yarway Impulse Steam Trap is unique in that there is only one moving part, the simple valve F. This trap is made of bar stock throughout, no castings used. Body and bonnet of cold rolled steel, cadmium plated; cap of tobin bronze, valve and seat of heat treated stainless steel. For pressures 400 to 600 lb, bonnet and cap are stainless steel.

Operation—Movement of the valve is governed by changes in pressure in control chamber (K). When handling ordinary condensate, tiny control flow bypassing through orifice in center of valve reduces chamber pressure below inlet pressure and valve opens, allowing free discharge through seat. As condensate approaches steam temperature, low chamber pressure causes flashing, flow through center orifice is choked and pressure builds up in control chamber closing valve (F).

Advantages

Light Weight—Yarway traps need no support—½ in. trap weighs only 1⅜ lb. 2 in. trap weighs 8⅝ lb.

Small Size—They practically eliminate radiation losses—can be installed in cramped quarters—½ in. trap measures 2¼ in. long—2 in. trap, 4¾ in. long.

Will not air bind.

Require no priming.

Insure quick heating.

Operate on exclusive Impulse principle (*U. S. Patents No. 2,051,732 and 2,127,649.*)

Low Price—Often cheaper than repairing old traps.

Factory set to operate at all pressures up to 400 lb (or 600 lb) without change of valve seat.

List Prices, Weights and Dimensions

No. 60 Series—up to 400 lbs. and

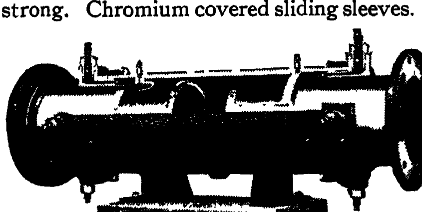
No. 70 Series—up to 600 lbs.

Size	Trap Complete	Weight Pounds	Length Inches
½″ Nos. 60 or 70	$15.00	1¼	2⅝
¾″ Nos. 61 or 71	22.00	2	3
1″ Nos. 63 or 73	31.00	2½	3⅜
1¼″ Nos. 64 or 74	48.00	4	3¾
1½″ Nos. 66 or 76	68.00	5¾	4¼
2″ Nos. 67 or 77	90.00	8½	4¾

For further information send for descriptive bulletin T-1735.

YARWAY GUN-PAKT EXPANSION JOINTS

All-steel welded construction; light but strong. Chromium covered sliding sleeves.



Cylinder guide and stuffing box integral, assuring perfect alignment. Internal limit stops. Gun-pakt and Gland-pakt types; Gun-pakt (illustrated) fitted with screw guns which permit insertion of plastic packing while joint is under pressure. Sizes 2 in. to 24 in., single end or double end, flanged or welding ends; 150, 300 and 400 lb pressures. For additional details send for bulletin EJ-1908.

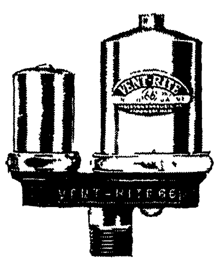
Anderson Products, Incorporated

Cambridge, Massachusetts

Vent-Rite Controlled-Venting Radiator Valves . . . Vent-Rite No. 66 Control Valves . . . Vent-Rite Balancer . . . Vent-Rite Unit Heater Valve. Originators of "Balanced Radiation by Controlled Venting," "The Vent-Vac Method" and "Vacuum Limitation."

THE VENT-VAC METHOD

The Vent-Vac Method provides more even room temperatures. This is accomplished by continuing the distribution of steam between firing periods. The steam is available through the use of heat left in the boiler, and it is distributed to the points of greatest heat loss. To insure fast, uniform distribution of steam during the firing periods, it breaks the vacuum used between firing periods for this purpose. This "breaking" of the vacuum occurs as soon as firing starts, restoring the system to atmospheric pressure. Vent-Rite Vacuum Valves, and a Vent-Rite Control Unit are used. The system is simple, economical, and amazingly effective. Vent-Rite Control Units not only create vacuum in the system between firing periods, but also limit the amount of vacuum that can be created to the point beyond which the distribution of excessively expanded vapor would be inefficient. This is another feature developed and pioneered by Vent-Rite and offered only in Vent-Rite Units.



No. 66

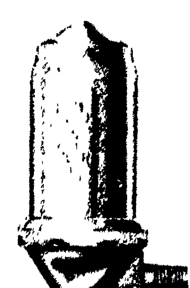
VENT-RITE CONTROL VALVES

Vent-Rite Control Valve No. 66 is the heart of the Vent-Vac Method of steam control for automatically-fired, one-pipe systems. It takes the place of a main line vent, limits the amount of vacuum created and breaks the vacuum at the beginning of the firing period. It is entirely mechanical. With the Vent-Vac Method, using a No. 66 Control Valve, a system is "Vacuum" between Firing periods, "Non-Vacuum" during Firing, combining the best of both systems, assuring "Balanced Radiation."

The Vent-Rite Line includes Nos. 1, 51, 3, 5A and 55 (Non-Vacuum); 2, 62, 4, 6A, 66, 68 and the Balancer (Vacuum).

VENT-RITE RADIATOR VALVES

Vent-Rite Controlled-Venting Radiator Valves are made in a wide variety of types, sizes, outlets, and venting capacities. Both Vacuum and Non-Vacuum. All are noiseless in operation, positive in action, close thermostatically under temperature. They may be taken apart for examination and cleaning. Venting is through an adequate straight-line venting orifice, accurately set by a newly designed inconspicuous stream-lined Adjusting Disc. For 1942 Vent-Rite also offers a new Siphon Tongue for use especially with small-tube radiation.



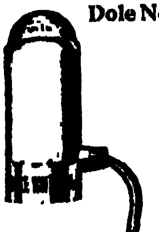
No. 1

The Dole Valve Company

Main Offices and Factory: 1901-1941 Carroll Avenue, **Chicago, Ill.**

THE ALL STAR LINE — Dole — **AIR AND VACUUM VALVES**

Selecting the right vent for a particular purpose is your assurance of the utmost efficiency and economy from one pipe steam heating systems. The Dole line covers every venting need and offers a complete choice for every purpose.

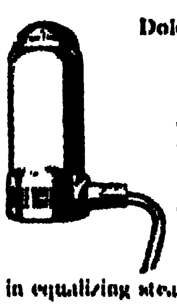


Dole No. 1A Vari-Vent Air Valve

Modern gas, oil or stoker fired one pipe steam systems require QUICK venting. This radiator valve lets air escape twice as fast as ordinary valves and balances the flow of steam at the first "breath" of boiler pressure. Adjustable vari-vent feature gets air out of those "far away" radiators as quickly as those close to the boiler.

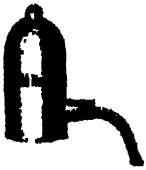
Dole No. 3 Air Valve

Vents radiators of hand fired gravity steam heating systems. Double shell construction provides separate passages for air and condensation extra large float defeats spitting or water leakage. Complete venting assured at pressures up to 10 lbs.



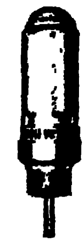
Dole No. 2B Vari-Vent Vacuum Valve

Adjustable radiator valve for "vacuumizing" and balancing gravity steam heating systems. Patented Dole bellows vacuum seal locks out air after it has been once expelled from the system. Easily adjusted vari vent feature assists in equalizing steam flow to all radiators.



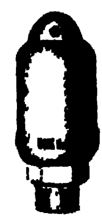
Dole No. 1943 Air Valve

Low cost valve for venting radiators of hand fired systems. Large float provides a seal against condensation to stop spitting.



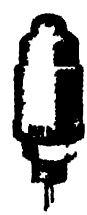
Dole No. 1B Vari-Vent Air Valve

Balances the flow of steam to convectors, either cast iron or copper, of automatically fired systems.



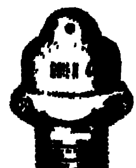
Dole No. 1C Quick Vent Float Valve

Vents mains and speeds flow of steam to radiators of automatically fired systems. Extra large venting port.



Dole No. 5 Quick Vent Float Valve

Vents steam mains on hand fired systems. Positive seal against water.



Dole No. 4 Quick Vent Valve

For quick venting mains that end 18 in. or more above the boiler water line.

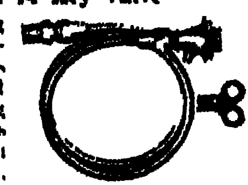
Dole No. 103 Vacuum Valve

For venting convectors, ceiling radiators and pipe coils of "vacuumized" gravity steam systems.

Dole No. 6B Vacuum Valve

Vents the mains of "vacuumized" one pipe steam systems. Prevents the return of air. Closes against water.

Dole No. 14 Key Valve



Low cost venting device for concealed radiators and convectors of hot water heating systems. Protects panel fronts from rusty water stain.

Write The Dole Valve Company for complete catalog and handy selector chart which indicates the Dole Air or Vacuum Valve most suited for a particular need.

Jenkins Bros.

BRONZE - IRON - STEEL VALVES

Mechanical Rubber Goods

80 White St., New York, N. Y.; 524 Atlantic St., Boston, Mass; 376 Spring St., N. W., Atlanta, Ga; 133 N Seventh St., Philadelphia, Pa.; 1514 Fulton St., Chicago, Ill.; 1112 Walnut St., Houston, Tex.

Bridgeport, Conn. (Office and Factory)

Jenkins Bros., Ltd.: Montreal; Factory, Lachine, Canada · London, Eng.

IN VALVES Jenkins GIVES YOU EVERYTHING

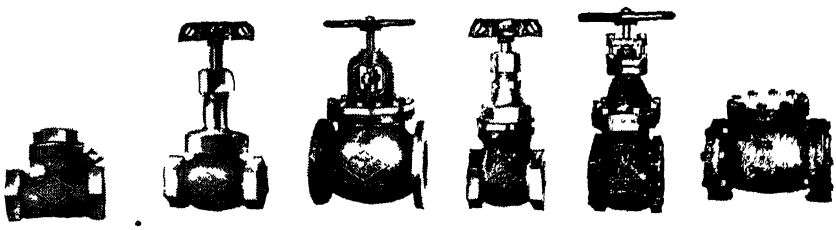


Fig 762 Bronze Regrinding Swing Check — *Fig. 106A* Bronze Globe, Renewable Comp Disc — *Fig. 613* Iron Body Regrinding Globe — *Fig. 370* Bronze Gate — *Fig. 825* Iron Body Gate — *Fig. 624* Iron Body Regrinding Swing Check

OVER 500 DIFFERENT JENKINS VALVES
COVER EVERY HEATING AND AIR CONDITIONING NEED

To adequately describe the complete Jenkins line of valves requires a Catalog of more than 400 pages. There are over 500 different types and patterns of valves that bear the trusted "Diamond" trade mark. Practically speaking, Jenkins can furnish any valve that you may require for plumbing, heating, air conditioning, general industrial or engineering service.

General Classifications of Jenkins Valves Include—Bronze Valves fitted with Jenkins renewable composition disc. Bronze Regrind-Renew Valves with bevel and plug type seats. Bronze Gate Valves. Iron Body Valves fitted with Jenkins renewable composition disc. Iron Body Regrinding Valves. Iron Body Gate Valves with solid wedge and double disc parallel seats. All-Iron Valves. Cast Steel Gate, Globe and Swing Check Valves. Electrically and Hydraulically Operated Valves. Radiator Valves. Fire Line Valves. Quick-opening and Self-closing Valves, Needle Valves, Y Valves, Solder-End Valves, Stainless Steel Valves.

Other Jenkins Products Are—Colored Valve Wheels with or without service markings molded in relief letters. Composition Valve Discs exactly suited to service conditions. Sheet Packing. Gaskets. Moncrieff Scotch Gage Glasses.

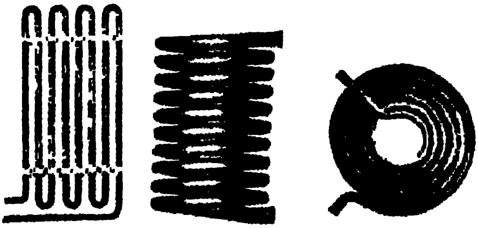
JENKINS VALVES ARE SOLD BY GOOD SUPPLY HOUSES EVERYWHERE

Arthur Harris & Co.

210-218 N. Aberdeen Street **Chicago, Ill.**

ENGINEERS · FABRICATORS OF NON-FERROUS METALS AND STAINLESS STEEL

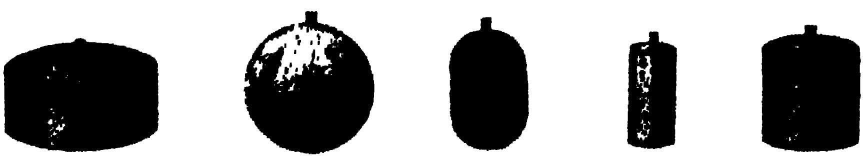
Metals Fabricated Aluminum, Block Tin, Brass, Bronze, Copper, Everdur, Monel, Nickel, Inconel, Stainless Steel and KA2 SMO. Bulletin on request.



Coils

For heating, cooling and condensing. All shapes made from any size pipe or tube—standard or special connections, of copper, brass, aluminum, stainless steel, KA2 SMO, monel, inconel, nickel, block tin, and Everdur.

Metal Floats



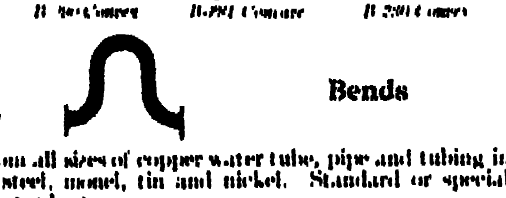
Flat Cylindrical *Ball* *Column* *Cylindrical* *Cylindrical*

Made of copper, plain steel, stainless steel, KA2 SMO, aluminum, brass, Monel, pure nickel, Admiralty and Everdur, for open tank and all pressures.

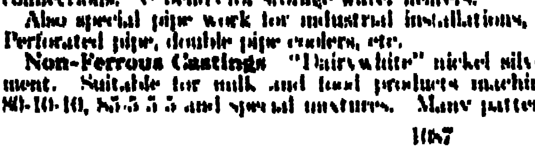
Seamless copper ball floats carried in stock in diameters of 3 in., 4 in., 5 in., 6 in., 7 in., 8 in., 10 in., 12 in. for open tank and pressures of 25, 50, 100 and 150 lb. Floats in special sizes and pressures made to order. Stainless steel ball floats 2½ in. to 12 in. for high pressure and corrosion carried in stock special stainless steel floats made to order stainless steel ball floats larger than 12 in. diameter can be made up specially. Float catalog sent on request

Copper Expansion Joints

For low pressure and vacuum. Made in two styles convex and concave. Sizes 4 in. to 60 in. diameter. Cast iron or steel flanges. Flanges drilled to American standard unless otherwise ordered: B 280 available only in sizes 4 in. to 15 in. inclusive.



B 280 Convex *B-281 Concave* *B 280 Convex*



Bends

We make bends in every shape from all sizes of copper water tube, pipe and tubing in copper, brass, aluminum, stainless steel, monel, tin and nickel. Standard or special connections. U bends for storage water heaters.

Also special pipe work for industrial installations, plumbing, heating and brewing. Perforated pipe, double pipe coolers, etc.

Non-Ferrous Castings "Dairywhite" nickel silver for Process Industries Equipment. Suitable for milk and food products machinery. Castings also of 88-10-2, 80-10-10, 85-5-5-5 and special mixtures. Many patterns available without charge.

The American Brass Company

General Offices: Waterbury, Conn.

Manufacturing Plants:

ANSONIA, CONN. TORRINGTON, CONN. WATERBURY, CONN. BUFFALO N. Y
DETROIT, MICH. KENOSHA, WIS.

Offices and Agencies in Principal Cities

ANACONDA
from mine to consumer

CANADIAN PLANT: ANACONDA AMERICAN BRASS LIMITED, New Toronto, Ontario

PRODUCTS—Anaconda Deoxidized Copper Tubes and Fittings; Anaconda "85" Red-Brass Pipe; Everdur Metal for storage heaters, storage tanks, ducts and air conditioning equipment

ANACONDA COPPER TUBES AND FITTINGS

For Heating, Plumbing and Air Conditioning

Anaconda Deoxidized Copper Water Tubes assembled with Anaconda Fittings offer an unusual combination of advantages in hot water heating systems at a cost only slightly higher than black iron and approximately the same as wrought iron pipe. These advantages may briefly be summarized as follows:

Low Friction Loss—Because the inside surfaces of copper tubes are inherently smoother than those of pipe and tubes made of ferrous materials and also because they do not become roughened by the formation of rust, these tubes offer a minimum resistance to flow In addition, the long radius turns of Anaconda Elbows and the smooth inside surface of Anaconda Wrought Copper Fittings further reduce friction losses.

These factors naturally increase the efficiency of the system, particularly when it includes a forced pressure circulator.

Ease of Installation—In many places the flexibility of copper tubes simplifies connections that ordinarily would be awkward and expensive to make with rigid pipe and threaded fittings. Anaconda Solder Fittings are compact. They can be installed in constricted space where the use of a wrench would be impossible.

Architects and builders naturally object to large holes and notches cut in the framing members of a building for the passage of piping. Anaconda Copper Tubes can be installed with a minimum of cutting in the structure, although holes should be large enough to permit movement of tubes due to expansion and contraction.

Appearance Anaconda Deoxidized Copper Water Tubes assembled with Anaconda Solder Fittings present an attractive appearance. It is a frequent practice to clean the tubes after they are installed and apply a coat of clear lacquer or similar substance. This keeps the tubes bright and makes an installation of which both plumber and owner can be proud.

Temper and Gauges - Anaconda Copper Tubes are made in both hard and soft temper and in standard wall thicknesses.

They meet the requirements for these types of tubes in U. S. Government Specification WW-T-799 and A.S.T.M. Specification B-88-41. Type K, the heaviest, is recommended for heating lines and general piping.

Accuracy of Dimensions Anaconda Deoxidized Copper Water Tubes are all finished to the close tolerances required by the A.S.T.M. and Federal Specifications, which have been found essential for efficient assembly with solder fittings.

Permanent Identification For permanent identification, the name "Anaconda" and the letter designating the type of tube is stamped in the metal at intervals of approximately 18 in., throughout every coil or straight length of tube.

The American Brass Company

Availability Anaconda Copper Tubes, in all standard sizes, are carried in stock by distributors of Anaconda Pipe, located in the principal trading areas of the country. These tubes, in sizes up to and including 1¼ in. are furnished soft in 30, 45 and 60-ft coils; also hard and soft in 20-ft straight lengths. Sizes over 1¼ in. are furnished, hard or soft, in straight lengths only.

ANACONDA "85" RED BRASS PIPE

Anaconda "85" Red Brass Pipe, in standard pipe sizes, is offered as the highest quality corrosion-resistant pipe commercially obtainable at a moderate price and is recommended for steam return lines.

Anaconda "85" Red Brass Pipe contains 85 per cent copper and conforms to government specifications for Grade "A" water pipe. The words "Anaconda 85" are stamped in the metal at one-foot intervals throughout each length.

EVERDUR*

For many years the efforts of metallurgists have been directed to finding some element or elements which, when added to copper would alloy with it to produce a metal with strength approaching that of steel and, at the same time, retain or augment the non-rusting and corrosion-resistant properties of copper.

The addition of silicon and manganese to copper, when their proportions are properly adjusted, produces copper-rich alloys of the solid solution type which attain the desired objective to a remarkable degree. Copper-silicon alloys, made and sold by The American Brass Company under its trademark "Everdur," were the first commercial applications of copper containing substantial proportions of silicon, and mark a decided advance in the metallurgy of copper alloys.

In addition to their non rusting properties and high strength, Everdur alloys possess many qualities not usually found in metals of this character. They are unusually resistant to general atmospheric conditions and other normally corrosive factors. Everdur alloys have excellent machining and working characteristics and can be fabricated into a variety of forms and shapes. They also weld readily by any of the commercial methods.

*"Everdur" is a trademark of The American Brass Company registered at the U. S. Patent Office.

CORROSION RESISTANCE

The corrosion resistance of Everdur is equal to that of pure copper and in some cases, slightly superior.

However, like copper and all copper alloys, Everdur is not equally resistant to all corroding agents, nor to the same corroding agents under all conditions. As with copper, the resistance to corrosion may be substantially reduced in some instances by the presence of oxidizing agents. Nevertheless, Everdur does offer excellent resistance to the corrosive action of many solutions and atmospheres.

Everdur Tanks - Everdur copper-silicon alloy is an ideal material for durable, rustless water tanks of every description -- from domestic range boilers to large storage heaters for hotels, laundries, hospitals, textile plants, schools or breweries.

Everdur is made in all commercial shapes including tank plates which have physical properties as given in A.S.T.M. Tentative Specification B96-41T.

Minimum specification requirements for hot rolled and annealed tank plates are: Tensile Strength, 50,000 psi.; Yield Strength (at 0.5 per cent elongation under load) 18,000 psi.; Elongation, 40 per cent in 2 inches.

Sound, double welded butt joints made on annealed Everdur tank plates have a minimum tensile strength of 47,500 psi. and single welded butt joints have a minimum tensile strength of 42,500 psi. after the beads have been removed.

For additional data and names of fabricators address our nearest office or agency.

EVERDUR FOR AIR CONDITIONING EQUIPMENT

Because of its strength and welding properties, Everdur may be substituted for steel and fabricated by substantially the same methods and with the same equipment as steel.

Everdur metal has been used with marked success for fans and blowers, ducts, humidifiers, cast and wrought parts of other equipment items subject to corrosive influences.

EVERDUR LITERATURE

Descriptive literature containing much pertinent tabular data will be sent upon request.

Wolverine Tube Company

1411 Central Avenue, **Detroit, Michigan**

SEAMLESS TUBE

COPPER - BRASS - ALUMINUM

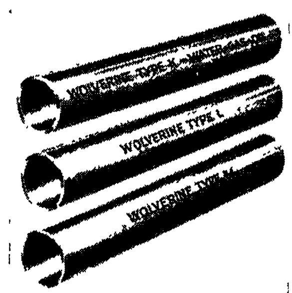
Sales Offices:

Atlanta, Ga.	542 Spring St.
*Baltimore, Md.	121 S. Gay St.
*Boston (Cambridge), Mass.	195 Albany St.
Buffalo, N. Y.	416 Jackson Bldg.
*Chicago, Ill.	1632 W. Lake St.
*Chicago, Ill.	3348 S. Pulaski Road
Cleveland, Ohio	1740 East 12th St
*Dallas, Texas	2813 Canton St.
Dayton, Ohio	Route No. 9
Denver, Colo.	3345 Steele St.
*Long Island City, N. Y.	47-31 31st Place
Los Angeles, Calif.	1015 East 16th St.
Louisville, Ky.	510 W. Main St
Milwaukee, Wis.	647 W. Virginia St.
*Minneapolis, Minn.	100 N Second St.
*Newark, N. J.	34 Providence St.
New York, N. Y.	420 Lexington Ave
*Philadelphia, Pa.	351 North 17th St.
*Pittsburgh, Pa	1000 California, N.S.
Portland, Ore	524 N.W. 11th Ave.
Richmond, Va	Mutual Bldg.
St. Louis, Mo	4565 McRee Ave.
San Francisco, Calif.	7 Front St.
Seattle, Wash.	1005 E. Pike St.
Washington, D. C.	1108 16th St.
Winnipeg, Man.	80 Lombard St.

EXPORT: Rocke International Electric Corp., 100 Varick St, New York, N. Y.

*Warehouse Stock

COPPER WATER TUBE



TYPE K—Recommended for Air Conditioning, Refrigeration, Oil Burner, and Plumbing and Heating installations.

TYPE L—For Oil Burner, Air Conditioning, Refrigeration and general plumbing uses.

TYPE M—Suitable for Air Conditioning and Refrigeration installations and for interior plumbing and heating purposes.

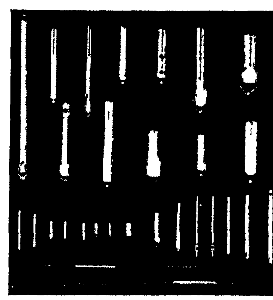
Types K and L furnished in hard or soft temper; Type M, hard only.

Wolverine Water Tube is made according to U. S. Government and A.S.T.M. specifications. For a complete list of these data, write Detroit for Form 575.

REFRIGERATION TUBE

Wolverine refrigeration tube has long been the standard of the industry. Dehydrated, sealed, paper-wrapped; uniform soft temper and moisture content well below minimum specified by A.S.R.E. Available from stock in standard coils.

ACCUMULATOR SHELLS

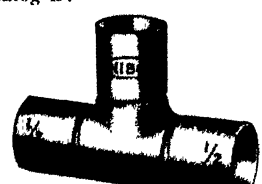


A new accumulator shell developed by Wolverine and produced to customers' specifications in a variety of shapes and sizes up to 3½ in. diameter.

It combines many advantages including one-piece construction and is especially adaptable to refrigeration problems. Send your blueprints or inquiries to Detroit.

WROUGHT FITTINGS

Wolverine-Nibco Solder Fittings are of the straight-line design ends not expanded. They make strong, neat joints; give trouble-free service, and longer life. A complete range of sizes is available. Write to Detroit or your nearest warehouse for Catalog D.



The experience of 25 years of seamless tube manufacture, the use of the latest equipment, and adherence to Government and customer specifications, are responsible for the uniform, high quality of Wolverine products.

INSULATION

Many different materials are used for insulating purposes—in their natural state or processed and fabricated into various forms. They include: Vegetable fibers, wood, tree bark, cork—processed into wools or other fibrous forms, and used in loose bulk or fabricated into boards, paper, blankets or batts. Natural wools, jute, hair—felted into blankets, pads, mats, etc., or used in loose bulk forms. Glass in block, sheet, or wool forms.

Mineral products such as natural rocks and furnace slags—processed into granulated form, or into wool form and used in loose bulk or fabricated into blankets, batts, or pads; and asbestos, asphalt, gypsum and magnesia—used in board form, blankets, felts, or in loose bulk. Many of these types of insulation are also used in plastic form. Metallic insulation, such as aluminum and steel are fabricated into sheet form and used separately or in conjunction with other insulating materials.

INSULATION, Building (p. 1092-1116)

Aluminum sheets, paper in sheets and fabricated forms, felts, cork, glass, glass and rock wools, cane fibre boards, wood products in board form and fibrous blankets and pads, or used in loose fibre form—all are utilized as insulation against heat or cold.

Technical data on this type of insulation will be found in Chapter 4.

Insulating materials, in board or slab form are adapted for use in walls as a plastic base, and thus serve as both a heat or cold insulation and a fire-retarding material.

INSULATION, Sound Deadening (p. 1092-1116)

Many of the insulating materials utilized in building construction are also suitable for sound deadening or acoustical control. Some of them are also adapted for use on machinery and in building to counteract or absorb vibration.

Technical data on Sound Control will be found in Chapter 33.

INSULATION, Underground (p. 1097, 1117-1119)

Asbestos, asphalt, mineral wools, magnesia—used in conjunction with underground piping and conduits of concrete, tile or cast iron.

Technical data is contained in Chapter 43.

INSULATION, Pipes and Surfaces (p. 1092-1119)

Asbestos, magnesia, and mineral wools in loose fibrous forms, blankets, or in plastic forms and suitable for use in extremes of high or low temperature service; also hair and felts, and cork in loose bulk or in molded or plastic forms.

Technical data will be found in Chapter 43.

Some of these insulating materials are also used as refractory materials.

INSULATION, Duct (p. 1092-1114)

Various of the insulating materials which may be fabricated into board or slab forms, and various felts and fibrous materials have been adapted for use as duct insulation—as a duct liner or applied to the outer surfaces. Some have been utilized to construct the walls of the duct itself, serving the dual purpose of duct and insulation.

Technical data is contained in Chapter 43.

INSULATION, Window (p. 1120-1123)

Single-pane and double-pane insulating window sash, metal fabric insulating window screens, weather stripping for windows and for interior and exterior doors.

Manufacturer's products shown in this division are designed for specific applications. Consult the Index to Modern Equipment for additional products of these manufacturers.

Aluminum Aircell Insulation Co.

Curtis Bldg., **Detroit, Mich.**

INSULATION FOR:

Air Conditioning
Furnaces
Stoves
Ovens
Bus Motor Insulation
Any High Temperature Insulation

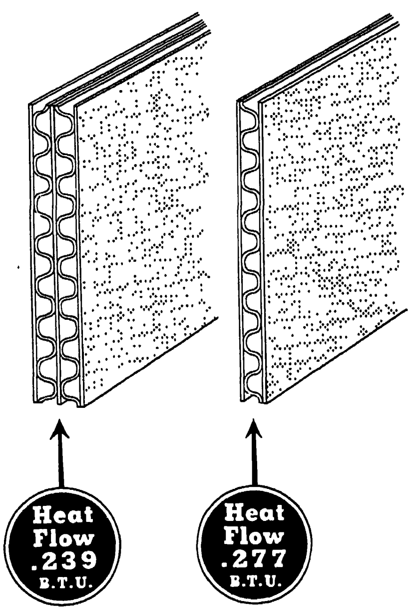
Licensed under Patents 1,757,479—1,890,418—1,934,174 others pending

INSULATION FOR:

Buses
Hot Water Heaters
Trailers and Trucks
Homes
Incubators
Cold Storage Rooms
Automobiles

Asbestos REFLECT-O-CELL

Asbestos Reflect-O-Cell is ideal wherever a high temperature insulation is required. Reflects 95 per cent of radiant heat. Provides a moisture barrier as well as a heat barrier. Light in weight. Economical to use. Easily cut to size. Easy to apply.



Furnished single-ply in 250 sq ft rolls, 36 in. wide.

Furnished 2-ply in 3 ft x 6 ft panels. Also Cut to Your Specifications.

Standard REFLECT-O-CELL

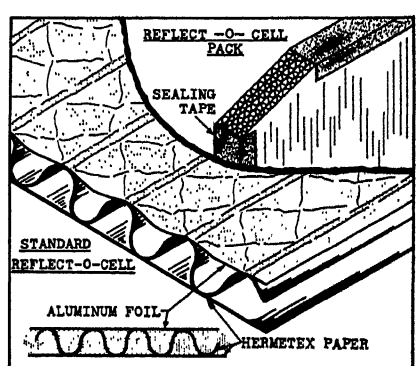
REFLECT-O-CELL is the modern insulating material based on the famous Dewar Principle of insulation, the outstanding example of which is the efficient Thermos Bottle.

The Structure of Reflect-O-Cell permits both surfaces of the foil to reflect radiant heat.

Moisture Sealing, which effectively reduces summer dehumidifying load and winter humidifying requirements by preventing vapor pressure losses.

Especially Indicated for exposed application to eliminate air-conditioning shock and to reduce cooling load.

K. FACTOR—.274



Furnished: 400 sq ft per roll 17 in. wide; 500 sq ft per roll—24, 36, 48 in. wide.

Made with composition foil, testing fully equal to aluminum.

The Philip Carey Company

Manufacturers of Heat Insulation and Asbestos Products

Lockland — **Cincinnati, Ohio**

Sales Offices

Atlanta, Ga
Baltimore, Md.
Boston, Mass.
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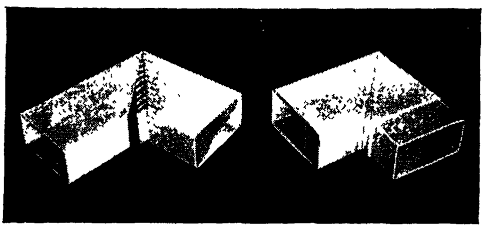
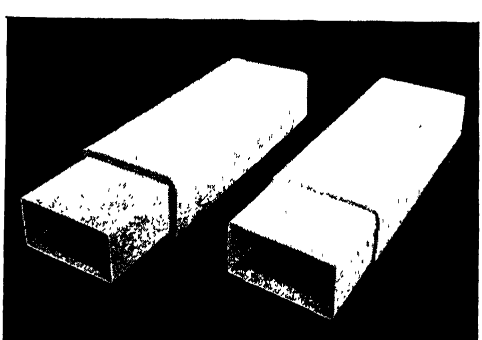


Standard 90 deg Elbow assembly. Left: Core opened to show duct vanes. Right: The completed fitting

Two standard 90 deg Elbows nested in a larger standard section to form a tee.



Standard 1 in. and ½ in. thick Careyduct sections with core extended.

Careyduct is a new prefabricated insulated duct built entirely of asbestos. The double layer construction consists of an inner core of hard, rigid asbestos, and the outer jacket is made of multiple layers of a fine corrugated asbestos structure. The combination results in great strength, is an excellent insulator, and has a definite sound deadening effect.

Careyduct fittings are made from standard sections of duct, and may be made in the field with comparative ease by men without special training. A simple mitre cut plus a few standard accessories make a complete fitting thus keeping costs at a minimum. Prefabricated fittings may be ordered from the factory if desired.

The telescopic assembly method practically eliminates leaks that are commonly found in other construction.

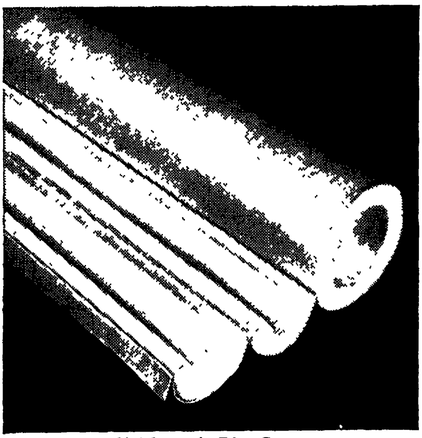
The standard sizes of Careyduct are designed so that a combination of smaller sizes will exactly nest in a larger size. All tees and take-offs are a combination of ells and straight duct.

Grilles and dampers are installed according to the accepted standard practice. Careyduct gives high insulating value. It materially reduces the transmission of extraneous and equipment noises. Careyduct costs decidedly less than properly insulated metal duct and compares very favorably with sheet metal duct of standard quality.

For more detailed information write for Catalog and Erection Manual.

Ehret Magnesia Manufacturing Co.

Valley Forge, Pa.



85% Magnesia Pipe Coverings

A FULL RANGE OF INSULATIONS FOR HEATING AND VENTILATING

The Ehret Company furnishes a broad range of thermal insulations for practically every industrial and architectural requirement. For full details of Ehret products, see the Ehret Insulation Manual.

Ehret's 85 Per Cent Magnesia

Known for nearly half a century in the industrial field, Ehret's 85 per cent Magnesia Pipe Coverings and Blocks are efficient, economical and they last indefinitely. Pipe coverings are available in a full range of sizes and thicknesses, and blocks can be furnished in thicknesses up to 4 in. An ideal material for use on heated pipes or surfaces whose temperatures do not exceed 600 F.

OTHER HEAT INSULATIONS

In addition to 85% Magnesia insulation, the Ehret Company furnishes a full line of other heat insulating materials, in the forms of pipe coverings, flat and curved blocks, sheets, lagging, blankets, cements and loose fills. These materials include Enduro (high temperature), asbestos cellular, asbestos sponge felt, mineral wool and many other products for use on heated pipes and surfaces.

COLD INSULATIONS

Ehret insulations for use on cold pipes and surfaces are made in a variety of forms and materials Pipe coverings include cork, wool felt. frostproof and anti-sweat. Standard Hair Felt, Punched Hair Felt and Insulfelt, in roll form are used to insulate both curved and flat surfaces. Ehret's Eroduct is a special material in ½ in. thickness that is applied to air conditioning and cold air ducts. Cork blocks, sheets and discs, as well as granulated cork are also furnished.

BUILDING INSULATIONS

For insulating the walls, floors and ceilings of buildings, Ehret's Heat Seal Wool is made in batts, strips, loose and granular forms. This material is high in insulating efficiency, is easy to install or apply and it will last indefinitely. Batts can be furnished with or without paper backing, as desired.

OTHER EHRET PRODUCTS

In addition to the insulations themselves, the Ehret Company can furnish insulation accessories such as water-proofing compounds, weathertight jackets, bands, wires, adhesives, sewing canvas, asbestos paper, wallboard and many other materials required for the application of insulations. Ehret packings and asbestos products as well as Durant Insulated Pipe (which is briefly described on the opposite page) are fully treated in the Ehret Insulation Manual. Write today for your copy of this 280 page handbook.

Ehret's Heat-Seal Wool Building Insulations

Two 3 in. steam lines of Durant Insulated Pipe, being installed one above the other in a narrow trench.

EHRET'S DURANT INSULATED PIPE

. . . for Underground and Outdoor Service

This unique system of pipe line protection consists of pipe that is insulated, sealed and protected *at our factory*, and shipped to the job ready for installation. Pipe lengths can be joined with screwed, flanged or welded fittings, and the system provides protection for expansion bends, joints, valves and similar pipeline appurtenances.

Field joints in Durant Insulated Pipe are easy to make, and once made the backfill can be begun and the trench flooded for tamping.

Ehret's Durant Insulated Pipe will not crack or leak and moisture or water is permanently excluded by the thick, time-defying layer of high-melting-point asphalt that encloses all parts of the system. Write for the special Ehret D.I.P. folder it gives full details.

Some Outstanding Advantages of Durant Insulated Pipe:

1. Permanently waterproof.
2. Elimination of electrolysis and corrosion.
3. Requires no sub-drains as even complete water submersion does no harm.
4. In multiple lines, individual Durant pipes can be added, removed or replaced without disturbing others.
5. Minimum trenching and field work.
6. No rollers or pipe supports required.
7. No breakage or waste of material during installation.
8. Tile or masonry protection not required.
9. Field costs are much lower than those of tile, tunnel and similar systems.
10. Insulation protection is absolutely dependable.

The Eagle-Picher Lead Company

General Offices: Temple Bar Building, **Cincinnati, Ohio**

Offices in Principal Cities

EAGLE
PICHER

A Remarkable Insulating Wool Made From Minerals

Years ago Eagle-Picher pioneered a method of fusing and fiberizing carefully selected minerals into a dark gray insulating wool. This mineral wool is chemically inert. Fibers are mechanically strong, extremely resilient and flexible. They withstand expansion and contraction without loss of efficiency even at elevated temperatures.

From this mineral wool, Eagle-Picher has fabricated a long list of insulating products to meet a wide range of temperatures and operating requirements.

Eagle H-2 Loose Wool

A clean fill insulation that is highly efficient for temperatures to 1200 F. Averages considerably lighter in weight than many rock and slag wools—goes farther. Fibers are soft and flexible. Approved by Underwriters Laboratories as fireproof and a non-conductor of electricity. Retains physical and chemical stability in presence of water. Packed in 40-lb. bags.

Eagle 7-B Granulated Wool

Another grade of fill insulation that has all the advantageous properties of Eagle H-2 Loose Wool. It consists of small pellets averaging $\frac{1}{8}$ to $\frac{1}{2}$ in. in size. For all fill jobs in irregular spaces. May be poured. Packed in 40-lb. bags.

Eagle Low Temperature Felt

A highly efficient insulating material for subzero and low temperatures (to 400 F). Inherently water-repellent (not specially treated). Available in densities 6-lb to 8-lb per cu ft. Recommended for refrigerator rooms, trucks, refrigerators, stoves, etc. Extensively used in marine field.

Eagle H-5 Felted Pads

Specially designed for use in pre-fabricated equipment where high insulating efficiency is required. Offers low conductivity in easy to apply form. Pads come rolled in paper, 6 to 12 ft in length. Thicknesses—$1\frac{1}{2}$ in., 2 in. and $2\frac{1}{2}$ in. Standard width is 24 in.

Paper Encased Batts and Blankets

These light-weight, sturdily constructed batts and blankets are easy to apply. Enclosed on four sides with paper, one side of which is an approved vapor barrier. Strong tacking flanges. Quickly cut with knife or shears. Three thicknesses—Ful-Thik, Semi-Thik and 1-in.

Eagle Super "66" Cement

A high temperature plastic insulation. Easy to apply and trowels to a smooth finish. Does not cause corrosion. Will stick on any clean, heated surface. Dry coverage 50-55 sq ft per 100 lbs. 100 per cent reclaimable up to 1200 F. Packed in 50-lb bags.

Eagle Supertemp Blocks

An all-purpose block insulation which will withstand elevated temperatures up to 1700 F without loss of efficiency or structural strength. Blocks are water-repellent. Light weight. Easily cut to fit irregularly shaped surfaces. Blocks resist attacks of steam and moisture, and withstand all normal vibration and abrasion. Available in all standard sizes.

Eagle Insulseal

A protective coating for Blankets, Supertemp, "66" Cement and other kinds of heat insulation. Provides a permanent seal that safeguards insulation against air infiltration, moisture, water, fumes; also against vibration and abrasion. Does not support combustion.

For more complete specifications and technical data on these and other Eagle Insulating Products, see Sweet's Engineering or Power Plant catalogs.

10807 Lyndon
at Meyers Road

IN SULATION DUSTRIES CORPORATED DETROIT

Detroit
Michigan

ROCK WOOL INSULATION PRODUCTS

BUILDING INSULATION PRODUCTS

Loose Rock Wool (paper bags)
Granulated Rock Wool (paper bags)
Rock Wool in Rolls (any length or thickness)
Rock Wool Batts (cartons) (with or without paper backs)
Rock Wool Batts (bags) (without paper backs)

Insulation Industries Incorporated owns and operates one of the most modern, up-to-date Rock Wool plants.

Rock Wool is manufactured by a patented, precision process that produces a superior grade of Rock Wool. It is light in weight, has long, silky and resilient fibers. It is clean and free from foreign particles.

Rock Wool is indestructible and will last as long as the building itself. It is fire-proof, vermin and rodent-proof and is resistent to moisture.

BUILDING INSULATION

Rock Wool is suitable for all types of building insulation requirements.

It can be applied in the granulated form by the pneumatic method to existing homes or buildings.

For new construction or for unfinished attic or wall spaces, Batts are furnished either 15 x 23 in. or 15 x 48 in. and 2 or 4 in. thick and with or without paper backs, packed in cartons.

Long fiber Rock Wool in loose form is available packed in 35-lb paper bags.

RESULTS

Results obtained in all types of buildings, both old and new, show substantial savings in fuel consumption with elimination of drafts and variation of temperatures between rooms and floors.

INDUSTRIAL INSULATION PRODUCTS
For

Stoves and Ranges
Water Heaters
Industrial Ovens
Bakery Ovens
Large Diameter Pipes
Boiler Settings, etc.

BLANKETS

Long fibered, especially treated Rock Wool, felted and secured between metal fabrics of different types. These blankets are made in standard sizes 24 in. x 96 in. and 24 in. x 48 in. and special sizes as required and any thickness from 1 in. to 8 in. Applicable to flat or curved surfaces.

INSULATING BLOCK

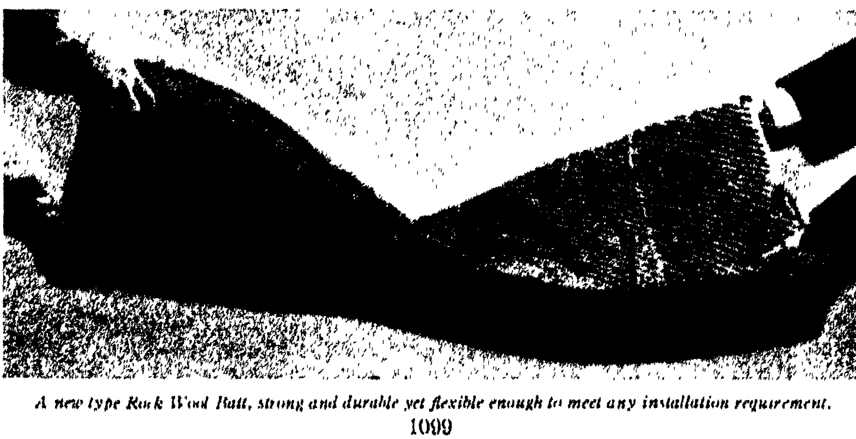
Rock Wool fabricated into sheet or board form from 1/2 in. to 4 in. thick, 24 in. x 36 in. or special sizes, as required. This block is widely used for insulating boilers, ducts, tanks, stills, etc., and for domestic furnaces, boilers, ranges and hot-water tanks.

INSULATING CEMENT

For finishing block and blanket insulation. For temperature conditions from 100 to 2000 deg. Is very plastic and is quickly and easily applied.

SPECIFICATIONS

Write for complete information and details on Insulation Industries products.



A new type Rock Wool Batt, strong and durable yet flexible enough to meet any installation requirement.

INSULITE

Division of

MINNESOTA AND ONTARIO PAPER COMPANY

General Offices

1100 Builders Exchange, **Minneapolis, Minnesota**

TWENTY-EIGHT YEARS PROVEN DURABILITY

For 28 years engineers and architects have specified Insulite materials for structural uses, interior finish, duct lining, and for other thermal insulation and sound control work. Insulite materials have proved themselves practical through their performance on the job.

STRUCTURAL MATERIALS

Lok-Joint Lath—An insulating plaster base, fabricated from Ins-Lite or from Graylite. Patented "Lok" firmly locks the sheets between supporting members. Thickness: ½ in. Size: 18 x 48 in.

Sealed Graylite Lok-Joint Lath—An insulating plaster base of Graylite, sealed on stud space side with an effective vapor barrier. Has patented "Lok" on long edges. Furnished in same thickness and size as Ins-Lite and Graylite Lok-Joint Lath.

Bildrite Sheathing is an asphalt-containing wood fiber board manufactured under an exclusive process which provides increased strength and moisture resistance. It is 25/32 in. thick and has a distinctive gray-brown color. Thermal conductivity: 0.36 Btu per inch thickness. Each sheet is marked to indicate proper nail spacing. Available in sizes 4 x 8 ft up to 4 x 12 ft with all edges square. Also available in 2 x 8 ft size with interlocking joint on long edges. Used as a structural sheathing board and as a roof boarding.

Condensation Control—Where low outside temperatures and high inside humidities may occur, authorities recommend "sealing the hot side and venting the cold side" of the wall to prevent condensation. An adequate vapor barrier, Sealed Graylite Lok-Joint Lath, should be used on the warm (room) side of the wall thereby effectively reducing vapor transmission into the stud space. Bildrite Sheathing is designed to allow any surplus vapor in the stud space to "breathe" or be vented to the exterior air. If vapor is trapped within the stud space and cannot escape through the sheathing, destructive condensation may occur.

INSULITE WALL OF PROTECTION

This construction consists of Bildrite Sheathing on the exterior of the frame work and either Lok-Joint Lath or Insulite Interior Finish Materials on the interior. Transmission coefficients (U) are shown below.

Exterior Finish and Sheathing	Interior Finish	
	No Insulation Between Studding	
	No plaster—Insulite Building Board, Interior Board, Tile-Board, or Plank (½ in.)	Plaster (½ in.) on Lok-Joint Lath (½ in.)
Wood Siding, 25/32 in. Bildrite Sheathing	0.16	0.15

The above values are typical of results which can be obtained by utilizing Insulite materials in frame construction. For further (U) values refer to Chapter 4, pages 102 and 103.

Applying Bildrite Sheathing

Applying Lok-Joint Lath

INTERIOR FINISH MATERIALS

Ins-Lite Building Board A wood fiber board with the light color of natural wood- burlap and linen textured surfaces. Thermal conductivity: 0.33 Btu/hr/sq ft/in./F; density: 16 lb/cu ft. Furnished in thicknesses of ½ and ¾ inch and sizes of 4 x 7 ft to 4 x 12 ft. Also available in 6 x 8 ft, 6 x 12 ft and 8 x 12 ft sizes.

Graylite Building Board An integrally treated asphalt containing wood fiber board of graysih brown color burlap and linen textured surfaces. Thermal conductivity 0.35 Btu per inch thickness. Furnished in same thicknesses and sizes as Ins-Lite Building Board.

Smoothcote Interior Board Coated Insulating Board with smooth, hard surface one side, having 68 per cent light reflection. Furnished in ½ inch thickness only and in sizes of 4 x 7 ft to 4 x 12 ft.

Satincote Interior Board Factory finished Insulating Board in colors buff, gray, coral and green. Light reflection from 64 per cent for green to 80 per cent for the buff color. Requires no further decoration. Highly resistant to abrasion and easily washable. In ½ inch thickness and in sizes of 4 x 7 ft to 4 x 12 ft.

TileBoard Available in Smoothcote and Satincote. TileBoard is furnished with the Lok-Grip Joint that permits concealed nailing and which together with the Lok-Pin (a flat diamond shaped metal dowel) definitely and mechanically safeguards against any falling units even though no face nailing is used.

Smoothcote and Satincote TileBoard available in ½ inch thickness and sizes of 12 x 12 inches to 16 x 32 inches.

Plank Available in Smoothcote and Satincote. Plank has the Lok-Grip joint which permits concealed nailing and is beveled and beaded both long edges. Smoothcote and Satincote Plank furnished in ½ inch thickness, widths of 8 to 16 inches and lengths of 8 to 12 ft.

Acoustilite or Fiberlite effectively quiet and control sound

Acoustilite A high efficiency acoustical material for sound control. Coefficient of sound absorption, at 512 cycles, is 0.79 when mounted on solid background and 0.80 when on furring strips. Noise reduction coefficient is 0.65 when mounted on solid background and 0.75 when on furring strips. Factory painted in buff, (light reflection 77 per cent) and in white (light reflection 80 per cent). Units have a butt joint and are beveled on four edges. Thickness, ¾ in.; sizes, 12x12 in. to 16x32 in.

Fiberlite An efficient sound absorptive and decorative material. Coefficient of sound absorption, at 512 cycles, is 0.53 when mounted on a solid background and 0.72 when on furring strips. Noise reduction coefficient is 0.55 when mounted on solid background and 0.65 when on furring strips. Factory painted in buff (light reflection 77 per cent) and in white (light reflection 80 per cent). Units have a butt joint and are beveled on four edges. Thickness, ½ in.; sizes, 12x12 in. to 16x32 in.

HardBoard Products

HardBoard materials are tough, durable, grainless, pressed wood fiber boards with a hard, smooth surface. Available in a range of densities from 55 to 68 lb/cu ft. Thicknesses are from 1/10 to 5/16 in. and sizes of 4 x 2 ft to 4 x 12 ft.

Industrial Insulation

Industrial Insulation is a wood fiber board for use in all types of manufacturing industries producing items such as refrigerators, coolers, showcases, brooders, partitions and cabinets.

It can be cut-to-size and fabricated to customer's specifications. Three types of industrial board are available.

Lowdensite Industrial Board A 10 to 14 lb density board with an average tensile strength of 100 lb sq in. and an average conductivity of 0.30 Btu/hour/sq ft/F inch thickness.

Ins-Lite Industrial Board A 14 to 18 lb density board with an average tensile strength of 250 lb/sq in. and an average conductivity of 0.33 Btu/hour/sq ft/F, inch thickness.

Graylite Industrial Board Differs from two above products in that it has an integral asphalt treatment which provides increased strength and moisture resistance as well as minimum thickness and linear expansion. A 16 to 20 lb density board with an average tensile strength of 350 lb/sq in. and an average conductivity of 0.35 Btu/hour/sq ft/F/inch thickness.

Johns-Manville

Executive Offices: 22 East 40th Street, New York, N. Y.

Offices in All Large Cities

Johns-Manville Home Insulation

Johns-Manville Rock Wool Home Insulation is a light, fluffy mineral wool, highly efficient in heat-proofing practically any building, old or new. It is durable, rot-proof, fire-proof and odorless, and will not corrode or settle. Full stud thickness of this material will cut fuel costs up to 30 per cent in winter and help keep rooms up to 15 deg cooler in hottest weather. J-M Rock Wool Home Insulation is furnished in two forms: for new construction,

Applying J-M Super-Felt Type B batts in new home

in easily handled batts; for existing buildings, in nodulated form to be installed pneumatically.

For New Construction
J-M Super-Felt Type B Batts

Super-Felt Type B Home Insulation is furnished in pre-fabricated batts of uniform thickness and density, in both full stud thickness and semi-thick, in sizes 15 x 23 in. and 15 x 48 in., designed to fill completely the space between studs, joists and rafters on the usual 16 in. centers. The sturdy felted "wool" is strong enough to be handled rapidly without damage. The batts are backed with waterproof, vapor-resistant paper, extending on both the long sides in 1½ in. wide flanges, by which the batt is fastened in place and which also aid in sealing the joints. This backing protects against penetration of moisture from wet plaster and also resists infiltration of moisture vapor from the house into the wall.

As a further protection against moisture, the felted wool is also waterproofed.

Super-Felt may also be obtained in blanket form, in Thick, Medium and 1 in. thicknesses. The blankets have a waterproof vapor barrier paper on one side and a permeable kraft paper on the opposite side, cemented together along the long edges to form a strong nailing flange.

For Existing Homes and Buildings
Type A "Blown" Rock Wool

Type A Rock Wool is blown pneumatically into the spaces between studs in outer walls and between rafters or joists in roofs or attic floors. Insulation thickness in walls corresponds to stud depth, approximately 3⅝ in.; the density, approximately 5 to 8 lb per cu ft, assures maximum thermal efficiency. This type of insulation is installed only by Approved J-M Home Insulation Contractors, who are equipped with the necessary apparatus and trained crews.

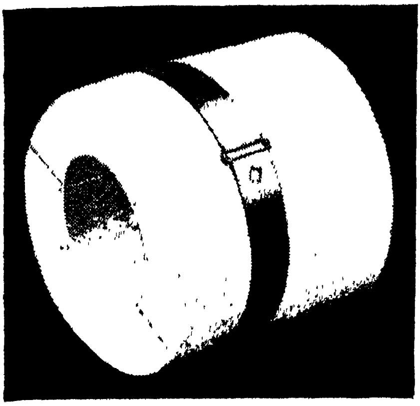
Write for Details

Complete information on all types of J-M Rock Wool Home Insulation will be furnished on request.

J-M Airacoustic Sheets for lining Air-Conditioning Ducts

J-M Airacoustic Sheets, for duct linings of air conditioning systems, are flame-proof, highly sound-absorbent and moisture-resistant, with a surface which will not materially increase friction losses in the duct system. Write for Bulletin AC-23A.

Johns-Manville Pipe and Boiler Insulation



J-M 85% Magnesia Pipe Insulation

J-M Pre-Shrunk Asbestocel Pipe Insulation

J-M Pre-Shrunk Asbestocel is a radically improved insulating material for hot water or low pressure steam piping, which, since it is made of moisture-proofed asbestos paper, minimizes objectionable shrinkage. Supplied in canvas, asbestos paper or aluminum finishes. All types furnished in 3-ft sections in standard thicknesses of 2 to 8 plies, each ply approximately 1/4 in. thick, for all commercial pipe sizes.*

J-M 85% Magnesia

Recommended as the most widely used insulation of the molded type for temperatures up to 600 F. Pipe insulation is furnished in sectional or segmental form for all commercial pipe sizes,* in thicknesses up to 3 in. Blocks are 3 in. by 18 in. and 6 in. by 36 in., flat or curved, from 1 in. to 4 in. thick. Minimum thickness for curved blocks, 1 1/4 in.

J-M Pre-Shrunk Wool Felt Pipe Insulation

Due to its Dual-Service Liner—an asphalt-saturated felt—J-M Pre-Shrunk Wool Felt is equally effective and durable on either hot or cold water service piping. By the use of waterproofed felts, shrinkage troubles have been minimized.

Supplied in two finishes, the regular canvas and a smooth, dull-coated aluminum. In either finish, it is furnished in 3-ft sections in thicknesses of 1/2 in. 3/4 in., 1 in., Double 1/2 in., and Double 3/4 in., for all commercial pipe sizes.*

J-M Asbesto-Sponge Felted Pipe Insulation

Recommended on all high pressure steam piping at temperatures up to 700 F where insulation may be subjected to rough usage or where maximum efficiency and durability are desired. Furnished in 3-ft sections up to 3 in. thick, for all commercial pipe sizes.

J-M Superex Combination

Superex Combination Insulation (an inner layer of high temperature Superex and an outer layer of 85% Magnesia) is recommended where temperatures exceed 600 F. Superex and Magnesia are both furnished in sectional and segmental pipe covering, and in block forms.

J-M Asbestocel Sheets and Blocks

Asbestocel Sheets and Blocks are used for insulating warm-air ducts, flues, heater casings and fan housings in the ventilating system. Temperature limit 300 F. Furnished 6, 9, 12, 18 and 36 in. wide by 36, 48, 72 and 96 in. long, from 1/4 in. to 2 in. thick.

J-M Rock Cork Sheets and Pipe Insulation

J-M Rock Cork is made of mineral wool and a moisture-proof binding ingredient molded into sheets for insulating refrigerated rooms and air conditioning ducts; and into sectional pipe insulation with an integral waterproof jacket, for all low temperature service. It is strong, durable, and will not support vermin. Because of its unusual moisture resistance, its high insulating efficiency is maintained in service.

Furnished in sheets 18 in. by 36 in., in 1, 1 1/2, 2, 3 and 4 in. thicknesses; also 18 in. by 18 in. by 1 in. thick. In lagging form, for curved surfaces, supplied 18 in. long by 1 1/2, 2, 3 and 4 in. thick, 2 to 6 in. wide, depending on diameter. In pipe covering form, in ice water, brine and heavy brine thicknesses, for all commercial pipe sizes.

Details on Request

Write for complete information on any Johns-Manville insulating material.

*Can also be supplied in sections to fit straight runs of copper pipe or tubing with outside diameter 3/8 in. and larger.

KIMBERLY-CLARK CORPORATION

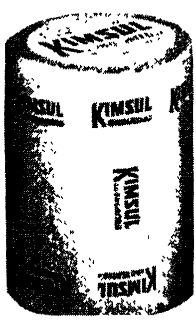
ESTABLISHED 1872 (*Building Insulation Division*) NEENAH, WISCONSIN

KIMSUL* is a trade-mark of Kimberly-Clark Corp. for its brand of laminated and asphalted, compressed insulation.

KIMSUL INSULATION

High in Thermal Efficiency . . . Easy to Install . . . Long-Lasting . . . Moisture Resistant . . . Clean . . . Light in Weight . . . and Low in Cost!

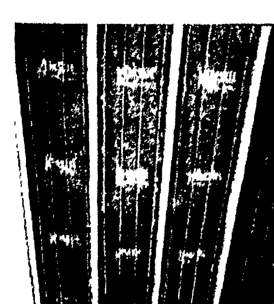
KIMSUL*, is a wood fibre product, made in long, flexible blankets composed of many creped layers or plies, providing a maximum number of dead air cells for efficient insulation. Being flexible and extremely light in weight, it is easy to install. Each blanket is stitched with rows of strong twine running the length of the blanket. This unique feature holds the installed KIMSUL blanket securely in place—prevents sagging or "packing down" inside the walls.



KIMSUL comes compressed, packaged as at left . . . is expanded on job to about 5½ times packaged length (right), saving on handling time, storage space and transportation cost.

Strong stitching keeps KIMSUL at its proper density, prevents it from sagging, sifting and settling. Once in place, KIMSUL stays "put."

KIMSUL Insulation is easy to cut to exact size, fits out-of-the-ordinary spaces as neatly and as easily as it fits standard spacings.



Sloping roofs present difficult insulation problems, but even in spots like this, one man can usually install KIMSUL quickly and easily.

Manufactured by the Kimberly-Clark Corporation, makers of wood fiber products since 1872, KIMSUL* Insulation is one of the most efficient insulating materials ever developed.

Thermal Insulation: For walls, floors, ceilings and roofs. "k" Factor is .27 Btu/hr/sq ft/degrees F/inch—J. C. Peebles.

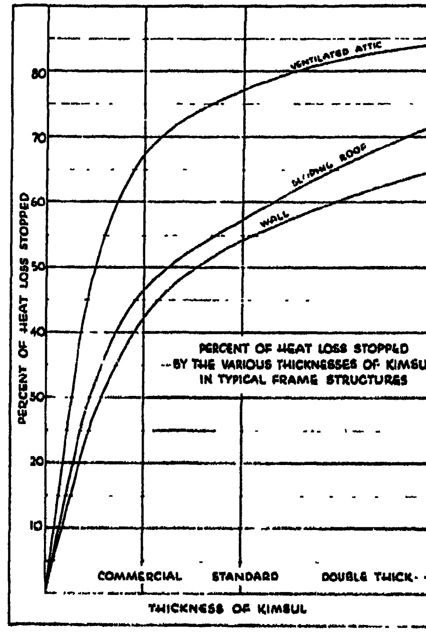
Acoustical Insulation: For walls and ceilings. Average coefficients of absorption: Commercial thickness .41. Standard thickness .59. Double-thick .67.

Sound-Deadening Insulation: For partitions and floors.

Physical Characteristics: KIMSUL Insulation is a laminated and asphalted wood fiber flexible blanket insulation . . . resistant to water . . . extremely light in weight (1.5 lbs per cu ft) . . . faced with a tough waterproof cover.

Available in 3 Thicknesses: Commercial Thick (nominally one-half inch) . . . Standard Thick (nominally one inch) . . . Double Thick (nominally two inches) . . . Furnished in correct widths for standard stud spacings.

Vapor Seal: Separate sealing recommended wherever vapor seal is required.

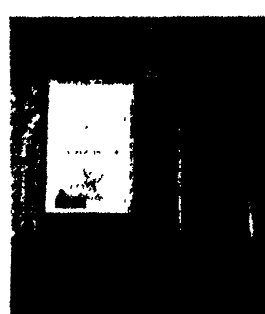


Graph shows how effectively KIMSUL reduces heat flow through typical frame structures. Note that greatest proportion of heat losses are stopped by the first inch of KIMSUL.

*Reg. U. S. & Can. Pat. Off.

Pipes and conduits don't interfere with the insulating efficiency of KIMSUL. It works around corners, tucks snugly into "tight" places.

Used for caulking around window frames and doorways, odd pieces of KIMSUL add to effectiveness of insulation job, eliminate waste.



When a Vapor Seal is needed to guard against condensation within walls, it should be installed as shown, separate from the insulation.

Mundet Cork Corporation

65 S. Eleventh St. INSULATION DIVISION **Brooklyn, N. Y.**

Manufacturers of Corkboard, Cork Pipe Covering, Compressed Machinery Isolation Cork, Natural Cork Isolation Mats, and all kinds and varieties of Cork Specialties.

Authorized contractors for high temperature insulation.

Mundet Branches

Albany, N. Y.	Chicago, Ill.	Houston, Texas	Philadelphia, Pa.
Atlanta, Ga.	Cincinnati, Ohio	Kansas City, Mo.	St. Louis, Mo.
Brooklyn, N. Y.	Dallas, Texas	Los Angeles, Calif	San Francisco, Calif.
Boston (No. Cambridge), Mass.	Detroit, Mich.	New Orleans, La.	Syracuse, N. Y.

Mundet Distributors are Located in the Following Cities—Names and Addresses on Request

Amana, Iowa	Hartford, Conn.	Oklahoma City, Okla.	
Baltimore, Md.	Johnson City, Tenn.	Portland, Oregon	Seattle, Wash
Buffalo, N. Y.	Memphis, Tenn.	Providence, R. I.	Tucson, Ariz.
Charlotte, N. C.	Minneapolis, Minn.	Richmond, Va.	Tulsa, Okla.
Cleveland, Ohio	Nashville, Tenn.	Rochester, N. Y.	Utica, N. Y.
Denver, Colo.	Norfolk, Va.	Salt Lake City, Utah	Youngstown, Ohio

Mundet "Jointite" Corkboard

—for all low temperature insulation and for acoustical correction. 100 per cent pure cork, fabricated in accordance with U. S. Government Master Specifications and unsurpassed in its field. Sold in standard 12 in. x 36 in. sheet. Standard thicknesses, ½ in., 1 in., 1½ in., 2 in., 3 in., 4 in., 6 in.

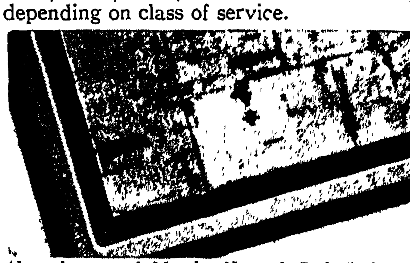
Mundet "Jointite" Cork Pipe Covering

Shown below, with fitting cover. Protects all types of low temperature lines. Made in 3 thicknesses, with complete line of standard covers, suitable for pipes carrying sub-zero to 50 F temperature.

Section of Mundet Moulded Cork Pipe Covering, with Fitting. The pipe covering is made in sections 36 in. long, to fit all sizes of pipes.

Mundet Cork Vibration Isolation

Machinery vibration encountered in heating and ventilating work is effectively controlled by the use of Mundet Natural Cork Isolation Mats. These consist of blocks of pure cork, held together within a rigid steel frame or bound with asphalt paper applied with hot asphalt top and bottom. Mundet steel bound mats are usually used under exposed mounts; asphalt paper bound mats under concrete foundations of the envelope type. Mats are made to fit under any type of machine foundation. For loads exceeding 2000 lb per square foot, we manufacture Mundet Machinery Isolation Cork, which is a board form of compressed granulated cork, available in 3 densities. All types of isolation are furnished in 1 in., 1½ in., 2 in., 3 in., 4 in., and 6 in. thicknesses, depending on class of service.



Above close-up of Mundet Natural Cork Isolation Mat shows how the blocks of cork are held together within a steel frame.

Engineering and Specification Service

Our engineering department is at the service of Architects and Engineers, to assist and advise in the preparation of specifications pertaining to cork. This service is also available without obligation to any one who has a low temperature insulation or a vibration isolation problem. Our complete catalogue is filed in Sweet's Catalogue File for Engineers and Architects and will be sent on request. It is replete with information and data of value to every specification writer whose field touches our products.

Mundet Contract Service

Covers the complete installation of our products, in accordance with best established practice. Divided responsibility is avoided. All materials and workmanship are guaranteed.

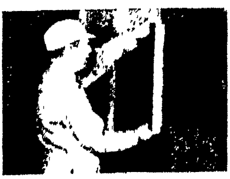
National Gypsum Company, Buffalo, N. Y.

Manufacturers of

ROCK Gimco WOOL

Rock Wool Plants located at

DOVER, N. J — ALEXANDRIA, IND. — DUBUQUE, IOWA



The Gimco Sealal Batt is probably the most frequently used insulating material on the market today. It is easy-to-handle, quick-to-apply, and unexcelled as a heat barricade. The Sealal Batt fits snugly between standard studdings and joists. Its semi-rigid quality eliminates need of extra support and insures against sagging.

Gimco Sealal Rock Wool Bats— Gimco Rock Wool Bats are made from long, tough rock wool fibres annealed and treated specially by the patented Gimco process. Installed 3⅝ in. thick Gimco's conductivity is only 0.067 Btu per hour per sq ft for that thickness. Gimco provides full "wall-thick" protection . . . keeps inside temperatures as much as 15 deg cooler in summer and pays for itself out of winter fuel savings. Gimco is as fireproof as the rock from which it's made, resists moisture, and will not decay, pack down or dust out. Gimco is as permanent as the house, and offers no attraction to vermin or termites.

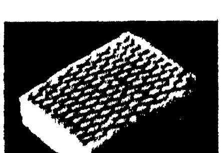
Gimco Bats are Self-Supporting— Gimco bats need only to be pushed between studdings or joists. Their own natural resiliency holds them permanently in place without additional support. Application costs are thus cut to a minimum.

Gimco Insulation for Present Homes— Gimco in granulated form can easily be blown into empty wall and ceiling spaces. It makes a permanent "wall-thick" insulation, and can be installed in any home, regardless of age, size or type of construction. For complete details, write National Gypsum Company, Buffalo, New York.

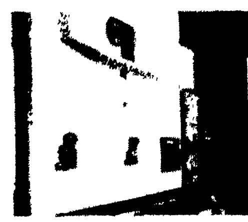
SPECIFICATIONS Gimco Sealal Batts are furnished with vapor-proof paper backing. They are made in two sizes: (1) 15 x 23 in. x wall thick; (2) 15 x 48 in. x wall thick. Gimco Sealal Batts are also available in a Double thickness (approx. 2 in. thick) in the same sizes as the Full thick Sealal Batts.

Present homes are easily and quickly insulated by blowing Gimco Rock Wool in empty wall and ceiling spaces. Insures a thick, protective blanket of uniform density.

GIMCO ROCK WOOL PRODUCTS FOR INDUSTRY



Flexfelt Blankets Efficient and adaptable rock wool insulation for boilers, heaters, furnace breachings, hot tanks and other industrial equipment. Recognized and accepted as one of the most rugged, durable insulating materials used by American industry. Besides unsurpassed insulating values, it provides fire protection, sound absorption and is adaptable to temperatures up to 1250 F.



Gimco No. 340 Insulating Cement To Engineers who are concerned with heated apparatus, Gimco 340 Insulating Cement is no stranger. For years, Gimco has been their standby for all types of jobs where an insulated cement covering is needed jobs ranging from a small heating furnace to a coating for giant boilers.

The Gimco Industrial Batt provides the most efficient and economical type of block insulation. Industrial Batts properly applied to heating and air conditioning ducts insures you of unexcelled insulating qualities. For encased application . . . in Industrial Oven and Heater Panels . . . around tanks . . . and on many types of heated equipment operating up to 800 F, the application of Industrial Batts is both highly adaptable and well recommended.

The Pacific Lumber Company

PALCO WOOL INSULATION

100 Bush Street	35 E. Wacker Drive	5225 Wilshire Blvd.	122 East 42nd St.
SAN FRANCISCO	CHICAGO	LOS ANGELES	NEW YORK

PALCO INSULATION WOOL

TRADE MARK REG U S. PAT OFFICE

WHAT IT IS

PALCO WOOL is a loose fill insulating material made from the bark of the Redwood tree, the protective covering of the world's oldest living thing. It is highly refined into an insulating material of light weight, wiry fibres of springy resilience. Recent improvements in manufacturing have made it clean, dustless and lighter in weight. In practical use PALCO WOOL has proved to be ideal for all types of construction, large or small, where resistance to conduction of heat is required. It is continuously efficient and reasonably priced, thus assuring economical performance.

USES

PALCO WOOL is suitable for any type of domestic or commercial construction as well as for the various types of Cold Storage construction.

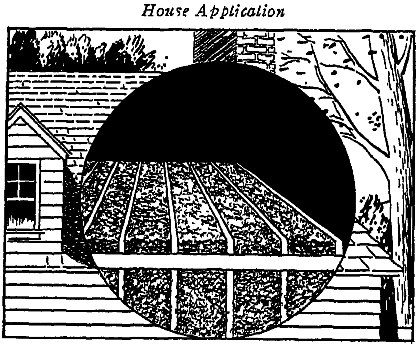
INSTALLATION

Approximately .8 of a lb of PALCO WOOL is required per square foot of 4 in. thickness. It is easily installed by hand or by machine. Between 100 and 150 lbs can be applied per hour per man. It comes in bales weighing approximately 100 lbs. Size 24 in. x 24 in. x 26 in.

Send for Insulation Manuals

Send for new 16-page booklets: "For Comfort Savings" on House Insulation or "Cold Storage Manual." Both give comparative charts and complete information on PALCO WOOL. Free sample on request.

House Application



8 PROPERTIES that make it AN IDEAL INSULATION

1. Thermal Efficiency: The established conductivity of PALCO WOOL is .26 Btu per hour per sq ft per inch of thickness per degree F difference in temperature by the Flat Plate Method.

2. Non-Settling: The fibres of PALCO WOOL possess such resilience that no settlement in a wall can occur under the most severe conditions of vibration.

3. Moisture Resistant: The fibres of PALCO WOOL are entirely lacking in capillarity and have little attraction for moisture, enabling it to remain dry and efficient when in use.

4. Permanent: The inherent antiseptic qualities of PALCO WOOL make the existence of fungus impossible. The fibres retain their resilience indefinitely.

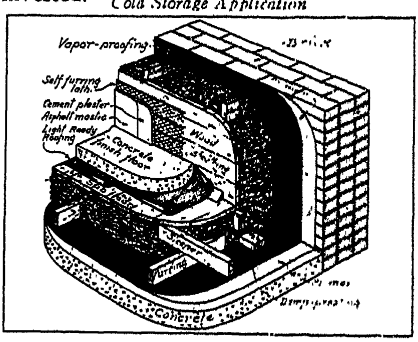
5. Vermin Repellent: PALCO WOOL is distasteful and repellent to rodents and insects.

6. Fire Resistant: PALCO WOOL, like the Redwood bark it comes from, is inherently fire resistant. As an additional protection it is *Saferized* to make it flameproof.

7. Odor Proof: PALCO WOOL is odorless itself and does not absorb or give off odors.

8. Economical: PALCO WOOL is light in weight and low in density, offering exceptional thermal efficiency per dollar invested.

Cold Storage Application



Reynolds Metals Company

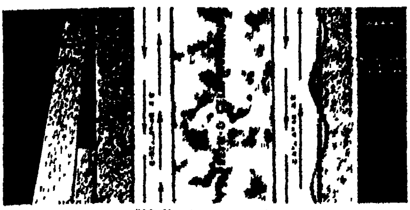
Federal Reserve Bank Building

Richmond, Virginia

NEW YORK CHICAGO LOUISVILLE SAN FRANCISCO

REYN-O-CELL

CELLULAR FIBRE INSULATION



Wall Cross Section Showing REYN-O-CELL Installed

Reyn-O-Cell stops up to 73 per cent heat flow and permits complete air circulation around framing. Thermal conductivity is .24 Btu per hour, per square foot, per 1 F. per inch thinkness. (Authority Prof. J. C. Peebles, Armour Institute of Technology).

Reyn-O-Cell is one of the most efficient barriers to the passage of heat that is commercially available today. It consists of heat retarding dead air cells. There are myriads of minute and hollow cellulose fibres, entwined and interlocked into a flexible, clean, resilient and light-weight mass.

AIR CIRCULATES FREELY

Reyn-O-Cell permits free circulation of air on both sides of the insulation, thus allowing rapid evaporation of any moisture which may occur. Possible damp rot, decay or other damage to structural materials is thereby minimized.

RODENT AND VERMIN PROOF

Reyn-O-Cell blanket type insulation insures utmost cleanliness, not only during installation, but during the lifetime of the structure. It is not subject to attack by rodents, and does not harbor vermin or other insects. It is odorless, and will not decay.

WATER-REPELLENT AND FLAME PROOF

Reyn-O-Cell will not absorb water or moisture. It has successfully withstood flame tests up to 1500 F.

REYN-O-CELL is one of the most effective sound absorption materials.

APPROVED AND ACCEPTED

Reyn-O-Cell is manufactured under constant United States Government inspection and in strict accordance with Department of Agriculture specifications. It is approved for home and industrial insulating purposes by Federal, State and Municipal bureaus, builders, architects, and heating engineers throughout the United States.

Reyn-O-Cell is ideally suited for equipment insulation. It can be furnished cut to size and in special widths up to 60 inches.

COSTS LITTLE TO INSTALL

Furnished in convenient blankets, or rolls, **Reyn-O-Cell** is adaptable to all constructions without expensive cutting or waste. REYN-O-CELL does not settle, sag or pack. For existing homes, as well as for walls, ceilings or roofs of new structures it provides maximum insulating efficiency. Labor costs for installation are exceptionally low.

REYN-O-CELL Is Easily Installed

Send for A.I.A. folder 37-a-1 describing Reyn-O-Cell; also 20-D-1 describing Reyn-O-Lath Plaster Reinforcement and Plaster Base; and 14M describing Reyn-O Wall System of 2 in. Non-Load Bearing Partitions.

The Ruberoid Co.

INSULATING PRODUCTS

Executive Offices

500 Fifth Avenue, **New York, N. Y.**

Divisional Offices

NEW YORK CHICAGO BOSTON (Millis) ERIE BALTIMORE MINNEAPOLIS MOBILE

The desire for increased efficiency of heating equipment as well as the need for fuel conservation prompts the engineer to seek the product that provides him with the most economical operating plant. The following Ruberoid Insulating Products are tabulated to enable you to choose quickly the one for the correct purpose. Greater detail and description are provided in the Ruberoid Catalog on "Heat and Cold Insulating Products," which will be sent on request.

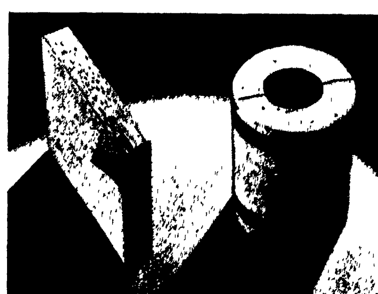
Product	Temp. Limit	Suggested Use
Hi-Temp	to 1600 F	Protective inner layer for low temperature insulations.
Sponge felt	to 750 F	For vibrating pipes and underground insulation—excellent efficiency.
85 per cent Magnesia	to 600 F	Combined efficiency and reasonable cost—General use in industrial work.
Imperial	to 500 F	For temporary lines that require efficiency and constant removal of insulation.
Watcocell	to 350 F	For a low-cost medium pressure industrial steam line.
Air Cell	to 350 F	Standard insulation for residential pipes.
Woolfelt	to 180 F	For cold and hot water lines. Recommended especially for air conditioning work.
Anti-Sweat	to 120 F	For cold water lines to prevent condensation.
Frost-proof	30 F to 100 F	To assist in the prevention of freezing in circulating water pipes exposed to cold.

Sheet and Block Insulations

All of the above products are also made in sheet and block form to whatever thickness may be required. Standard sizes are usually 6, 12, 18 or 36 in. wide x 36 in. long. In this form they are used for insulating flat or irregular surfaces, such as tanks, breechings, furnaces, etc.

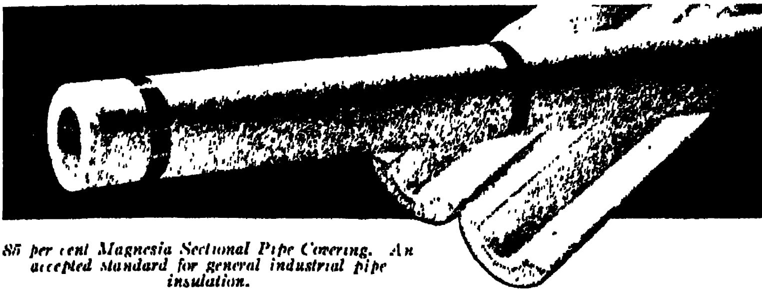
Imperial Pipe Covering

This is a laminated asbestos paper insulation that has been indented to use 24 laminations of asbestos paper per inch thickness. Its efficiency makes it satisfactory for medium pressure steam work in industrial plants. Its construction is ideal for vibrating conditions. It can be used on temperatures to 500 F. Furnished with either plain canvas jacket or the new pyroxylin finish. Made in 3 foot sections. Brass lacquered or black Japanned bands furnished for application.

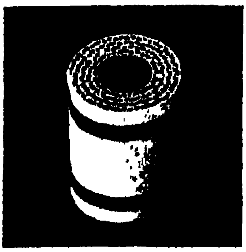


Sponge Felt

Sponge Felt Pipe Insulation is for temperatures up to 750 F. Composed of asbestos-felt paper with sponge content evenly distributed. There are 37 to 42 laminations to the inch. This produces a uniformly round and rigid insulation of extreme density. Over the outside is a heavy canvas jacket. Particularly adapted for use on vibrating lines because of its rigidity. Packed in crates in 3 foot sections. Brass lacquered or black Japanned bands are furnished for application.

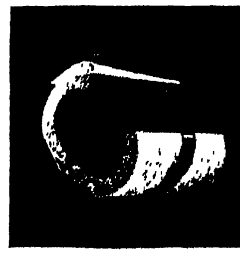


85 per cent Magnesia Sectional Pipe Covering. An accepted standard for general industrial pipe insulation.



Air Cell Pipe Covering A low-cost insulation for residential use.

Woolfelt Pipe Covering For the insulation of pipes carrying hot or cold water also prevents condensation under normal operating conditions.



Insulating Cements

For the finishing of sheet and block insulation and the insulating of irregular surfaces, such as valves, unions, flanges, etc., the Ruberoid line of insulating cements is complete. This group of plastics not only uses as its base asbestos, but also takes advantage of such excellent natural products as magnesia, mineral wool and Vermiculite.

Asbestos Cements Factory Prepared Grades AA, A, IIF.

Asbestos Cements Mine Run Grades 115, 214.

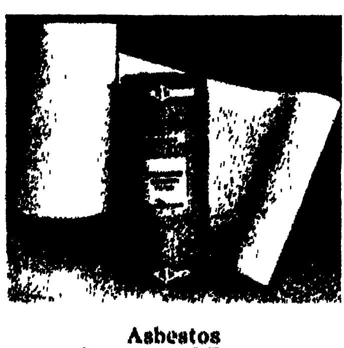
Magnesia Cement 85 per cent Magnesia.

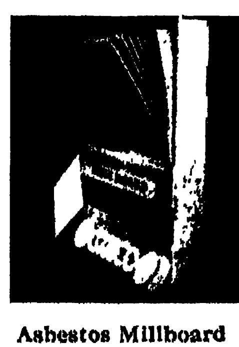
High Temperature Cement Grade H.T.

Mineral Wool Cement Grade R-W.

Vermiculite Cement Grade A-11.

RU-BER-OID Asbestos Insulating Papers and Millboard





Asbestos Paper

Made of pure asbestos fibre and small percentage of binding material. Possesses unusual strength. This fine fire-resisting sheet may be obtained in 8, 10, 12, 14, 16 and 32 lb weights ranging in thickness from 0.019 to 0.0625 or $\frac{1}{16}$ in. Rolls 18, 24 and 36 in. wide, weight 50 or 100 lb. Color, blue white.

Asbestos Corrugated Paper

Made entirely from high quality asbestos felt paper by cementing flat sheet firmly with a $\frac{1}{4}$ in. corrugated sheet which forms dead air spaces. Flexible. Efficient for insulating warm air pipes and ducts. 36 in. wide. Available in 250 sq ft. rolls weighing approximately 46 lbs.

Asbestos Millboard

Asbestos Millboard is a rigid insulating board made of high quality asbestos fibres and non-organic binder. Has exceptional strength and whiteness. Cuts or drills easily. Standard or embossed finish. For temperatures to 1100 F. Sheets 42 x 48 in. Various thicknesses and weights.

United States Gypsum Company

General Offices: 300 W. Adams Street, **Chicago, Ill.**

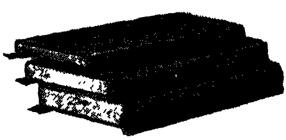
INSULATION PRODUCTS

Blanket **Decorative** **Structural**

Reflective **Vapor Barriers**

BLANKET

Red Top Insulating Blankets—Mineral wool fibers of great strength and uniformity, clean and free from non-insulating material are felted into light-weight blankets and securely fixed to an efficient asphalt-type vapor barrier. The vapor barrier forms the warm side of a complete paper enclosure, providing proper resistance to the passage of vapor, while a tough, perforated, vapor permeable paper on the cold side prevents the accumulation of any moisture within the blanket.



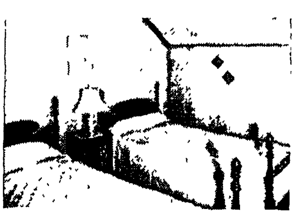
Red Top Insulating Blanket

Red Top Insulating blankets are so constructed that the insulating material is uniform in thickness throughout the length of each roll. They are as thick at the edges as in the center. Flanges at either side create an air space immediately back of the lath and plaster and effectively seal the joint against vapor leakage. Made in three thicknesses: One inch, medium and thick, in rolls of 50, 75 and 150 square feet (net area), respectively. Also available in bats 3 feet long in the same thicknesses.

Red Top Junior Bats 11 x 14 In.—Composed of the same fibers, 4 in. thick, to fit standard stud and joist spaces reduce costs for buildings which do not require vapor protection. Their "springy" fibers help hold the bats securely in position.

DECORATIVE INSULATION PRODUCTS

Weatherwood Building Board—An insulating wallboard 4 ft wide which fits across three standard stud or joist spaces, made in full foot lengths from 4 ft to 12 ft, inclusive, ½ in. and 1 in. thick in a "skin" finish in Ivory and a textured finish in either Ivory or Tan shades. Nailed directly to studs and joists this pleasing wall finish effectively insulates and decorates.

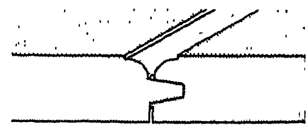


Decorative

Weatherwood Plank—These half-inch "planks" of insulative wood fiber are manufactured in widths of 6, 8, 10, 12 and 16 inches, and in lengths 8, 9, 10 and 12 feet. They take advantage of the ingenious "Ogee" edge on their long edges (see cut) which conceals the nails and fully accommodates any necessity for movement in the board because of expansion or contraction without producing a crack through which air may leak to discolor the surface and reduce the insulation value of the material. When applied horizontally, they are blind nailed to the studding, and to cross furring when used vertically.

All sizes are shipped in a blend of gray and tan shades, in bundles of one size. When combined in the variations in shade and width, Weatherwood Plank produce maximum values in both insulation and decoration.

Weatherwood Tile -Thicknesses of ½, ¾ and 1 in. in the following sizes: 12 x 12 in., 12 x 24 in., 16 x 16 in., 24 x 24 in., 16 x 32 in. and 24 x 48 in. The "Ogee" design, on all edges, produces a result comparable to that were a completely continuous single piece of board used and likewise enhances the decorative effect. Nailed "blind" to construction in new buildings or they may also be cemented over plaster to both redecorate and increase insulation values in existing buildings.



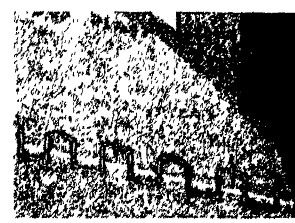
Ogee Edge

STRUCTURAL INSULATION

Weatherwood Insulating Lath Adds all of the purposes of a good lath to ½ in. of wood fiber board insulation. Each piece is 18 x 48 in. long. The long edges have a steel reinforcing member (optional), which the plaster completely surrounds, reducing possibilities of cracks at the joints between adjacent boards. The excellent bond between the fibrous board face and the plaster eliminates any necessity for plaster keys, saving plaster and labor quantities.

Weatherwood Sheathing Manufactured in pieces 2 ft x 8 in. each 25/32 in. thick, with the long edges tongued and grooved for horizontal application as sheathing material to the exterior walls of frame buildings. Both sides and all edges are asphalt coated, making the board highly water resistant.

Roof Insulation In sheets 22 x 47 in. ½, 1, 1½ and 2 in. thick. All but the ½ in. size are supplied laminated, and may be obtained with square or "shiplapped" edges.



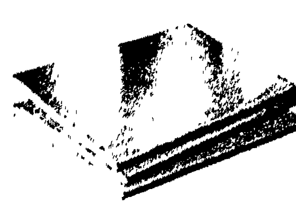
Weatherwood Insulating Lath

REFLECTIVE INSULATION

Insulating Rocklath This universally used plaster base is also available, with a highly polished aluminum foil back, augmenting its well known fireproofing and structural qualities with real insulation value at very slight additional cost.

Insulating Sheetrock Where unit or dry wall materials are used for the interior finish Insulating Sheetrock in ⅜ in. or ½ in. thickness combines its qualities with those of aluminum foil reflective insulation for use in exterior wall construction and top floor ceilings.

Vapor Barriers In existing buildings, where condensation has caused difficulties, Insulating Sheetrock, applied over the plaster will supply a high efficiency aluminum foil vapor barrier and fully restore the interior finish. Sheetrock will take any paint or wall paper decoration.



Reflective Insulation

Wood Conversion Company

First National Bank Building, **St. Paul, Minn.**

NEW YORK CHICAGO TACOMA DALLAS

BALSAM-WOOL AND NU-WOOD INSULATIONS

BALSAM-WOOL	NU-WOOD	NU-WOOD
Sealed Insulation	**Kolor-Fast Tile**	**Lath**
Acoustical Blanket	**Kolor-Fast Plank**	**Roof Insulation**
Sound Deadening	**Kolor-Fast Board**	**Industrial Insulation**
Industrial Insulation	**Kolor-Fast Wainscot**	**Refrigerator Insulation**
Refrigerator Insulation	**Kolor-Fast Sheathing**	**KOLOR-TRIM Pre-decorated Moldings**

BALSAM-WOOL—The Double Value Sealed Insulation

The basic rightness of Balsam-Wool insulation principles has been recognized for 19 years. Constant improvement has made this insulation an acknowledged leader. Today, the new DOUBLE-VALUE BALSAM-WOOL offers greater moisture protection, increased efficiency, and increased thickness. In addition, Balsam-Wool SEALED insulation provides such outstanding Double Advantages as Double Sealing, Double Moisture Barriers, Double Wind Barriers, Double Air Spaces, Double Bonding, Double Fastening.

Balsam-Wool Spacer Flange

Balsam-Wool is an insulating mat of fleecy wood fibers, enclosed between a protective covering of double layers of asphalted craft, chemically treated to resist fire, rot, termites and vermin—92 per cent of the mat volume is dead air.

Balsam-Wool SEALED Insulation is fabricated at the factory to a controlled density of 2.2 lb per cubic foot. The mat has a coefficient of 0.246 Btu per hour, per square foot, per 1 degree F difference in temperature, per 1 in. thickness.

As applied, factory efficiency is assured. The Spacer Flange on each edge folds over and is fastened to framing members with a staple hammer, assuring important air space front and back.

Application is quick and easy

Double-Value Balsam-Wool is available in two new increased thicknesses—STANDARD and DOUBLE-THICK, in widths of 16, 20, 24 and 33 inches. Wall-thick Balsam-Wool is available in widths of 16, 20 and 24 inches.

NU-WOOD INTERIOR FINISH—STRUCTURAL INSULATION

Nu-Wood Kolor-Fast and Sta-Lite Interior Finish (Tile, Plank, Board and Wainscot) is applicable either to new construction or to existing buildings. It offers varied and pleasing decoration, also insulation and acoustical value.

Nu-Wood Insulating Lath has several times the bonding strength of wood lath—continuous surface eliminates dirty lath marks, reduces cracks, V-joint resists trowel pressure in both directions assures unbroken insulation value.

Nu-Wood Insulating Sheathing is surfaced on both sides with double coats of special moisture proofing compound. Large board—speed erection · stronger, windproof, insulated construction.

OWENS-ILLINOIS
INSULUX
Glass Block

Owens-Illinois Glass Company

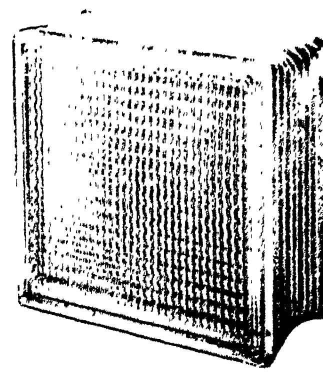
INSULUX PRODUCTS DIVISION

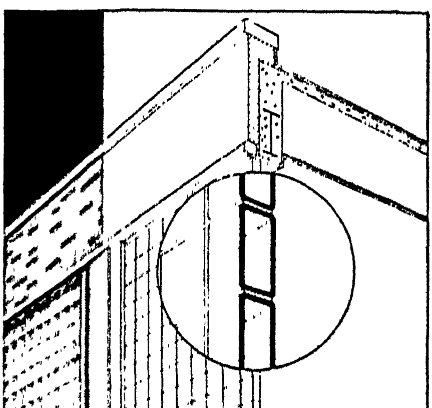
Toledo, Ohio

Dealers in All Principal Cities

Insulux Glass Block Give Better Control of Interior Conditions

Insulux Glass Block are hollow, partially evacuated block, $3\frac{7}{8}$ inches thick, with ribbed or patterned faces. Laid up in mortar in solid panels, they form a light-transmitting area that also offers high insulation value. Proper use of Insulux Glass Block results in better control of interior conditions, and therefore greater efficiency and lower initial and operating costs for cooling and heating plants.





Lower Heat Transmission

Tests on conductivity of Insulux Glass Block show that the heat transmission of Insulux is approximately the same as for a concrete wall 16 inches thick or a brick wall 8 inches thick. The U factor for smooth face block is .49 Btu per sq ft per hour per degree difference in temperature. For ribbed block, the U factor is .46. This test data is available for inspection by engineers.

Reduction of Solar Heat

In a comparative test of solar heat transmission, a single glazed steel sash transmitted 94 per cent more heat than an Insulux panel. As with sash, however, Insulux panels transmit less solar heat if properly oriented and well shaded. There is variation in the solar heat transmission of different designs of Insulux data will be furnished on request.

Better Insulation

The cross-section drawing above shows why Insulux Glass Block panels give higher insulation than ordinary light-transmitting areas. The glass block, partially evacuated and with thick faces, lower the conductivity and solar heat transmission of the light-transmitting area. Air infiltration is eliminated. The better insulation provided by Insulux is a factor in planning air conditioning and heating equipment and operating costs.

Designs, Sizes, Erection

There are 11 designs of Insulux for both residential and industrial use. Block available in three sizes. Panels are easily and quickly erected by bricklayers. We will gladly supply any technical information and advice on installations on request.

Pittsburgh Corning Corporation

Grant Building, Pittsburgh, Pa.

Distribution through Pittsburgh Plate Glass Company warehouses in principal cities and by the W. P. Fuller Company on the West Coast.

PC **Glass Blocks** allow the economical use of large glass areas, reduce heat loss in cold weather and materially aid air-conditioning. This is because each PC Glass Block contains a sealed-in dead-air space that is an effective retardant to heat transfer.

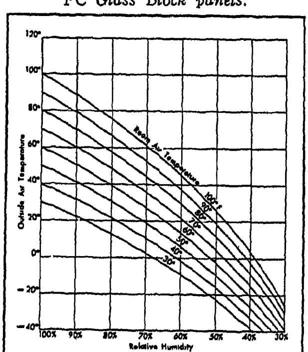
Thermal Insulation

Tests run by nationally recognized laboratories have established the value of glass blocks for insulation of light-transmitting areas. These tests have proved that with glass block panels, heat loss is slightly less than half that experienced with single-glazed windows. In computing heat losses through panels for most design purposes, it is recommended that a "U" value of 0.46 to 0.49 be used for all block sizes and face patterns. For complete data on heat transfer values see the section on heat transfer elsewhere in this Guide—page 111.

Surface Condensation

Due to high insulating value, condensation will not start forming on the room side of glass block panels until outside air has reached a temperature much lower than that necessary to produce condensation on single-glazed windows. The accompanying chart shows at what temperatures condensation will form.

Outdoor temperature required to produce condensation on the room side surface of PC Glass Block panels.



For example, with inside air at 70 F and relative humidity at 40 per cent, condensation will not begin to form on the interior surfaces of a glass block panel until an outdoor temperature of -14 deg is reached.

Under similar conditions with single-glazed sash, moisture will begin to form when the outdoor temperature reaches $+33$ F.

Solar Heat Gain

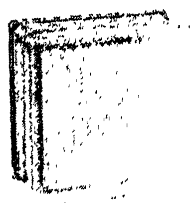
The use of glass blocks for light-transmitting areas results in a marked reduction in total solar heat gain as compared with ordinary windows. This factor is of considerable advantage in buildings that are properly air conditioned, but does not eliminate the need for adequate ventilation or shading in non-air-conditioned rooms.

For data on solar heat gain through glass blocks see the table in the solar radiation section of this Guide—page 149. This table is for standard pattern glass blocks. Where LX-75 blocks (with Fiberglas screen insert) are used, these figures may be reduced approximately 30 to 40 per cent.

PC Glass Blocks Aid Air-Conditioning

The three chief aims of air-conditioning - temperature control, humidity control and cleansing of air are all aided by the use of PC Glass Blocks. Heat loss is less in winter—heat gain is less in summer. Ideal humidity conditions are much more easily maintained without undue condensation. Solar heat transmission and radiation are reduced. Dirt can't filter in, for each panel is a tightly sealed unit.

Sizes and Shapes Available



PC Glass Blocks are available in eight attractive patterns, some of the patterns being designed for special control and direction of transmitted daylight. For complete information on the sizes and shapes of PC Glass Blocks, and for illustrations of the many patterns available, write the Pittsburgh Corning Corporation, Pittsburgh, Pa., or call the nearest Pittsburgh Plate Glass Company warehouse.

• • •

Additional technical data, including detailed figures on thermal insulation, solar heat gain, surface condensation, light transmission and construction data, will be furnished on request.

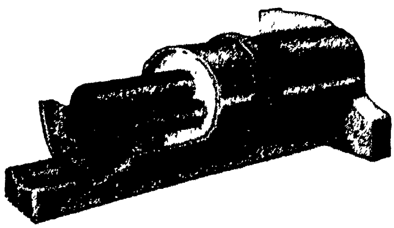
ADSCO
PRODUCTS
for STEAM
SERVICE

AMERICAN DISTRICT STEAM COMPANY

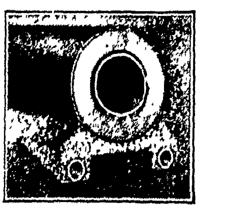
NORTH TONAWANDA, N.Y.

IN BUSINESS OVER SIXTY YEARS

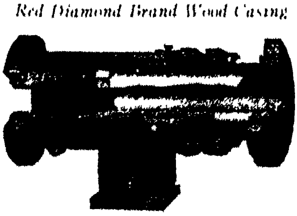
Branches and Agents in Principal Cities



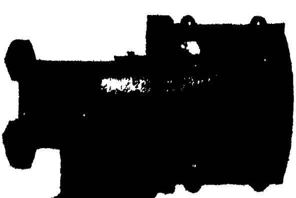
Tile Conduit with ADSCO Filler Insulation - a "Fiberglas" Product



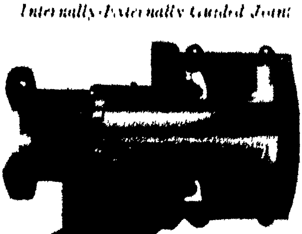
Red Diamond Brand Wood Casing



Internally Guided Joint



Internally-Externally Guided Joint



Piston-Ring Type Joint

OVER SIXTY YEARS EXPERIENCE IS BUILT INTO THE DESIGN AND MANUFACTURE OF DEPENDABLE ADSCO PRODUCTS FOR PIPE LINES

For over sixty years ADSCO engineers have specialized in the design and application of pipe fittings and accessory equipment for underground and surface steam, water, oil and other piping systems. An extensive, modern plant including foundry, machine shop, casing mill, shipping and storage facilities enable ADSCO to produce high grade products by skilled workmen under expert supervision.

LEADING MAKERS OF EXPANSION JOINTS

As pioneer manufacturers of expansion joints for pipe lines, ADSCO is the largest single producer of such equipment in the world. We offer the most extensive line of packless and slip type joints in various types to meet the requirements of any pipe line expansion and contraction problem. In addition, ADSCO produces all of the related equipment necessary to the permanent installation of efficient pipe lines, including tile conduit and wood casing for underground lines, pipe supports, saddle plates, alignment guides, steam traps, condensation and flow meters, storage and instantaneous water heaters, strainers, manhole frames, and vapor heating specialties.

ENGINEERING ASSISTANCE

ADSCO engineers welcome the opportunity of working with industrial plants, utility companies, colleges, institutions, and government departments in the solution of their pipe line expansion and contraction problems and correspondence is invited giving the details of any proposed piping installation.

WRITE FOR ADSCO CATALOG No. 35

All ADSCO products are illustrated and described in the latest ADSCO Catalog No. 35 containing over 130 pages of informative data for the specification and purchase of dependable products for underground or surface pipe line distribution systems. Write for your copy today to the American District Steam Company, 65 Bryant St., North Tonawanda, New York.

H. W. Porter & Co.

INCORPORATED

Newark, New Jersey

Permanent Protection and Insulation for Underground Pipe Lines

BALTIMORE, MD. CHARLOTTE, N. C. RICHMOND, VA. WASHINGTON, D. C.

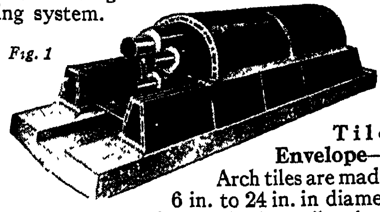
Therm-O-Tile

REG US PAT OFF

STEAM CONDUIT SYSTEMS

For Central Heating—Therm-O-Tile is a complete conduit system for the permanent support, protection, and insulation of the underground mains of a central heating system.

Fig. 1



Tile Envelope—Arch tiles are made 6 in. to 24 in. in diameter, and with 5 different size base tiles they produce 27 different conduit cross sections.

Foundation—The base of the system is a thick concrete slab poured directly in the trench bottom, reinforced with steel when installed over a filled or boggy ground. See Figs. 1, 4, 6 and 7.

Drainage—The drainage system of the conduit is entirely internal, accurately and permanently sloped, open to inspection at manholes, and of ample capacity to keep the pipe space dry at all times without any possibility of becoming clogged with silt or vegetation. See Figs. 6 and 7.

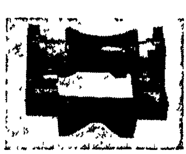


Fig. 2—Pipe Support for Single Pipe.

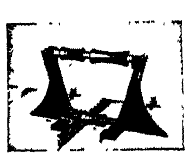


Fig. 3—Pipe Support for Three Pipes.

Pipe Support—All pipes are supported on cast-iron adjustable supports resting directly on the concrete base independent of the tile envelope. Figs. 2, 3, 4, 6 and 7.

Accessibility—All piping is installed before tile is placed, giving complete accessibility for welding, testing and insulation. Pipe fitters work on convenient concrete slab "walkway." Figs. 1 and 4.

Strength—Due to immovable concrete base and arch construction of extra heavy tile members, this conduit will sustain any roadway traffic load usually encountered without extra reinforcement.

Insulation—Either sectional pipe covering or Thermobestos waterproof fibre filling may be used for insulation, as the insulation space is kept dry at all times, by the internal drain. For single or double pipe lines, sectional insulation of economical thickness is recommended; for multiple pipe lines, a filler type of insulation is usually more economical in first cost. See Figs. 6 and 7.

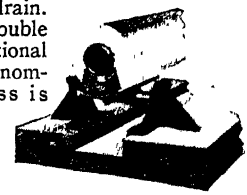


Fig. 4—Pipe Saddle. Permits full thickness of insulation between pipe and roller.

Waterproofing— Under normal soil conditions, this conduit is waterproof. If marshy ground or partially submerged conditions are encountered, the conduit may be made completely waterproof by the use of membrane waterproofing applied under the slab on a sub-base and carried completely over the tile envelope.

Fig. 5—Anchor Block. Fits directly in line with Base Tiles.

Efficiency—The degree of thermal efficiency secured depends upon the type and thickness of insulation used. This conduit, due to its sealed air chambers in the tile and dry insulation space, adds to the normal efficiency of the insulating material on the pipe lines.

Representatives—Therm-O-Tile is also sold and installed by Johns-Manville Construction Units located in all principal cities.

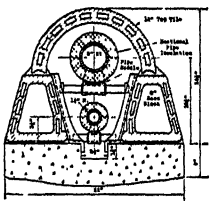


Fig. 6—Single or Double Pipe Lines Using Sectional Pipe Insulation.

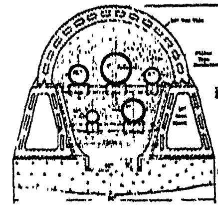


Fig. 7—Multiple Pipe Lines Using Filler Type Insulation.

The Ric-wiL Company

Agents in Principal Cities

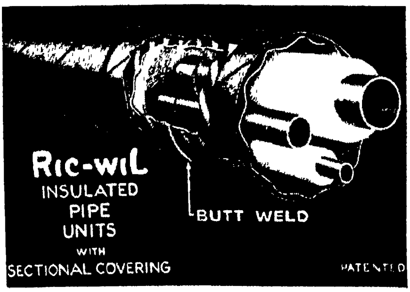
RIC-WIL

Established in 1910

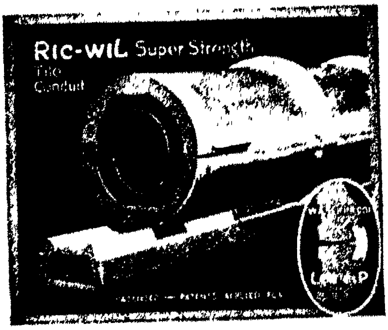
CONDUIT SYSTEMS FOR UNDERGROUND STEAM PIPES

Union Commerce Bldg., **Cleveland, Ohio** **New York, Chicago, San Francisco**

Ric-wiL Insulated Pipe Units Pre-fabricated, pre-sealed, ready-to-install units, ideal for speed and economy. Armco Iron Conduit is the foundation supporting heavy asphalt shell of any desired thickness a permanent housing for the insulated pipe which is surrounded with a protective air-space. Ample structural strength, lightweight and watertight. Furnished in any lengths, for single or multiple pipes, with any kind of steam pipe or insulation, for underground or overhead steam lines. *Welded* couplings used if preferred. Write for latest Unit Bulletin.



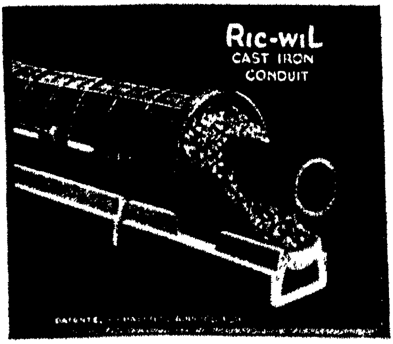
Types of Sectional Conduit Ric-wiL SuperTile Conduit, shown below, with Dry-paC Insulation, is an extra weight, heavy duty system designed for use under highway traffic or in especially wide or deep trenches. Vitrified tile, split on the job, with Loc-liP Side Joints, interlocking construction throughout. Same design also furnished in standard weight tile (Type F). Tile is bell and spigot design, lined or unlined, and comes in 24 in. sections, 4 in. to 27 in. inside diameter. For extra heavy duty under railways, Ric-wiL is made of cast iron in 2 or 4 foot sections. Where continuous concrete base, poured on job, is desired, and reduced labor cost not essential, Ric-wiL Universal Type System is recommended. Each system supplied complete with proper pipe supports, accessories, and insulation as specified. Separate bulletin on any one of these Ric-wiL types supplied on request.



Base Drain- Standard Base Drain is vitrified salt glazed tile for tile conduit and extra heavy tile or cast iron for the cast iron conduit, in 24 in. lengths. Made in three sizes to support and drain properly all conduit sizes.

Insulation Ric-wiL Dry-paC Waterproof Insulation is high-grade long fibre asbestos, specially processed. Any grade of commercial hand packed insulation can be furnished, also sectional pipe covering. For lined conduit, diateomaceous earth mixture is molded and keyed inside the tile.

Engineering Service—Full cooperation with architects and engineers. Installation supervision if desired. Write for Catalog Bulletin with valuable underground data.



Ingersoll Steel & Disc Division

Borg-Warner Corporation

310 So. Michigan Ave., **Chicago, Illinois** — Distributors in the Principal Cities

Ingersoll **KOOLSHADE*** *Sun Screen*

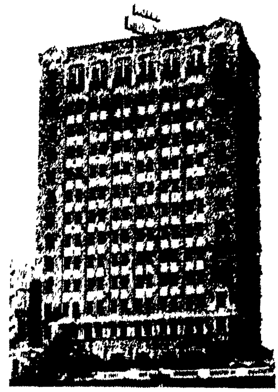
"It's cooler in the shade!"

A Metal Fabric for Screening Windows that Reduces the Solar Load in Sun-Exposure Rooms as Much as 80 to 85 per Cent.

KOOLSHADE is still considered a new product, but more than 10,000 installations—in every knid of building, from giant defense plants to small dwellings—attest the acceptance it has already had. Made like a tiny bronze Venetian Blind . . . with minature slats fixed at a 17 degree angle to screen out sun heat and let in soft, glareless light . . . KOOLSHADE is unlike any other product.

KOOLSHADE has had such ready acceptance among Air Conditioning Engineers because it offers the highest efficiency in cutting down the Solar Load—and does this without shutting off light or vision.

- By stopping 80 to 85 per cent of the Solar heat outside the window glass, KOOLSHADE often reduces the original cost of air conditioning equipment.
- It saves in operating cost—as much as 25 to 50 per cent where there are large windows.
- It reduces the need for zoning for sun effect.
- It forestalls complaints due to extreme sun heat.
- It maintains lower temperatures in non-cooled rooms.



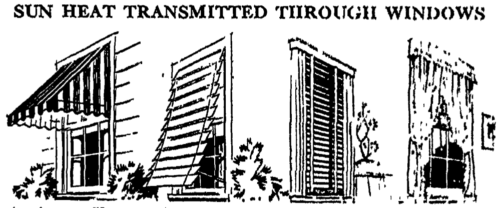
Plaza Hotel, Corpus Christi - Engineers report KOOLSHADE saved $4,000 on first cost of air conditioning, $45 monthly on operation.

Automatic Sun Shade—KOOLSHADE is always in position . . . functions automatically . . . protects most when needed. The flat horizontal wires, held at a FIXED angle of 17 deg, fully stop direct heat rays when the sun is 38½ deg or more above the horizon—the heat of the day in all seasons. KOOLSHADE gives sun-exposure rooms the effect of cool "north" light.

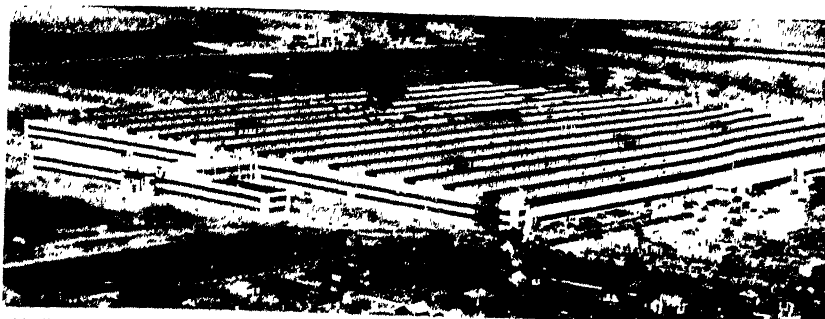
*Registered Trade Mark—Property of Ingersoll Steel & Disc Division, Borg-Warner Corporation.

KOOLSHADE
SUN LOAD 19 PER CENT

SUN HEAT TRANSMITTED THROUGH WINDOWS



Awnings in Use	Outside Venetian Blind or Shutter	Inside Venetian Blind	Half-Drawn Shade
Sun Load 22 Per Cent to 28 Per Cent	Sun Load 22 Per Cent	Sun Load 58% or more	Sun Load 68% or more



AMERICA'S LARGEST AVIATION FACTORY Wright Aeronautical Corporation's new 50-acre plant at Cincinnati has more than 800 sun-exposed windows equipped with KOOLSHADE.

Economy. KOOLSHADE is strong, tough and should last for many years. Nothing to rot, rattle, or blow off.

Other Advantages include full insect protection, free ventilation, reduction of fading of fabrics, and fire-safety.

Specifications of Fabric. Made of fine quality bronze, with 17 horizontal wires per in. and vertical wires spaced ½ in. apart. In widths up to 72 in.

Framing and Installation. Framed and installed like ordinary insect screens. Frames may be either wood or metal always full length and always outside the window glass. Re-wiring of present screen frames is entirely practical.

Where to Buy. Authorized KOOLSHADE distributors and dealers in all principal cities offer adequate facilities to assure correct frame design and proper installation.

Technical Proof of Performance. Reports of independent tests and calculations provide ample evidence of KOOLSHADE'S efficiency (one table from a report by the Pittsburgh Testing Laboratory is reproduced below). Of particular interest is a recent comparison test by Clyde R. Place, noted New York engineer, showing effect of KOOLSHADE on the operating cost and efficiency of air conditioning.

Full Literature, including these reports, will be sent on request.

DATA FROM PITTSBURGH TESTING LABORATORY

FROM CALCULATIONS BASED ON ACTUAL TESTS

Solar Radiation Transmitted Through Windows Equipped with KOOLSHADE Sun Screen

For 40 deg. latitude, on July 21st All figures given represent B.t.u. per sq. ft. per hour.

▼TIME ➤	SOLAR RADIATION	NE	EAST	SE	SOUTH	
6 AM	Intensity Incident to Vertical Surface (1)	72	80	40		6 PM
	Transmitted thru Window with KOOLSHADE (2)	33	38.5	13		
7 AM	Intensity Incident to Vertical Surface	143	180	112		5 PM
	Transmitted thru Window with KOOLSHADE	38.5	60.5	22		
8 AM	Intensity Incident to Vertical Surface	143	211	155	8	4 PM
	Transmitted thru Window with KOOLSHADE	17	42.5	20.5	0	
9 AM	Intensity Incident to Vertical Surface	104	192	168	46	3 PM
	Transmitted thru Window with KOOLSHADE	8	22	17	2	
10 AM	Intensity Incident to Vertical Surface	46	143	150	77	2 PM
	Transmitted thru Window with KOOLSHADE	1.5	11.5	13	4	
11 AM	Intensity Incident to Vertical Surface		75	121	95	1 PM
	Transmitted thru Window with KOOLSHADE		3.5	8.5	5.5	
12 M	Intensity Incident to Vertical Surface			73	103	12 M
	Transmitted thru Window with KOOLSHADE			3.5	6	
		NW	WEST	SW	SOUTH	◄TIME▲

(1) Data from the A.S.H.V.E. Guide 1940.

(2) Figures represent Solar Heat Gain in EXCESS of heat gain by conduction through window glass.

Libbey•Owens•Ford Glass Company

Nicholas Building, **Toledo, Ohio**

BLUE RIDGE HEAT-ABSORBING AND GLARE-REDUCING GLASS

What is Blue Ridge AKLO?

Blue Ridge AKLO is a blue-green heat-absorbing and glare-reducing glass which is unique in its low expansion and high solar heat absorbing properties. It is unusually successful in commercial buildings and industrial plants, while its glare-reducing qualities, when frosted, are equally important in promoting employee safety and increased production.

AKLO Resists Thermal Shocks

AKLO industrial glass cannot be compared with ordinary glass in heat absorption or expansion characteristics. Its coefficient of expansion is .0000040 per degree Fahrenheit—approximately 20 per cent less than ordinary glass. Its use provides a positive reduction in room and shop temperatures, creating more comfortable working conditions. Frosted AKLO, in ¼-inch thickness, absorbs approximately 48 per cent of the solar heat.

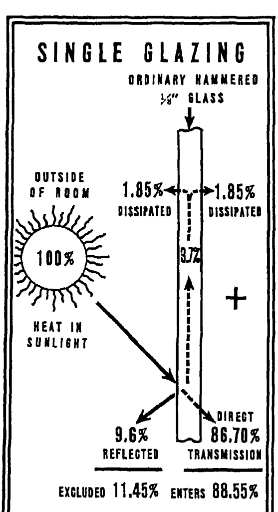
Provides Many Indirect Savings

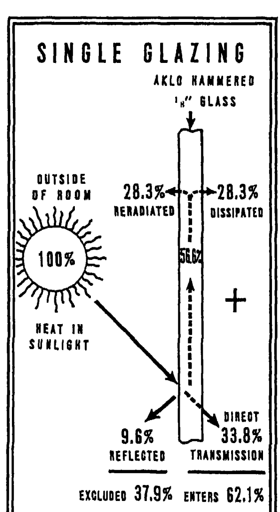
Indirectly, FROSTED AKLO Glass, by its glare-reducing quality, provides additional savings in the shape of reduced rejections, decreased errors on the part of workmen and substantially improved working efficiency.

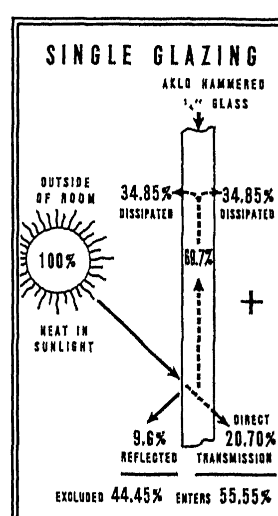
Improves Air Conditioning Efficiency

Because it eliminates up to 48 per cent of the solar heat, AKLO Heat-Absorbing and Glare-Reducing frosted glass increases the efficiency of air conditioning equipment. This is particularly important in commercial and office installations for it not only makes it easier to attain a more even temperature and humidity, but results in more satisfied owners and customers.

Additional information is available from your local L·O·F Distributor or by writing direct to the Blue Ridge Sales Division, Libby·Owens·Ford Glass Company, Nicholas Building, Toledo, Ohio.







Libbey·Owens·Ford Glass Company

Nicholas Building, **Toledo, Ohio**

WINDOW CONDITIONING (Double Glazing) FOR HEATING ECONOMY

If America's 37,000,000 homes were insulated with Window Conditioning, (double sash or storm windows and doors) as well as wall and attic insulation, the nation's fuel bill could be reduced by as much as $1,000,000,000, according to government statisticians.

Every heating and ventilating and air conditioning engineer knows the enormous potential savings available with complete insulation. With Window Conditioning alone, exacting tests have shown fuel reductions as much as 24.2 per cent (see table below).

Today, insulation of homes, commercial buildings and certain industrial plants is more important than ever before. With the need for reduced fuel consumption, complete insulation is the ideal solution for a large portion of our defense savings.

Window Conditioning, however, means far more than a reduction in fuel bills. It eliminates cold drafts . . . reduces to a minimum the possibility of window fogging . . . is essential to satisfactory winter air conditioning with its higher healthful humidity. It keeps the inner pane relatively warm even though the outside glass is as cold as the outside atmosphere—reducing to a minimum the condensation on the inner pane.

Window Conditioning is made possible by a wide variety of double windows and it is possible to match practically every style of window without sacrificing visibility. Types range from the ordinary low-cost, single pane, hook-on storm sash to more elaborate prefabricated storm windows with removable glass sections for easy cleaning from the inside.

This Table Shows Fuel Savings and Comparative Heating Costs of Four Types of Houses with and without Window Conditioning and Insulation.

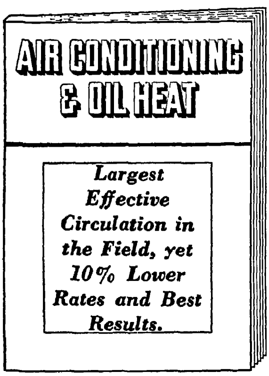
								
Attic area vented above insulation	1488.5 sq. ft.		770 sq. ft.		1143 sq. ft.		995 sq. ft.	
Sidewalls net	2447.7 " "		1634 " "		1332 " "		1197.5 " "	
Window area	540.3 " "		326 " "		363 " "		285 " "	
Crack length	590.4 lin. "		389 lin. "		422 lin. "		365 lin. "	
Unheated floor	None		None		None		None	
ZONE 9 — Heating cost — no insulation oil 7 cents per gallon	$315.50		$190.60		$211.00		$173.80	
ZONE 9 — Heating cost if attic is insulated	254.50		159.50		164.70		133.90	
ZONE 9 — Heating cost with window conditioning	241.25		144.55		159.50		131.90	
ZONE 9 — Savings due to insulation 3", minimum wool in attic floor	61.00	19.3%	31.10	16.3%	46.30	21.9%	39.90	23.0%
ZONE 9 — Savings due to window conditioning	74.25	23.5%	46.05	24.2%	51.50	24.4%	41.90	24.1%
ZONE 9 — Savings with both TOTAL	$135.25	42.8%	$ 77.15	40.5%	$ 97.80	46.3%	$ 81.80	47.1%
ZONE 7 — Heating cost — no insulation oil 7 cents per gallon	$245.00		$149.30		$161.00		$136.00	
ZONE 7 — Heating cost if attic is insulated	198.00		124.90		125.00		104.40	
ZONE 7 — Heating cost with window conditioning	186.25		112.80		123.00		103.80	
ZONE 7 — Savings due to insulation 3", minimum wool in attic floor	47.00	19.2%	24.40	16.3%	36.00	22.4%	31.60	23.3%
ZONE 7 — Savings due to window conditioning	58.75	24.0%	36.50	24.4%	38.00	23.6%	32.20	23.7%
ZONE 7 — Savings with both TOTAL	$105.75	43.2%	$ 60.90	40.7%	$ 74.00	46.0%	$ 63.80	47.0%
ZONE 5 — Heating cost — no insulation oil 7 cents per gallon	$176.00		$106.35		$116.30		$ 96.30	
ZONE 5 — Heating cost if attic is insulated	142.50		88.95		91.20		74.00	
ZONE 5 — Heating cost with window conditioning	134.25		80.80		88.40		73.85	
ZONE 5 — Savings due to insulation 3", minimum wool in attic floor	33.50	19.0%	17.40	16.4%	25.60	21.9%	22.30	23.2%
ZONE 5 — Savings due to window conditioning	41.75	23.7%	25.55	24.0%	28.40	24.3%	22.65	23.5%
ZONE 5 — Savings with both TOTAL	$ 75.25	42.7%	$ 42.95	40.4%	$ 54.00	46.2%	$ 44.95	46.7%

All Libbey·Owens·Ford Distributors and Dealers have complete Window Conditioning information. Ask them about it, or write direct to the Libbey·Owens·Ford Glass Company, Nicholas Building, Toledo, Ohio.

AIR CONDITIONING & OIL HEAT

232 Madison Ave. Lex. 2-4566 **New York, N. Y.**

San Francisco	Mid-West	Baltimore
Don Harway & Co.	A. E. Delgado	Candler Bldg.
420 Market St.	811 Michigan Ave.	Lexington 7065
Exbrook 6029	Evanston, Ill.	
	University 7550	



Established in 1928 as "OIL HEAT," this paper covered the manufacture, sale, installation and servicing of oil burners its first 7 years. In 1935, its title was changed and the editorial content expanded to cover air conditioning and heating also. This inspired and kept pace with the field itself.

Oil burner manufacturers and dealers, being progressive in both merchandising and technical problems, dominate air conditioning in many sections. Oil fired heating and air conditioning has grown steadily in public favor. 2,400,000 burners are now operating in the U. S. A. and our readers are servicing them.

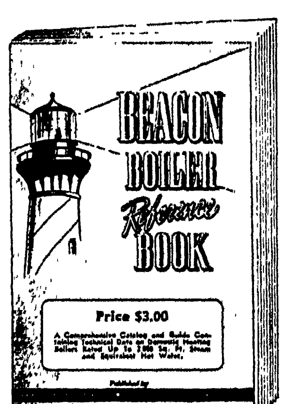
Of the 13,358 oil burner dealers, 7,143 handle air conditioning. We reach them and also fuel oil dealers, heating contractors, accessory jobbers, manufacturers, etc. Member, CCA. Total Average Edition shown in May, 1941 Report; 16,643 Copies.

Subscription price: $3 a year. Issued monthly.

Many fine booklets and reprints on all phases of oil burners, heating and air conditioning are available at small charges.

Advertisers find this paper a profitable advertising medium. We also have a "Direct Mail Service" covering our readers at low cost for advertisers. Our Beacon Trade List Division sells excellent lists of oil burner dealers, furnace dealers, heating contractors, etc., at low prices.

BEACON BOILER REFERENCE BOOK



Contains data on 7116 Heating Boilers and Boiler-Burner Units. Up to 2000 sq ft Steam and Equivalent Hot Water. Covers 195 makes of old, new and obsolete boilers. (405 trade names).

554 pages; 1½ in. thick; 8¼ in. x 5¼ in. Handy for pocket, desk or brief case. Eliminates bulky files.

$3.00 per copy; $3.25 if sent COD.

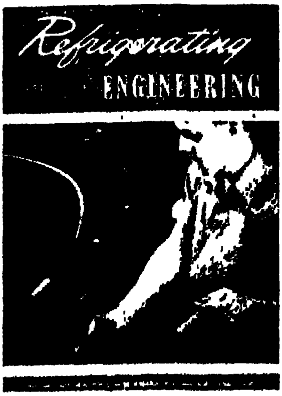
A necessary reference for everybody working with heating boilers and boiler-burner units. Data include ratings, firing rates, combustion chamber measurements, heating surface, floor area, base height, chimney and smoke pipe data, etc.

Also given in the listings are the trade names or numbers of the various series made by the manufactuers; in most cases of older boilers, the date of the catalog and whether it is obsolete; location or disposal of manufacturer; in many cases the source of parts. Shape of boiler, and whether steel, cast iron, etc., also given.

American Society of Refrigerating Engineers

50 West 40th Street, New York, N. Y.

REFRIGERATING ENGINEERING



The most rapidly growing magazine in the refrigeration field

REFRIGERATING ENGINEERING'S

CIRCULATION has increased more than 34 per cent from Jan. 1, 1939, to Sept. 1, 1941. Long acknowledged the most authoritative periodical in the field, it has added steadily to the practical value of its contents, and its number of readers has grown in proportion. A wide variety of material is presented, all from the viewpoint of its usefulness to the reader in his own business. Improved in format and appearance in 1941, this magazine is a *must* for men who keep in touch with all that is new and important in refrigeration and air conditioning.

THE REFRIGERATING DATA BOOK

THE REFRIGERATING DATA BOOK is now an essential tool in the refrigeration and air conditioning industries. Editions have been published in 1932, 1934, 1936 and 1938. The 1940 Edition (Volume II) is *entirely different* from any preceding volume. It consists wholly of practical, how-it is-done chapters on all the known applications of air conditioning and refrigeration. This *Applications Edition* carries information of a scientific and popular nature to the scores of industries using refrigeration processes.

The 1942 Edition of the Data Book, now being compiled, will combine the material included in Volumes I and II, and will give the latest information available in all branches of the refrigeration and air conditioning field. With all contents entirely rewritten and brought up to date, this volume will be off the press the summer of 1942.

APPLICATION DATA BULLETINS

AN *outstanding addition to* REFRIGERATING ENGINEERING since 1939 is the APPLICATION DATA Bulletins which appear regularly in each issue. These bulletins are also available separately at reasonable prices for single copies or quantity orders.

The APPLICATION DATA Bulletins tell precisely how refrigeration is used in various fields, giving examples and specific information on the best practice up to date. Some of these subjects have been covered to date: refrigeration of locker plants, of restaurants, of liquids, of apples and pears, blower coils in refrigeration, humidity in refrigeration, refrigeration service charts, refrigeration for skating rinks, butter and cheese making, milk plants, retail stores, citrus fruits, beer dispensing, retail stores, wine making, ships' stores, load calculations, operation of ammonia machines, how to figure air conditioning, etc.

CODES AND STANDARDS

THE A.S.R.E. further contributes to refrigeration progress by its participation in establishing codes and standards in the industry. Among the recent codes made available are: No. 13 Rating and Testing Air Conditioning Equipment; No. 14 Rating and Testing Mechanical Condensing Units; No. 15 Mechanical Refrigeration Safety Code; No. 16 Rating and Testing Self-Contained Air Conditioning Units; No. 17 Rating and Testing Refrigerant Expansion Valves; No. 18 - Testing Drinking Water Coolers; No. 19 - Standard Water Content Limits for Refrigerating System Parts; No. 20 Testing and Rating Evaporative Condensers; No. 21 Testing and Rating Milk Coolers.

MEMBERSHIP ACTIVITIES

IT is the policy of the A.S.R.E. to treat in its meetings current subjects touching upon all phases of the art of refrigeration. Membership is in two grades with dues from \$7.50 to \$15.00. Sections hold meetings in the following cities: Boston, New York, Philadelphia, Detroit, Chicago, Milwaukee, St. Louis, Los Angeles, Baltimore-Washington, Richmond, Pittsburgh, Cincinnati, Cleveland, Kansas City, and Utica, N. Y. (Central New York State).

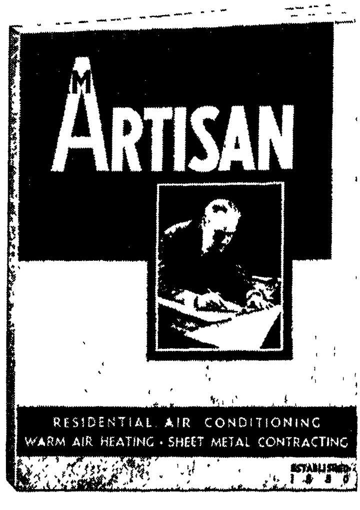
To keep apace with progress in refrigeration and air conditioning, read the publications and follow the activities of THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS, 50 West 40th St., New York, N. Y.

American Artisan

Published by

KEENEY PUBLISHING COMPANY

6 North Michigan Avenue, **Chicago, Ill.**



AMERICAN ARTISAN, now in its 63rd year of publication, covers the field of warm air heating, residential air conditioning, and sheet metal contracting. A special section of each issue has been devoted to air conditioning since 1932, when it first became apparent that air conditioning for homes was to be along the lines of the central, forced warm air heating system.

Its readers are warm air heating and sheet metal contractors, dealers, jobbers and manufacturers, and also architects, engineers, and public utility companies who take it for its thorough coverage of air conditioning for the home field.

To answer the industry's need for a dependable guide to equipment purchases, it publishes in each January issue a complete and up-to-the-minute directory of warm air heating, air conditioning and sheet metal products and equipment. This directory lists all products used in the field, their trade names, and the full names and addresses of all manufacturers. It is used by readers as a buying reference throughout the year.

Almost from the day interest in residential air conditioning began to develop, the advantages of the warm air type of heating system, with its duct distribution of air, were plain to see. It was adapted to all air conditioning factors, either through a self-contained central unit or through a central furnace to which could be added step-by-step or as a whole, fan, washer, humidifier, filters, controls, cooling, and automatic firing.

Today, as a result of this ready adaptability as well as economy, tens of thousands of homes have winter air conditioning—supplied through forced warm air heating with air cleaning and humidification. Cooling apparatus can be attached to these systems readily whenever complete, year-'round air conditioning is desired.

This trend in residential air conditioning has placed a premium on air handling knowledge, and has brought to the fore the one man experienced in "treating" air at a central place and getting it properly distributed—the warm air heating and sheet metal contractor. The warm air heating industry has, furthermore, undertaken and made notable progress toward the solution of the many new engineering problems involved. All this has helped to put warm air heating in the center of residential air conditioning.

In aiding to develop this trend and assist in the solution of new problems, AMERICAN ARTISAN has provided a service to its field which has made it the recognized authority on residential air conditioning practice.

To manufacturers whose products are used in residential air conditioning, AMERICAN ARTISAN offers full coverage of the leading buying factors. Such manufacturers are invited to write for complete information about this expanding market.

AMERICAN ARTISAN is published monthly. It is a member of the Audit Bureau of Circulations and Associated Business Papers.

Subscription rates—$2.00 per year, $3.00 for two years in U. S., Canada, Mexico, Central and South America. Foreign $4.00 per year.

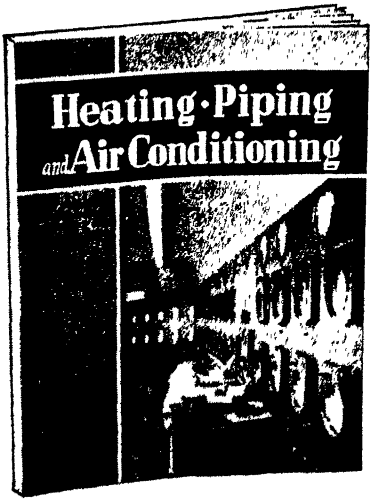
Advertising rates furnished upon request.

Heating, Piping and Air Conditioning

Published by

KEENEY PUBLISHING COMPANY

6 North Michigan Avenue, **Chicago, Ill.**



HEATING, PIPING AND AIR CONDITIONING is the publication which carries in each issue the official JOURNAL OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in addition to its own regular editorial section.

Its field is that of industry and large buildings. Editorially, it gives specialized attention to the design, installation, operation, and maintenance of heating, piping, and air conditioning systems in such plants and buildings.

In addition, there is published in each January issue a complete Directory of Commercial and Industrial Heating, Piping and Air Conditioning Equipment, which lists all products used in the field, their trade names, and the full names and addresses of all manufacturers. This directory has been established as the industry's buying and specifying guide, and is consulted by readers throughout the year, whenever equipment purchases are up for consideration.

H. P. & A. C. is read by consulting engineers and architects . . . contractors . . . and engineers in charge of heating, piping, and air conditioning in industrial plants, large commercial and public buildings, federal, state, and city governments, school boards and public utilities. Among its subscribers are numbered all members of the A.S.H.V.E., who represent about 30 per cent of its total circulation.

Such a coverage means, for the advertiser, consideration at all points in the selling of a heating, piping, or air conditioning product . . . consideration in the selection of a product during the preparation of plans and specifications; consideration in the actual purchase of a product for installation; consideration in the year 'round buying of a product for operating and maintenance requirements.

It has been evident for some time that the air conditioning field is made up of two distinct markets: (1) Industrial and Commercial; (2) Residential.

These two markets are different in equipment used; different in engineering problems involved, different in engineering, distributing, and consuming personnel . . . require, therefore, different selling jobs.

To sell the industrial and large building field for air conditioning, the manufacturer must win acceptance from the engineers who design, specify, install, operate, and select the system to meet the particular requirements of the plant or building. The system may be central, unit, or "split," but it is these engineers who are the influencing or purchasing factors.

It is to such groups that HEATING, PIPING AND AIR CONDITIONING editorially caters · exclusively in the industrial and large building field. Without waste, the manufacturer of air conditioning products and accessory equipment, such as motors, drives, controls, etc., can reach through its pages those from whom he is seeking the necessary engineering acceptance.

Manufacturers interested in this field can obtain complete information by writing to the address given above.

HEATING, PIPING AND AIR CONDITIONING is a member of the Audit Bureau of Circulations and Associated Business Papers.

Subscription rates $2.00 per year; $3.00 for two years in U. S., Canada, Mexico, Central and South America. Foreign, $4.00 per year.

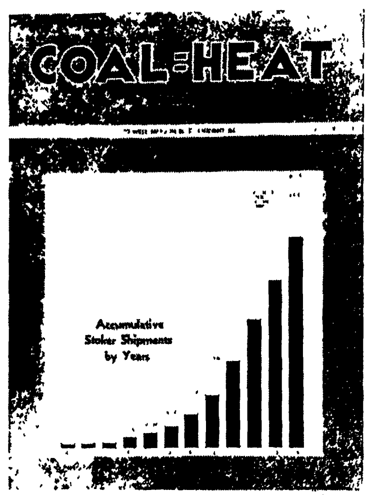
Advertising rates furnished upon request.

Coal-Heat

Published at

20 W. Jackson Blvd., **Chicago, Illinois**

FOR information on the sale and use of stokers, coal and coal heating equipment, you can turn to COAL-HEAT with complete confidence. Here is a magazine that appeals to every man concerned with the marketing and utilization of solid fuel and coal-burning equipment. Having long since recognized the extreme importance of properly designed and efficiently operated equipment to the successful use of coal, and therefore to the welfare of the coal industry, COAL-HEAT constantly emphasizes and reiterates the significance of the "equipment factor" in solid fuel merchandising. It is only natural that COAL-HEAT was the first trade magazine to recognize and promote the small stoker; to introduce many new developments in coal-burning equipment to the coal industry; to support the widespread use of dustless treatment in coal preparation; and to urge the sale of equipment by coal men.



COAL-HEAT has at its disposal an almost unlimited number of sources of authentic information on the topics it covers; its articles are written by the best informed men in the coal, stoker and heating industries. It enjoys quite a following, not only among the most progressive merchants in these industries, but among the coal industry's leading combustion engineers. For a number of years COAL-HEAT has championed the importance of the fuel engineer to the coal and stoker industries, and each year prints many articles *for* and *by* fuel engineers.

COAL-HEAT's fundamental editorial policy is "to further the more satisfactory use and increased sale of coal and modern coal-burning equipment." Therefore it follows that COAL-HEAT actively supports the application of scientific and engineering knowledge to the burning of coal and the use of coal-burning equipment. COAL-HEAT also recognizes and calls attention to the importance of engineering knowledge to the sales of both fuel and equipment. It has directed its editorial program to both the merchandising and utilization sides of the coal, stoker and heating industries, believing that the two are inseparable.

With nearly a million stokers in use today, the importance of COAL-HEAT's field is clearly evident. Each stoker installation involves both a sales and an engineering problem. It has been and is COAL-HEAT's job to supply coal and stoker men with all of the information they need to insure satisfaction for stoker users. The same is true with hand-fired heating plants and all kinds of household and commercial coal heating equipment. COAL-HEAT does not specialize in heavy industrial or central station stokers and coal burning problems.

In addition to providing its readers with an authentic and diversified editorial program, COAL-HEAT also publishes a number of books and booklets, manuals and reprints covering a wide range of subjects of interest to coal, stoker and heating men. These are available at small cost. At the beginning of each year it also publishes a new and revised list of stoker manufacturers, complete in every detail.

Four special issues are printed each year: the *Market Data Issue* for January; the *Spring Stoker Number* for April; the *Annual Merchandising Number* for August; and the *Combustion Number* for November.

Subscription rates—$1.00 a year; $2.00 for three years. Rates apply for both United States and Canada. Foreign rates —$2.00 a year; $4.00 for three years.

Advertising rates and other information will be furnished upon request.

"FAN ENGINEERING"

FOURTH EDITION

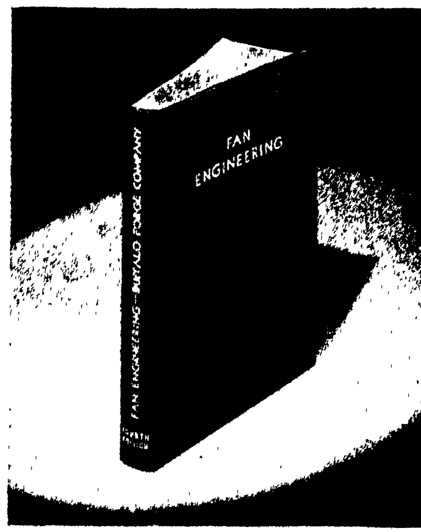
BUFFALO FORGE COMPANY

450 Broadway, **Buffalo, N. Y.**

AN ENGINEER'S HANDBOOK on Air, Its Movement and Distribution in Air Conditioning, Combustion, Conveying and Other Applications Employing Fans.

With thousands of copies of "FAN ENGINEERING" in use, orders for the Fourth Edition of this accurate handbook on air engineering continue to arrive.

Because it contains complete information on the movement, conditioning and application of air in a large variety of services "FAN ENGINEERING" belongs on your desk.



PART I Physics of Air

Properties of Air; Humidity, Heat; Fluid Flow; Proportioning The Flow In Pipes; Air Flow In The Fan; Sound.

PART II Air Conditioning and Other Fan Applications

Heating; Ventilation; Humidifying and Dehumidifying; Cooling and Refrigeration; Drying; Combustion and Mechanical Draft; Dust Elimination; Exhausting and Conveying; Codes, Specifications and Guarantees.

PART III Fans and Air Conditioning Apparatus

Selection of Air Conditioning Equipment; Fan Power Equipment; Fan Details; Multiblade Ventilating Fans; Miscellaneous Fans; Heaters; Coolers; Air Conditioners; Evaporative Condensers; Air Washers.

Copies are still available at cost; $4.00 each postpaid in U. S. A.

There will be no new edition of "FAN ENGINEERING" in 1942.

Send your order with check or money order to "FAN ENGINEERING"

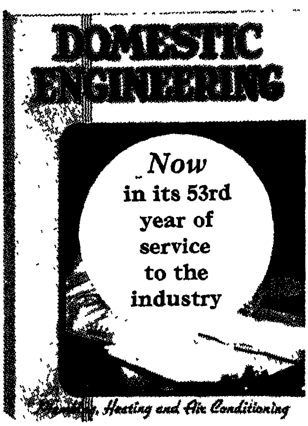
BUFFALO FORGE COMPANY, 450 Broadway, Buffalo, N. Y.

Domestic Engineering Magazine

Published by

DOMESTIC ENGINEERING PUBLICATIONS

1900 Prairie Avenue Chicago, Illinois



Yesterday . . . It was the 35 per cent who did 81 per cent of the business! Today and Tomorrow . . . it's the man with the priorities!

YESTERDAY, emphasis was placed on sales . . . many sales . . . all kinds of sales. At that time the first consideration of every manufacturer in the plumbing, heating, air conditioning and piping industry was the building of a sound position among contractors, wholesalers and engineers who, in turn, had gained for themselves dominant positions in their respective localities—namely the 35 per cent who did 81 per cent of the business.

Yesterday, when manufacturers were concerned primarily with sales, *Domestic Engineering Publications* made available to them a reader audience and acceptance which provided the basis for these sales.

Today, emphasis must be placed on special types of sales . . . sales which carry priority ratings . . . sales which accomplish purposes consistent with the urgent demands of the current war effort.

Domestic Engineering Publications, in keeping abreast of changing conditions (and, in many instances a few steps ahead) are in the position of giving manufacturers a reader audience composed of buyers and specifiers for projects which carry high priority ratings.

Domestic Engineering Publications are able to offer manufacturers in our industry that which they require for meeting today's conditions. This is because of its organization, a portion of which includes men who served this industry with *Domestic Engineering* during the last war and thus are able to recognize and make use of parallels having constructive applications to the problems arising out of the present war . . . because of its close contacts with Washington where the influences affecting today's business are generated . . . because of its vigorous and energetic determination to serve its industry . . . because of its long-range editorial program having as its principal objective the furtherance of our industry as an integral part of modern American life.

TODAY, the plumbing, heating, air conditioning and piping industry is mobilized for the opportunities and responsibilities of a great war effort. *Domestic Engineering* is proud of its part in this mobilization and in the continuing effort now being emphasized in *Domestic Engineering's* "Victory" program. Readers and the industry at large . . . both look to *Domestic Engineering Publications* as the clearing house of ideas and information on which to base their plans.

Manufacturers of plumbing, heating, air conditioning and piping products, regardless of where these products are ultimately installed . . . whether they be for a Navy Yard in Virginia, a bomber factory in California or an armament plant in Illinois or Ohio . . . these manufacturers are urged to make full use of the facilities available to them through *Domestic Engineering* Magazine and *Domestic Engineering Catalog Directory*, in planning their advertising and customer-relations programs for 1942.

For complete data concerning *Domestic Engineering*, the field it serves, advertising rates, circulation, etc., write to Advertising Department, 1900 Prairie Avenue, Chicago, Illinois.

Domestic Engineering Catalog Directory

Published by

DOMESTIC ENGINEERING PUBLICATIONS

1900 Prairie Avenue | Chicago, Illinois

NOW in its twentieth year of service to its industry, the 1942 edition of *Domestic Engineering Catalog Directory* has been designed to meet present day plumbing, heating and air conditioning buying and specifying requirements. A complete, dependable, centralized source of product and technical information, arranged for quick and easy reference, *Domestic Engineering Catalog Directory* has placed buying and specifying on a greatly simplified basis.

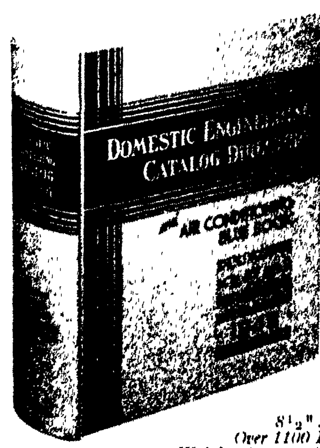
The following are the major sections of *Domestic Engineering Catalog Directory:*

I. MANUFACTURERS' CATALOG SECTION—**Containing the up-to-date buying and specifying information on the products of each co-operating manufacturer.**

II. CLASSIFIED DIRECTORY **Listing virtually every known product used in heating, plumbing and air conditioning and names of manufacturers who make it. Now condensed and greatly simplified to facilitate its use.**

III. TRADE NAME SECTION **This section lists all known trade names of products used in the heating, plumbing and air conditioning field; also gives the names and addresses of the manufacturers.**

IV. MANUFACTURERS' NAMES AND ADDRESSES **This section gives the name, street address and city of every known manufacturer of heating, plumbing and air conditioning equipment.**



8½" x 11"
Over 1100 Pages
Weight, Approx. 10 lbs.

V. TABLES AND RULES **This section consists of hundreds of charts, tables, standard rules and layout diagrams and explanatory material required in the selection, coordination and design of equipment for heating, plumbing and air conditioning installations.**

In constant use, throughout the year, by the leading buyers and specifying engineers in the industry, *Domestic Engineering Catalog Directory* presents to manufacturers a most productive method for the presentation of their product information.

For details of the many services available to manufacturers of heating, plumbing and air conditioning equipment through *Domestic Engineering Catalog Directory*, write Manufacturers' Catalog Service Department, 1900 Prairie Avenue, Chicago, Illinois.

OILHEATING & AIRCONDITIONING fueloil JOURNAL

Published Monthly at

420 Madison Avenue

New York

MARKET: The oilheating market is a closely knit 4-way market—oilburners, heating, airconditioning, and fueloil. The progressive oilheating dealer sells all four—a good oilburner, using good fueloil, firing a good heating or airconditioning system.

From 1919 to 1930, the principal oilheating product sold by burner dealers was the conversion burner. In 1930 the sale of conversion burners represented 77.3 per cent of the dealers' gross income.

By the end of 1940, the average oilheating dealer got only 21.8 per cent of his income from coversion burners. But, beginning in 1932, he added three other major oilheating lines—heating, fueloil and winter airconditioning. 1940 gross dollar volume of the average dealer was divided:

Conversion burner units	21.9 per cent
Heating equipment, including boiler-burner units	17.4 per cent
Fueloil	53.2 per cent
Winter airconditioning, including furnace-burner units	17.5 per cent

In 1940, 22 per cent or 75,227 conversion oilburners were sold with new cast iron or steel boilers. In addition, dealers sold 25,496 boiler-burner units. Total boiler sales by oilheating dealers increased 36 per cent over 1939. These dealers did a winter airconditioning dollar volume in 1940 of $21,445,264.

SERVICES FOR ADVERTISERS

Specific Products Reports.

Key Market Studies.

Merchandising News.

Unit Sale Brand Preference Studies.

Booklets, reprints of special articles.

OILHEATING & AIRCONDITIONING: FUELOIL JOURNAL covers this integrated 4-way market.

It is the oldest paper in the field—established 1922. Editorially, it has consistently fostered every progressive development in the field and it has encouraged the trend to the complete oilheating dealer.

Every issue is carefully balanced editorially to cover the dealers' need for usable information on all four sides of his business.

Heating equipment manufacturers have long known FUELOIL JOURNAL as a powerful sales aid. Its reader interest is unique among trade papers.

CIRCULATION: Like its editorial content, the circulation of FUELOIL JOURNAL is carefully controlled to give complete coverage of this great 4-way market. A detailed breakdown from the latest circulation statement (June, 1941) shows:

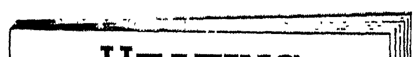
Power oilheating and airconditioning dealers and distributors	12,394	
Key heating contractors, plumbing and heating contractors, and engineers.....	37	
Fueloil distributors, selling fueloil and range oil, and their branches	3,053	
Accessory and heating supply distributors	1,275	
Total dealers and distributors		16,759
Power oilheating and airconditioning manufacturers and their executives.	515	
Accessory manufacturers.	769	
Total manufacturers. .		1,284
Total dealers and manufacturers.		18,043
Per cent of total circulation		99.21
Other miscellaneous ...		128
Grand total. ..		18,171

FUELOIL JOURNAL ciculation covers the oil heating and air conditioning field at the minimum rate per thousand copies. It will pay you well to get full details. Write, wire or telephone.

HEATING AND VENTILATING

The Air Conditioning Monthly

THE INDUSTRIAL PRESS . . . *Publisher*
140-148 Lafayette St. New York, N. Y.



HEATING & VENTILATING is edited for engineers, contractors, and equipment manufacturers who have the final word in the specification, installation, production and maintenance of mechanical equipment for heating, air conditioning and ventilating.

The editorial content is designed to be of practical use to engineers engaged in the design, installation or operating of heating, ventilating or air conditioning equipment, and is prepared under the direction of field-experienced professional engineers. A maximum amount of space is given each month to articles showing how specific problems have been met, authoritative discussions of timely subjects, compilations of useful data, and descriptions of the latest practice, techniques and equipment.

Generally speaking, the emphasis is on practical rather than on technical considerations.

Each month an original Reference Data sheet is included for permanent use in a standard binder (back copies are available).

Special issues or special sections are published from time to time. Each September a comprehensive Buyers Guide (directory of manufacturers) is included with the usual issue. Special Reference Sections are published several times throughout the year on subjects of timely interest.

A special issue on Industrial Air Conditioning was published in January, 1942.

CIRCULATION

HEATING & VENTILATING'S total distribution (May, 1941) 10,827, classified as follows:

Consulting Engineers (419) and Architects (203) Engineers Employed by them (197)	819
Contractors (1,539) and Engineers Employed by Contractors (220)	1,759
Governments and School Boards, and their Engineers	860
Public Utility Group	645
Industrial Firms, their Executives, Engineers and other Employees	1,955
Buildings, Real Estate Management Companies, Their Engineers	623
Manufacturers of Air Conditioning, Heating, Piping and Ventilating Equipment, Their Officials and Employees (727) and Designing Engineers (267)	994
Manufacturers' Agents and Sales-Engineering Firms (163) Sales Engineers and Salesmen (829)	992
Wholesalers (105) and Dealers (321)	426
Educational Institutions, Libraries, Technical Associations	661
Miscellaneous and Unclassified	653
	10,387
Field Staff, Correspondents, Exchanges and Advertising Agencies	440
TOTAL	10,827

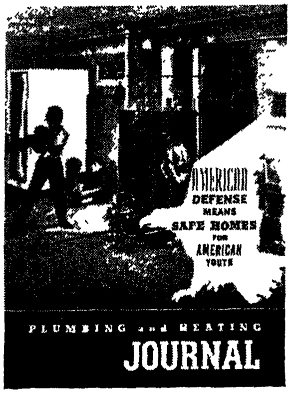
Subscriptions to HEATING & VENTILATING are $2.00 a year.

Plumbing and Heating Journal

Published by

THE ANGUS CO., INC.

515 Madison Ave., **New York City**



PLUMBING and Heating Journal is edited to furnish a well-rounded, efficient service to the men engaged in the plumbing, heating, ventilating and air conditioning fields. It covers both the technical and business phases of their work.

It gives free technical service through a staff of practical engineers; expert merchandising assistance, and its technical and business articles are by men of recognized competence.

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Sheet Metal Worker

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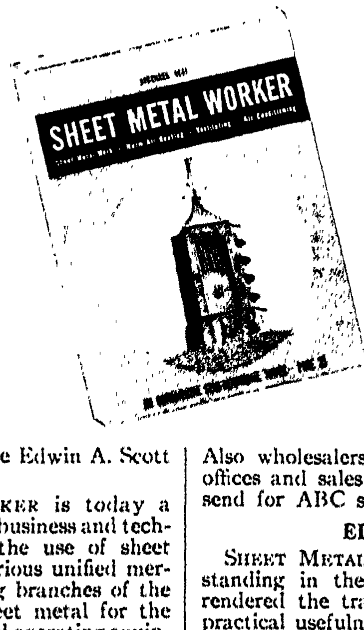

THE January 1942 issue of SHEET METAL WORKER will be its Sixty-Eighth Anniversary and Directory Number. It is the oldest publication in its field and is of vital importance to men interested in sheet metal work—air conditioning—warm-air heating and ventilation. Founded and published to 1909 by David Williams Company; 1909 to 1920 by United Publishers Corp.; since 1920 by the present publisher, the Edwin A. Scott Publishing Co.

SHEET METAL WORKER is today a monthly merchandising, business and technical journal basic to the use of sheet metal. It serves the various unified merchandising and installing branches of the industry, consuming sheet metal for the erection, maintenance and operating equipment of homes and buildings, including central air conditioning equipment, warm-air heating, ventilating, dust and refuse removal, and systems for handling material by air; kitchen and restaurant work; a wide variety of interior and exterior work for commercial, industrial, institutional, and residential buildings.

Subscribers are mainly merchandising contractors purchasing practically all products and equipment which they fabricate, erect or install. Manufacturers, jobbers and distributors also subscribe.

The market has three main divisions:

(1) Equipment for resale in connection with erection or installation work.

(2) Materials for fabrication.

(3) Shop equipment and supplies.

CIRCULATION

SHEET METAL WORKER is a member of the Audit Bureau of Circulations and the Associated Business Papers. It has a uniform distribution, with the greater part of its circulation centered in states showing the greatest industrial activity. Readers of SHEET METAL WORKER are made up of warm-air heating, air conditioning and sheet metal contractors and dealers. Also wholesalers, manufacturers, branch offices and salesmen. For further details send for ABC statement.

EDITORIAL

SHEET METAL WORKER has been outstanding in the editorial service it has rendered the trade and is noted for the practical usefulness of its articles and the timeliness of its editorials. Its editor is a noted author in this field and the author of several well-known books.

SHEET METAL WORKER also publishes books on heating, ventilating, sheet metal work, air conditioning, etc.

The Annual Issue published in January, contains a comprehensive and valuable Directory Section.

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SHEET METAL WORKER has an enviable record of long term advertising and is proud of its long list of regular advertisers.

Because of its intimate contact with this field, SHEET METAL WORKER is well qualified to cooperate with manufacturers in their sales and advertising programs.

Subscription rates—$2.00 per year, U.S., and Mexico. Canada $2.50; Foreign $3.00. Advertising rates on request.

In the Index to Modern Equipment are complete detailed listings of heating, ventilating and air conditioning equipment and materials.

Arranged alphabetically according to names of products are more than 300 items listing not only those products shown in the Catalog Data Section but also many other products made by the manufacturers represented in The Guide.

On pages 1137-1160, under each index heading—Air Cleaning Equipment, Fans, Humidifiers, Ventilators, etc.—will be found, fully cross-indexed, a complete list of manufacturers of any desired products and page numbers in the Catalog Data Section where the products are described. By reference to these index headings, the manufacturers names and the page numbers, any item of equipment or materials may be located quickly.

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AMERICAN SOCIETY *of*
HEATING *and* VENTILATING ENGINEERS

1942

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ENGINEERS OF HUMAN COMFORT

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1941-42

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The real accomplishments of life are usually measured by the service one has rendered to his fellows and the true cultural refinement of mind, the finest sense of personal and professional ethics, factors transcending all material elements in what man calls "success," are developed through association with those of high ideals and cherished ambitions in the same field of activity. THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS offers to him whose work is definitely within its province an opportunity for such association and an opportunity for real service to his profession.

Every man in the heating, ventilating and air conditioning profession needs the Society—

1—Because of the contacts that it brings through national and local meetings.

2—Because of the information supplied by Society Publications.

3—Because of the opportunities that research reveals in new applications for engineering services and equipment.

4—Because of the satisfaction to be derived in contributing to human comfort and well being.

A Candidate must make application on the printed form "Membership Application" which is available at the headquarters office or from Chapter Officers and members. A statement of qualifications and engineering experience is required and four members must act as sponsors except under certain conditions noted in *Article B*-III of the By-Laws.

Initiation Fees for 1942 are: Members and Associate Members $10.00; Junior Members $5.00. The Initiation Fee must accompany application.

For 1942 the annual dues of the Society are: Members and Associate Members $18.00; Junior Members $10.00; and Student Members $3.00. Dues of new members are pro-rated on a quarterly basis.

ARTICLE C-II—MEMBERSHIP

Section 1. Persons connected with the arts and sciences related to heating, ventilating or air conditioning are eligible for admission into the Society.

Section 4. A Member shall be thirty (30) years of age or over and shall be a person of experience in the science of heat transfer in its application to the art of heating, ventilating or air conditioning, and shall have been in active practice of his profession and in responsible charge of important work for five (5) years and shall be qualified to design as well as to direct such engineering work.

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Section 6. An Associate Member shall be twenty-five (25) years of age or over. He need not be an engineer, but must have been so connected with some branch of engineering or the art of heating, ventilating, air conditioning or the industries relating thereto, that he may be considered as qualified to co-operate with heating and ventilating engineers in the advancement of professional knowledge.

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AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS

1942

(Corrected to January 5, 1942)

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(*Asterisk indicates authorship of papers; • indicates address for mail)

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A

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ADAMS, Frank L. (*M* 1939) Htg.-Air Cond. Engr., •Public Service Co. of Colorado, 900 15th St., and 1185 Grape St., Denver, Colo.

ADAMS, Harold E. (*M* 1930) Chief Engr., •The Nash Engineering Co., Wilson Rd., South Norwalk, and Merrill Heights, Norwalk, Conn.

ADAMS, Neil D. (*M* 1929; *A* 1925; *J* 1922) (Council 1938-40) Supt., •Franklin Heating Station, 220 Second Ave. S.W., and 836 Eighth Ave. S.W., Rochester, Minn.

ADDAMS, Homer (*Charter Member*; *Life Member*) (*Presidential Member*) (Pres., 1924; 1st Vice-Pres., 1923; Treas., 1915-22; Council, 1915-25) Pres., Kewanee Boiler Co., Inc. and Fitzgibbons Boiler Co., Inc., 101 Park Ave., New York, N. Y.

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AHLFF, Albert A. (*M* 1923; *A* 1918) Sales Supt., Tuttle & Bailey, Inc., 1600 Arch St., Philadelphia, and •1521 Powder Mill Rd., Upper Darby, Pa.

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AKERMAN, Joseph Reid (*A* 1941; *J* 1937) Process Engr., U. S. Rubber Co., Munitions Div., Des Moines, Iowa, and • 831–15th St., Augusta, Ga.

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ALFSEN, Nikolai (*M* 1933) Engr., Alfsen og Gunderson, Prinsensgt. 2b, Oslo, and • Utsiktsveien 22, Stabekk, Norway.

ALGREN, Axel B.* (*M* 1930) Asst. Prof. Mech. Engrg., Asst. Dir. Engrg. Experiment Station, University of Minnesota, and • 5109–17th Ave. S., Minneapolis, Minn.

ALLAN, William (*A* 1938) Pres., • Allan Engineering Co., 724 E. Mason St., and 2735 N. Farwell Ave., Milwaukee, Wis.

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ANDREWS, W. M. (*M* 1941) Partner, • Lockwood & Andrews, 904 Union National Bank Bldg., and 2254 Shakespeare, Houston, Tex.

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ARENBERG, Milton K. (*A* 1920) Pres., • Robert Barclay, Inc., 122 N. Peoria St., Chicago, and Wildwood Lane, Highland Park, Ill.

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ARKLEY, Lorne M. (*M* 1922) Head of Dept., Mech. Engrg., • Queens University, and 22 Kensington Ave., Kingston, Ont., Canada.

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ARMSTRONG, Clyde C. (*A* 1941) Mgr., • Commercial and Air Cond. Dept., Frigidaire Div., General Motors Sales Corp., 824 Mulberry St., and 803 Douglas Ave., Des Moines, Iowa.

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ARNOLD, Robert S. (*A* 1926; *J* 1922) Owner, Robt. Arnold Sales & Engineering Co., 409 Otis Bldg., Philadelphia, and • Haverford Mansions, Haverford, Pa.

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BACKSTROM, Russell E.* (*A* 1931; *J* 1928) Mgr., • Ind. Sales Dept., Wood Conversion Co., First National Bank Bldg., and 1655 Hillcrest St., St. Paul, Minn.

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BAILEY, Charles F. (*J* 1939) Virginia Engineering Co., N.O.B., Norfolk, and • Windsor, Va.

BAILEY, Edward P. (*M* 1925) Sales Engr., Detroit Stoker Co., 5–125 General Motors Bldg., Detroit, and • 151 Crocker Blvd., Mt. Clemens, Mich.

BAILEY, Frederick A., Jr. (*A* 1939) Prop., • Bailey's, 130 King St., and 70 Warren St., Charleston, S. C.

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BAILEY, W. Mumford (*M* 1930) Managing Dir., British Trane Co., Ltd., Vectair House, Clerkenwell Close, London, E.C. 1, England.

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BAKER, Irving C. (*M* 1921) Vice-Pres. in Charge of Sales, • Chrysler Corp., Airtemp Div., 1119 Leo St., and Box 242, Route 7, Dayton, Ohio.

BAKER, Roland H. (*M* 1928; *A* 1924) Lt. Comdr., • U. S. Naval Reserve, U. S. S. American Legion, c/o Postmaster, New York, N. Y., and Elkins, N. H.

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BALSAM, Charles P. (*M* 1932) Gen. Mgr., National Home Equipment Co., 50 Church St., New York, and • 324 Fourth St., Brooklyn, N. Y.

BANACH, Casimer J. (*J* 1939) Chief Draftsman, Johnson Fan & Blower Corp., 1319 W. Lake St., and • 2346 W. Thomas St., Chicago, Ill.

BANNER, F. L. Dan (*M* 1937) Dist. Mgr., • Minneapolis-Honeywell Regulator Co., 3101 Gilham Plaza, and 615 East 73rd Terrace, Kansas City, Mo.

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BARNES, Arthur R. (*M* 1924) Engr., • U. S. Supply Co., 1315 West 12th St., and 326 East 70th Terrace, Kansas City, Mo.

BARNES, Hugh S. (*J* 1940) 1st Lt., • 99th C. A. (AA), Camp Davis, and 2152 Sherwood Ave., Charlotte, N. C.

BARNES, Lewis L. (*A* 1942; *J* 1937) Air Cond. Engr., Carrier Atlanta Corp., 348 Peachtree St., and • 3995 N. Stratford Rd., Atlanta, Ga.

BARNES, N. W. (*A* 1940) Sales Repr., • The Fulton Sylphon Co., 568 Wrigley Bldg., and 505 N. Michigan Blvd., Chicago, Ill.

BARNES, Raymond W. (*M* 1939) Htg., Vtg. and Air Cond. Contractor, 1208 N. Main Ave., San Antonio, Tex.

BARNES, Walter E. (*M* 1933) Pres., Barnes & Jones, Inc., 128 Brookside Ave., Jamaica Plain (Boston), and • 7 Woodlawn Ave., Wellesley Hills, Mass.

BARNEY, William E. (*M* 1936) Mgr. and Consulting Engr., • Hydraulic-Press Brick Co., Ohio and Michigan Div., South Park, and 4929 East 108th St., Cleveland, Ohio.

BARNEY, William J. (*J* 1941) Hoffman Specialty Co., 1001 York St., and • 2049 N. Meridian St., Indianapolis, Ind.

BARNUM, Willis E., Jr. (*M* 1933; *J* 1930) Sales Engr., York Ice Machinery Corp., 5051 Santa Fe Ave., Los Angeles, Calif.

BARR, George W. (*Life Member*; *M* 1905) (Board of Governors, 1910), Dist. Mgr., • Aerofin Corp., 2030 Land Title Bldg., Philadelphia, and Woods End, Villa Nova, Pa.

BARRETT, Campbell M. (*A* 1941) Canadian Active Service, Newton, Ont., Canada.

BARRY, Patrick I. (*M* 1920) Managing Dir., • M. Barry, Ltd., 4 Marlboro St., and 8 Sidney Park, Cork, Ireland.

BARTELS, Everett M. (*A* 1941; *J* 1939) Supvr. of Mech. Equip., • Independent School Dist., 629 Third St., and 824 E. Sheridan Ave., Des Moines, Iowa.

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BARTH, John W. (*J* 1939) 340 Olive Ave., Long Beach, Calif.

BARTLETT, Amos C. (*M* 1919) Mgr. Htg. and Vtg. Dept., • B. F. Sturtevant Co., Hyde Park, Boston, and 22 Weston Ave., Braintree, Mass.

BARTLETT, C. Edwin (*M* 1922) Pres., • Bartlett & Co., Inc., 3112 North 17th St., and 3111 W. Coulter St., Philadelphia, Pa.

BARTLEY, Henry E. (*M* 1938) Dir. and Works Mgr., Matthews & Yates, Ltd., Cyclone Works, Swinton, and • "The Grange," Hospital Rd., Pendlebury, Lancs., England.

BARTON, Edmund H. (*A* 1939) Htg. Dept. Mgr., • Moosomin Hardware, and Box 308, Moosomin, Sask., Canada.

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BASSETT, James W. (*A* 1938) Sales Engr., • McQuay, Inc., 2832 E. Grand Blvd., Detroit, 123 Merrill St., Birmingham, Mich.

BASTEDO, Albert E. (*M* 1919) Vice-Pres. and Treas., • Burnham Boiler Corp., Irvington, and 55 Burnside Drive, Hastings-on-Hudson, N. Y.

BASTEDO, George R. (*A* 1942; *J* 1937) Htg.-Vtg. Engr., George A. Fuller Co. and Merritt-Chapman & Scott Corp., Quonset Point, R. I., and • 102-36-86th Rd., Richmond Hill, L. I., N. Y.

BATES, John H. (*J* 1941; *S* 1939) Sears, Roebuck & Co., 925 S. Homan St., and • 3210 Arthington St., Chicago, Ill.

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BAUER, Albert E. (*M* 1935) Chief Engr., Stainless & Steel Products Co., 1000 Berry Ave., and • 59 S. Victoria St., St. Paul, Minn.

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BAUMGARDNER, C. M. (*M* 1928) Branch Mgr., • U. S. Radiator Corp., 3254 N. Kilbourn Ave. Chicago, and 416 Cumnor Rd., Kenilworth, Ill.

BAXTER, Julian F., Jr. (*A* 1941) Vice-Pres. and Sales Mgr., • Automatic Coal Burning Corp., 499 Peachtree St., and 197 Brighton Rd. N. E., Atlanta, Ga.

BAXTER, William E. (*A* 1939) Pres., • W. E. Baxter, Ltd., 87 Vitre St. W., Montreal, and 89 51st Ave., Lachine, Que., Canada.

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BAYLES, Robert Wm. (*A* 1940) Mgr. Htg. Div., James Morrison Brass Manufacturing Co., Ltd., 276 King St. W., and • 34 Gormley Ave., Toronto, Ont., Canada.

BAYSE, Harry V. (*Life Member*; *M* 1923) Chairman of Board, • American Furnace Co., 2719-31 Delmar Blvd., and 6959 Hancock Ave., St. Louis, Mo.

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BEAIRD, Benjamin J. (*M* 1939) Engr., • Chas. G. Heyne & Co., 2002 Rothwell, and 1621 Kipling, Houston, Tex.

BEALS, Dowell E. (*M* 1941; *A* 1941; *J* 1940) Chief Field Engr., Plbg.-Htg.-Air Cond., • C. Wallace Plumbing Co., Ft. Worth Aircraft Assembly Plant, and 3790 W. Sixth St., Fort Worth, Tex.

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BEAVERS, G. R. (*M* 1929) Chief Engr., Canadian Blower & Forge Co., Ltd., and • 230 Cameron St. N., Kitchener, Ont., Canada.

BECHTOL, J. J. (*A* 1942; *J* 1937) Engr., L. E. Stevens Co., 626 Broadway, and • 4766 Rapid Run Rd., Cincinnati, Ohio.

BECKER, C. S. (*M* 1939) Branch Mgr., • American Blower Corp., 1011 Majestic Bldg., Milwaukee, and 2532 North 62nd St., Wauwatosa, Wis.

BECKER, Roger K. (*M* 1938) Dept. Mgr., Ohio Valley Hardware & Roofing Co., and • 1017 E. Powell Ave., Evansville, Ind.

BECKER, Walter A. (*M* 1935) Sales Engr., • Grinnell Co., Inc., 4425 S. Western Ave., and 5728 N. Rockwell St., Chicago, Ill.

BECKWITH, F. J. (*M* 1940) Htg. Engr., Crane Co., 1007 W. Bay St., and • R.F.D. 1, Box 150, Jacksonville, Fla.

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BEERY, Clinton E.* (*M* 1913) Pres., • Heat & Fuel Engineering Co., Inc., 1454 Hood Ave., Chicago, Ill.

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BELL, E. Floyd (*M* 1933) Branch Mgr., • Buffalo Forge Co., 2102 Foshay Tower, Minneapolis, and 5224 Oak Lawn Ave., Edina, Minneapolis, Minn.

BELL, Sydney R. (*M* 1939) Principal, • Sydney R. Bell & Associates, Consulting Engrs., 374 Little Collins St., and 83 Queens Rd., Melbourne, Australia.

BELSKY, George A. (*A* 1937) Chief Engr., Justin C. O'Brien Co., 734 Lexington Ave., New York, and • 78 Randolph Rd., Fulton Park, White Plains, N. Y.

BEMAN, Myron C. (*M* 1926) (Council 1934-39) Partner, • Beman & Candee, 374 Delaware Ave., Buffalo, and 262 E. Quaker St., Orchard Park, N. Y.

BENHAM, Colin S. K. (*A* 1940; *J* 1937) Dir., • Benham & Sons, Ltd., 66 Wigmore St., London, W.I., and 5 Constable Close, London, N.W. 11, England.

BENHAM, Ford C., Jr. (*A* 1942; *J* 1938) Sales Engr., • C. H. Ruebeck Co., P.O. Box 141, and 2221 Maple St., Waco, Tex.

BENNETT, Charles A. (*M* 1936) Asst. Mech. Inspector, Dept. of Justice, Bureau of Prisons, Dept. of Justice Bldg., Room 4245, and • 1752 Lamont St. N. W., Washington, D. C.

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BENOIST, LeRoy L. (*M* 1934) Pres., • Benoist Bros. Supply Co., 117 S. Tenth St., and 1500 Main St., Mt. Vernon, Ill.

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BENSON, Merrill L. (*M* 1938) Engr., • American Blower Corp., 666 Marion Rd., and 2087 Lower Chelsea Rd., Columbus, Ohio.

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BERGER, J. L. (*M* 1939) Vice-Pres., • The W. R. Rhoton Co., 5915 Bonna Ave., and 2652 Edgerton Rd., Cleveland, Ohio.

BERMAN, Louis K. (*M* 1908) Pres., • Raisler Corp., 129 Amsterdam Ave., and 285 Central Park West, New York, N. Y.

BERMEL, Alfred H. (*A* 1933; *J* 1928) Chief Engr. and Estimator, • August Arace & Sons, Inc., 642 Third Ave., Elizabeth, and Salem Rd. and Cambridge Drive, Union, N. J.

BERNARD, Edgar L. (*J* 1941; *S* 1940) Heat Engr., • Reynolds Metals Co., 810 E. Franklin St., and 3132 Park Ave., Richmond, Va.

BERNERT, Lawrence A. (*A* 1937) Local Repr., • Warren Webster & Co., 316 Liberty Trust Bldg., and 515 Kensington Ave., Raleigh Court, Roanoke, Va.

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BISHOP, Joseph W. (*M* 1939) Mgr., Air Cond. Div., Toronto Dist., Canadian General Electric Co., Ltd., 214 King St. W., Toronto, and • 62 Highland Crescent, R.R. No. 2, York Mills, Ont., Canada.

BISHOP, M. W. (*A* 1939; *J* 1935) Sales Engr., • American Blower Corp., 228 N. LaSalle St., and 2641 Estes Ave., Chicago, Ill.

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BLACK, Edgar N., III (*M* 1922) Philadelphia Mgr., • Fitzgibbons Boiler Co., Inc., 1717 Sansom St., Philadelphia, and 111 Woodside Rd., Haverford, Pa.

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BLACK, Harry G. (*M* 1917) Prop., • P. Gormly Co., 155 N. Tenth St., and 927 North 65th St., Philadelphia, Pa.

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BLACKHALL, W. R. (*M* 1922) Partner, • McKellar & Blackhall, 1104 Bay St., and 332 Waverly Rd., Toronto, Ont., Canada.

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BLACKMORE, F. H. (*M* 1923) Mgr., Mfg. Dept., • U. S. Radiator Corp., 1056 First National Bank Bldg., Detroit, and 515 Tooting Lane, Birmingham, Mich.

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BLANDING, Robert L. (*M* 1938) Vice-Pres., • Taco Heaters, Inc., 123 South St., and 1385 Smith St., Providence, R. I.

BLANKIN, Merrill F. (*M* 1927; *A* 1926; *J* 1919) (Treas., 1939-41; Council, 1939-41) Pres., • Haynes Selling Co., Inc., S.E. Cor. Ridge Ave. and Spring Garden St., and 528 E. Gates St., Roxboro, Philadelphia, Pa.

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BLAZER, Benjamin V. (*A* 1940) Owner, • M. Blazer & Son, 173 Market St., Passaic, and 48 13th Ave., Paterson, N. J.

BLOOM, Louis (*M* 1935) Partner, • Freeport Plumbing and Heating Engineers, 84-A Broadway, Freeport, L. I., N. Y.

BLUM, Herman, Jr. (*J* 1936) Mech. Engr., C. Wallace Plumbing Co., Box 1209, and • 3903 Bran Mawr, Dallas, Tex.

BLUM, Richard J., Jr. (*A* 1940) Sales Engr., • The Kirk & Blum Manufacturing Co., 2830 Spring Grove Ave., 3909 Vine St., Cincinnati, Ohio.

BLUMENTHAL, M. I. (*M* 1936) Engr. and Instructor, • National Schools, 4000 S. Figueroa St., and 651 West 40th Place, Los Angeles, Calif.

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BODEN, Walter F. (*A* 1937) Branch Mgr., • Modine Manufacturing Co., 424 E. Wells St., Milwaukee, and 606 Milwaukee Ave., South Milwaukee, Wis.

BODINGER, Jacob H. (*M* 1931) Pres., • J. H. Bodinger Co., Inc., 530 Tenth Ave., New York, and Valley Cottage, N. Y.

BODMER, Emmanuel (*M* 1937) Head of Tech. Dept., Etablissements Dieny & Lucas, 223, Boulevard Pereire, and • 20 Rue Leon, Paris, France.

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BOLTON, Reginald P.* (*Honorary Member; Life Member; M* 1897) (*Presidential Member*) (Pres., 1911; 1st Vice-Pres., 1905, 1910; 2nd Vice-Pres., 1903; Board of Governors, 1901, 1905, 1910-13) The R. P. Bolton Co., 116 East 19th St., New York, N. Y.

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BONTHRON, Robert C. (*A* 1935) Syndicate Repr., Air Cond. Dept., Westinghouse Electric & Manufacturing Co., 40 Wall St., New York, and • 44 Ingraham Blvd., Hempstead, L. I., N. Y.

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BOTELHO, Nanto J. (*A* 1937) Chief Engr.-Mgr., • Celbrasil Representacoes, Ltda., Rua General Camara, 64-7°. andar., Rio de Janeiro, Brazil, South America.

BOTTUM, Edward W. (*A* 1942; *J* 1938) Chief Engr., • J. L. Skuttle Co., 12080 Greeley Ave., and 13220 Woodward Ave., Detroit, Mich.

BOUEY, Angus J. (*A* 1937; *J* 1930) Sales Engr., • The B. F. Sturtevant Co., 681 Market St., and 4810 Fulton St., San Francisco, Calif.

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BOWERS, Arthur F. (*A* 1919) Pres., • Industrial Heating & Engineering Co., 828 N. Broadway, and 2832 N. Hackett Ave., Milwaukee, Wis.

BOWLES, Potter (*A* 1928) Pres., • Hoffman Specialty Co., Inc., 1001 York St., Indianapolis, and West Newton, Ind.

BOXALL, Frederick (*M* 1937) Export-Air Cond. & Refrig. Engr., Worthington Pump & Machinery Corp., Worthington Ave., Harrison, and • 36 Kenwood Ave., Verona, N. J.

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BOYDEN, Davis S.* (*Life Member; M* 1909) (*Presidential Member*) (Pres., 1937; 1st Vice-Pres., 1936; 2nd Vice-Pres., 1935; Treas., 1933-34; Council, 1917, 1930-38) Consultant, Box 386, Shirley, Mass.

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BRANDT, Allen D. (*M* 1940) P. A. Sanitary Engr., U. S. Public Health Service, National Institute of Health, and • 137 N. Chelsea Lane, Bethesda, Md.

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BRAUN, Louis T. (*M* 1921) Exec. Secy., • Chicago Master Steamfitters Assn., 228 N. LaSalle St., and 1548 Pratt Blvd., Chicago, Ill.

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BREDESEN, Bernhard P. (*A* 1931) Engr., • Reese & Bredesen, 403 Essex Bldg., and 3623 Knox Ave., N., Minneapolis, Minn.

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BREWER, Frank M. (*J* 1941) Chief Engrg. Draftsman, Norfolk Navy Yard, Portsmouth, and • 1217 Colonial Ave., Apt. B-1, Norfolk, Va.

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BREX, Irving E. (*A* 1939) Asst. Mgr., • Brex & Bieler Div., The Excelsior Steel Furnace Co., 45th St., and First Ave., and 7200 Ridge Blvd., Brooklyn, N. Y.

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BRITTAIN, Alfred, Jr. (*M* 1938) Engr., Weathermakers (Canada), Ltd., 593 Adelaide St., and • 138 Wheeler Ave., Toronto, Ont., Canada.

BROCHA, John F. (*M* 1936) Buyer Plbg. and Htg., Montgomery Ward & Co., 619 W. Chicago Ave., and • 5475 Hirsch St., Chicago, Ill.

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BROD, Bernard M. (*J* 1941) Engr., Suburban Air Conditioning Corp., 10 Brookdale Place, and • 669 N. Terrace Ave., Fleetwood, Mt. Vernon, N. Y.

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BRODNAX, George H., Jr. (*M* 1938) Sales Engr., • Georgia Power Co., Electric Bldg., and 1564 Westwood Ave. S. W., Atlanta, Ga.

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BRONSON, Carlos E.* (*M* 1919) Chief Mech. Engr., • Kewanee Boiler Corp., Kewanee, Ill.

BROOKE, Irving E. (*M* 1938) Consulting Engr., • 189 W. Madison St., Chicago, and 830 Keystone Ave., River Forest, Ill.

BROOM, Benjamin A. (*M* 1914) Sales Promotion Engr., Weil-McLain Co., 641 W. Lake St., and • 1639 Fargo Ave., Chicago, Ill.

BROOME, Joseph H. (*A* 1936) Sales Engr., Minneapolis-Honeywell Regulator Co., 604 Central Ave., East Orange, and • 180 Walnut St., Montclair, N. J.

BROWN, Alfred P. (*M* 1927) Vice-Pres., • Reynolds Corp., 1400 Wabansia Ave., Chicago, and 439 Maple St., Winnetka, Ill.

BROWN, Aubrey I.* (*M* 1923) Prof. of Htg. and Vtg., • Ohio State University, and 169 Richards Rd., Columbus, Ohio.

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BROWN, Foskett* (*M* 1926) Pres., • Gray & Dudley Co., 222 Third Ave. N., and Hillsboro Rd., Nashville, Tenn.

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BROWN, John S., Jr. (*J* 1937) Sales Engr., Frigidaire Div., General Motors Corp., Plant No. 1, Taylor St., and • 428 Hadley Ave., Dayton, Ohio.

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BROWN, Mack D. (*M* 1938; *J* 1936) Engr., • Northup & O'Brien, Archts., 602-03-04 Reynolds Bldg., and 2914 Bon Air Ave., Winston-Salem, N. C.

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BRUNDAGE, F. Ward (*J* 1940) First Lt., Battery E., 423rd C.A. Sep. Bn. A.A., U. S. Army, and • 512 N. Park St., Kalamazoo, Mich.

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BURR, Griffith C. (*M* 1937) Mech. Engr., Carr & Greiner, Marine Base, and •823 S. Lumina Ave., Wrighsville Beach, N. C.

BURRITT, Charles G. (*A* 1916) Mgr., Minneapolis Office, •Johnson Service Co., 922 Second Ave. S., and Leamington Hotel, Minneapolis, Minn.

BURRITT, Edward E., Jr. (*A* 1941) Sales Engr., General Electric Co., Plymouth Bldg., Room 806, and •5415 Fremont Ave. S., Minneapolis, Minn.

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BUSSE, Herbert (*M* 1938) Chief Engr., Fisher Bldg. Div., Fisher & Co., 417 Fisher Bldg., and •16760 Greenview Rd., Detroit, Mich.

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C

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CAMPAU, W. R. (*M* 1940) Secy. and Gen. Mgr., •Kendall Heating Co., 1636 N.W. Lovejoy St., and 4418 Northeast 11th, Portland, Ore.

CAMPBELL, Alfred Q., Jr. (*A* 1940; *J* 1933) Lt. of Field Artillery, U. S. Army, 20th F.A. Branch, Fort Benning, Ga.

CAMPBELL, Bowen (*M* 1938) Engr., •Campbell Heating Co., P.O. Box 833, and 2404 East 29th St., Des Moines, Iowa.

CAMPBELL, Everett K.* (*M* 1920) (Council, 1931-33; 1939-41) Pres., •E. K. Campbell Heating Co., 2441-3-5 Charlotte St., and 3717 Harrison, Kansas City, Mo.

CAMPBELL, George Summers (*A* 1941; *J* 1937) Consulting Engr., •Geo. S. Campbell, Mech. Engr., 1018 Cotton States Bldg., and 1100-17th Ave. S., Nashville, Tenn.

CAMPBELL, George W. (*J* 1939) First Lt., U. S. Army Air Corps., Savannah Army Air Base, 36 Air Base Squadron, Savannah, Ga., and •325 A St. S.E., Washington, D. C.

CAMPBELL, Robert E. (*M* 1941; *A* 1940; *J* 1934) Chief Engr., General Cooling & Heating Corp., 120 E. Forsyth St., and •1521 Catherine Court, Jacksonville, Fla.

CAMPBELL, Roger P. (*J* 1939) Secy., •E. K. Campbell Heating Co., 2445 Charlotte St., Kansas City, Mo., and 921 Caroly Ave., Nashville, Tenn.

CAMPBELL, Thomas F. (*M* 1928) •T. F. Campbell Co., 1013 Penn Ave., and 101 Evernia Dr., Wilkinsburg, Pa.

CANDEE, Bertram C. (*M* 1933) Partner, Beman & Candee, Cons. Engrs., 374 Delaware Ave., Buffalo, and •19 Tremont Ave., Kenmore, N. Y.

CAPLE, Ira (*J* 1941; *S* 1938) Engr., •Super Radiator Corp., and 715 University Ave. S.E., Minneapolis, Minn.

CARBONE, James H. (*M* 1937) Htg.-Vtg. Inspector, City of New York, New York, and •121-13-198th St., St. Albans, L. I., N. Y.

CAREY, Paul C. (*M* 1930) Member of Firm, •Runyon & Carey, Cons. Engrs., 33 Fulton St., Newark, and 31 Claremont Drive, Maplewood, N. J.

CARLE, William E. (*M* 1920) Pres., •Carle-Boehling Co., Inc., 1641 W. Broad St., and 4015 W. Franklin St., Richmond, Va.

CARLOCK, Marion F. (*M* 1936) Dist. Repr., American Foundry & Furnace Co., and •7008 Amherst, University City, Mo.

CARLSON, C. O. (*A* 1937) Owner, •C. O. Carlson Heating Co., 1627 Washington Ave. N., and 3526 Humboldt Ave. N., Minneapolis, Minn.

CARLSON, Everett E. (*M* 1932; *A* 1929) Branch Mgr., •The Powers Regulator Co., 2726 Locust St., and 6675 Washington Ave., St. Louis, Mo.

CARNAHAN, John H. (*A* 1940; *J* 1937) Sales Dept., •Oklahoma Gas & Electric Co., P.O. Box 1498, and 3116 Northwest 26th St., Oklahoma City, Okla.

CARNEY, Edward J. (*A* 1939) Partner, •John C. Kohler Co., 554 North 16th St., and 1020 North 64th St., Philadelphia, Pa.

CARON, Hector (*A* 1938) Mgr., Hector Caron, 324 Lincoln Highway, and •421 S. Third St., Rochelle, Ill.

CARPENTER, Randolph H. (*M* 1921) (Council, 1930-35) Mgr., New York Office, •Nash Engineering Co., Graybar Bldg., 420 Lexington Ave., New York, and 20 Jefferson Ave., White Plains, N. Y.

CARR, Maurice L.* (*M* 1931) Dir. of Research, •Pittsburgh Testing Laboratory, Stevenson and Locust Sts., and Roosevelt Hotel, Pittsburgh, Pa.

CARRIER, Earl G. (*M* 1936; *J* 1929) Branch Mgr., Carrier Corp., 704 Statler Bldg., Boston, and •68 High St., Winchester, Mass.

CARRIER, Willis H.* (*M* 1913) (*Presidential Member*) (Pres., 1931; 1st Vice-Pres., 1930; 2nd Vice-Pres., 1929; Council, 1923-32) Chairman of the Board, • Carrier Corp., 302 S. Geddes St., and 2570 Valley Drive, Syracuse, N. Y.

CARROLL, Daniel E. (*A* 1941) Pres., Carroll Sheet Metal Works, Inc., 46-10-70th St., and • 37-22-68th St., Woodside, L. I., N. Y.

CARROLL, Edgar E. (*A* 1939) Owner, • Kleenair Furnace Co., 5329 N.E. Sandy Blvd., and 2434 Northeast 43rd Ave., Portland, Ore.

CARROLL, William M. (*J* 1938) Sales Engr., • Tom Dolan Heating Co., 614 W. Grand, and 908 East Drive, Oklahoma City, Okla.

CARTER, Alexander W. (*M* 1940; *J* 1936) • Monarch Brass Manufacturing Co., Ltd., 71 Browns Ave., and 117 Elmer Ave., Toronto, Ont., Canada.

CARTER, Doctor (*M* 1934) Consulting Engr., 50 Nevill Rd., Hove, Sussex, England.

CARTER, John H.* (*M* 1936) Mgr., Refrig. Dept., • Kupferle-Hicks Heating Co., 3974 Delmar Blvd., St. Louis, and 504 Tuxedo Blvd., Webster Groves, Mo.

CARY, Edward B. (*M* 1935) Lt. Comdr. (CEC) U.S.N.R., • U. S. Naval Training Station, Great Lakes, and 412 Douglas Ave., Waukegan, Ill.

CASE, Delbert V. (*M* 1937) Engr., Edw. W. Lochman Co., 1421 Cherry St., Kansas City, Mo., and c/o Fisher Memphis Aircraft Div., G.M.C., Memphis, and • 1973 Poplar Ave., Apt. 5, Memphis, Tenn.

CASE, Walter G. (*A* 1930) Mgr., Ideal Boilers & Radiators, Ltd., Ideal House, Great Marlborough St., London, W.1, and • 66 The Ridgeway, Kenton, Harrow, Middlesex, England.

CASEY, Byron L. (*M* 1921) Sales Engr., • Ilg Electric Ventilating Co., 222 N. LaSalle St., Chicago, and 404 Vine Ave., Park Ridge, Ill.

CASKEY, Luther H., Jr. (*J* 1941; *S* 1938) 2nd Lt., Personnel Officer, 38th Engineers, Fort Jackson, S. C., and • 513 N. Queen St., Martinsburg, W. Va.

CASSELL, John D.* (*Life Member*; *M* 1913) (Council 1930-35) Retired, 2008 Walnut St., Philadelphia, Pa.

CASSELL, William L. (*M* 1936) Owner, • William L. Cassell, Mech. Engr., 912 Baltimore Ave., Kansas City, and R.F.D. No. 6, Independence, Mo.

CHALMERS, Charles H. (*M* 1925) Gen. Mgr., • Chalmers Oil Burner Co., 318 First Ave. N., and 523 Seventh St. S.E., Minneapolis, Minn.

CHAMBERS, Fred W. (*M* 1936) Pres., • F. W. Chambers & Co., Ltd., 96 Bloor St. W., and 55 Glengowan Rd., Toronto, Ont., Canada.

CHAMPLIN, Robert C. (*A* 1938) Mgr., Air Cond. Engrg. Dept., • Timken Silent Automatic Div., 100-400 Clark Ave., and 13640 Mendota Ave., Detroit, Mich.

CHAPIN, C. Graham (*M* 1933) Treas., • Hopson & Chapin Manufacturing Co., 231 State St., and 66 Faire Harbour Place, New London, Conn.

CHAPIN, Harvey G. (*M* 1935) Sales Engr., • Westerlin & Campbell Co., 1113 Cornelia Ave., and 8352 Maryland Ave., Chicago, Ill.

CHAPMAN, D. Bascom (*M* 1941) Dist. Sales Mgr., • Clarage Fan Co., 179 Whitehall St., and 2119 McKinley Rd. N.W., Atlanta, Ga.

CHAPMAN, William A., Jr. (*M* 1936) Product Development & Application Dept., • Frigidaire Div., General Motors Sales Corp., 300 Taylor St., and 574 Daytona Pkwy., Dayton, Ohio.

CHARLES, Paul L. (*M* 1938) Mgr., • Walsh & Charles, 406 Tribune Bldg., and 145 Ash St., Winnipeg, Man., Canada.

CHASE, Arthur M., Jr. (*M* 1938) Sales Engr., • York Ice Machinery Corp., Box 359, 2201 Texas Ave., and 3333 Ozark St., Houston, Tex.

CHASE, Chauncey L. (*M* 1931) 222 Chapel Rd., Manhasset, L. I., N. Y.

CHASE, L. Richard (*M* 1938; *J* 1931) Vice-Pres. and Gen. Mgr., • Transport Clearing House, Inc., 111 W. Jackson Blvd., Chicago, and 420 Leonard, Park Ridge, Ill.

CHASE, Peter S. (*A* 1940) Owner, • Chase Co., 936 Oak St., and 1167 Ferry St., Eugene, Ore.

CHASE, Roger E. (*A* 1939) Pres., • R. E. Chase & Co., Inc., Tacoma Bldg., and 117 N. Tacoma Ave., Tacoma, Wash.

CHASE, R. E., Jr. (*J* 1941) Branch Mgr., • R. E. Chase & Co., 506 Railway Exchange Bldg., and 831 S.W. Sixth Ave., Portland, Ore.

CHEESEMAN, Evans W. (*J* 1937; *S* 1934) Lt., Personnel Adjutant, • Personnel Office, 6th Engrs. Training Group, E.R.T.C., No. 1902, Ft. Leonard Wood, Mo., and 1503 Willow St., Coffeyville, Kan.

CHENEVERT, J. Georges (*M* 1938) Consulting Engr., • Arthur Surveyer & Co., Room 1203, 1010 St. Catherine St. W., Montreal and 536 Outremont Ave., Outremont, Que., Canada.

CHERNE, Realto E. (*M* 1938; *J* 1929) Dist. Chief Engr., Carrier Corp., South Geddes St., and • 314 Strathmore Drive, Syracuse, N. Y.

CHERRY, Lester A.* (*M* 1921) Consulting Engr., • Cherry, Cushing and Preble, Cons. Engrs., 271 Delaware Ave., Buffalo, and 151 Euclid Ave., Kenmore, N. Y.

CHESTER, Frank L.* (*A* 1940) Mgr., • W. G. Chester & Son, 179 Bannatyne Ave., and 219 Kingston Row, Winnipeg, Man., Canada.

CHESTER, Thomas* (*M* 1917) Consulting Engr., • 230 Fifth Ave., New York, N. Y.

CHEYNEY, Charles C. (*A* 1913) Asst. Sales Mgr., • Buffalo Forge Co., 490 Broadway, and 255 Lincoln Pkwy., Buffalo, N. Y.

CHILDS, Lewis A. (*M* 1938) Dist. Sales Mgr., • Clarage Fan Co., 520 Commercial Trust Bldg., Philadelphia, and 330 Harrison Ave., Glenside, Pa.

CHRISTENSON, Harry (*A* 1931) Co-Partner, Hunter-Prell Co., 13-19 E. Jackson St., Battle Creek, Mich.

CHRISTIERSON, Carl A. (*A* 1939; *J* 1937) Mgr., • Carrier Engineering S. A., Ltd., Box 2421, and 407 Buckingham Court, 91 Smith St., Durban, South Africa.

CHRISTMAN, William F. (*A* 1931) Engr., • Kroeschell Engineering Co., 215 W. Ontario St., and 6551 N. Maplewood Ave., Chicago, Ill.

CHRISTOPHERSEN, Andrew E. (*M* 1935) Engr.-Custodian, • Board of Education, Spalding School, 1628 Washington Blvd., and 2923 N. Kilpatrick Ave., Chicago, Ill.

CHURCH, H. J. (*M* 1922) Mgr., • Darling Brothers, Ltd., 137 Wellington St. W., Toronto, and 358 Main St. N., Weston, Ont., Canada.

CITRON, Daniel J. (*S* 1938) 2055 Ryer Ave., New York, N. Y.

CLAR, Robert, Jr. (*A* 1938) Sales Engr., • U. S. Radiator Corp., 127 Campbell Ave., Detroit, and Lee Crest Apts., 6898 Livernois, Rochester, Mich.

CLARE, Fulton W. (*M* 1927) • 935 Plymouth Rd. N.E., Atlanta, Ga.

CLARK, Albert C. (*A* 1939) Engr., Mueller Furnace Sales Co., 4069 N.E. Union Ave., and • 3742 N.E. 68th Ave., Portland, Ore.

CLARK, E. Harold (*M* 1922) Mfrs. Agent, 600 Michigan Theatre Bldg., and • 2539 Lakewood, Detroit, Mich.

CLARK, Lynn W. (*A* 1938) Engr. and Sales, • Hall-Neal Furnace Co., 1324 N. Capitol Ave., and 737 West 32nd St., Indianapolis, Ind.

CLARK, Robert L. (*A* 1918) Pres., The Clark Asbestos Co., 1893 East 55th St., Cleveland, and • 927 Caledonia Ave., Cleveland Heights, Ohio.

CLARKE, John H. (*A* 1941) Assoc. Marine Engr., U. S. Maritime Commission, Washington, D. C., and • 882 N. Harrison, Arlington, Va.

CLARKSON, John R. (*J* 1941; *S* 1938) 2832 Burd, St. Louis, Mo.

CLAUSEN, Arnold H. (*M* 1939) Owner and Mgr., • Clausen Engineering Co., 307 Wall St., and 404 E. Howell St., Seattle, Wash.

CLAY, Wharton (*M* 1939; *A* 1938) Secy., • National Mineral Wool Assn., 1270 Sixth Ave., New York, and 127 S. Broadway, Nyack, N. Y.

CLEGG, Carl (*M* 1922) Dist. Mgr., • American Blower Corp., 311 Mutual Bldg., and 3513 Gillham Rd., Kansas City, Mo.

CLEMENS, Joseph D. (*J* 1942; *S* 1940) Olds Motor Works, and • 826 Townsend St., Lansing, Mich.
CLIFTON, John A. (*A* 1938) Mgr., • Renown Plumbing Supplies, Ltd., 235 Parliament St., and 369 Belsize Drive, Toronto, Ont., Canada.
CLO, Harry E. (*J* 1939) Sales Engr., • American Air Filter Co., Inc., 228 N. LaSalle St., and 10565 S. Hale Ave., Chicago, Ill.
CLOSE, Paul D.* (*M* 1928) Tech. Secy., • Insulation Board Institute, 111 W. Washington St., Chicago, and 757 Maclean Ave., Kenilworth, Ill.
CLOSE, Robert (*M* 1938) Chief Air Cond. Engr., National Broadcasting Co., 30 Rockefeller Plaza, New York, N. Y., and • 185 Glenwood Ave., Leonia, N. J.
COCHRAN, L. H. (*M* 1934) Dist. Mgr., • American Blower Corp., 625 Market St., and 130 Camino Del Mar, San Francisco, Calif.
COCKINS, William W. (*A* 1941; *J* 1937) Engr., Scott Co., 243 Minna St., San Francisco, and • 1700 Madera St., Berkeley, Calif.
CODY, Henry C. (*M* 1936) Sales Engr., Pierce Butler Radiator Corp., 15th and Glenwood Ave., and • 7336 North 21st St., Philadelphia, Pa.
COGHLAN, Sherman F. (*A* 1937) Mech. Engr., J. M. Montgomery & Co., 306 W. Third St., Los Angeles, and • 414 Ninth St., Santa Monica, Calif.
COHAGEN, Chandler C. (*M* 1919) Archt., • Chandler & Cohagen, Box 2100, and 235 Avenue G, Billings, Mont.
COHEN, Philip (*M* 1932) Dist. Mgr., • B. F. Sturtevant Co., 401 E. Ohio Gas Bldg., and 7100 Euclid Ave., Suite No. 6, Cleveland, Ohio.
COLBY, John H. (*J* 1939) Sales Engr., • Johnson Service Co., 20 Winchester St., Boston, and 25 Jefferson Rd., Wellesley Hills, Mass.
COLCLOUGH, Otho T. (*A* 1933) Custodian, • American Foreign Service, American Legation, and 399 Hamilton Ave., Ottawa, Ont., Canada.
COLE, C. Boynton (*M* 1940; *J* 1937) Owner, Boynton Cole, Contracting Engr., 1873 Piedmont Rd. N.E., and • 1843 Flagler Ave. N.E., Atlanta, Ga.
COLE, Grant E. (*A* 1925) Vice-Pres. and Gen. Mgr., • Trane Co. of Canada, Ltd., 4 Mowat Ave., and 112 Tyndall Ave., Toronto, Ont., Canada.
COLEMAN, John B. (*M* 1920) Chief Engr., • Grinnell Co., Inc., 275 W. Exchange St., and 237 Cole Ave., Providence, R. I.
COLFORD, John (*A* 1937) Pres., John Colford, Ltd., 2007 Guy St., Montreal, and • 51 Upper Bellevue Ave., Westmount, Que., Canada.
COLLE, S. S. (*A* 1938) Engr., • Air Conditioning Engineering Co., 361 Youville Square, and 4908 Fulton St., Montreal, Que., Canada.
COLLIER, William I. (*M* 1921) Pres., • W. I. Collier & Co., 3414 Duvall Ave., Baltimore, and Ellicott St., Ellicott City, Md.
COLLINS, John F. S., Jr. (*M* 1933) (Council, 1940-41) Secy.-Treas., National District Heating Assn., 827 N. Euclid Ave., Pittsburgh, Pa.
COLLINS, Leo F. (*M* 1941) Chemist, • The Detroit Edison Co., 2000 Second Ave., and 14615 Prevost, Detroit, Mich.
COLMAN, Robert C. (*A* 1940) Vice-Pres., McQuay, Inc., 1600 Broadway N.E., Minneapolis, and • 102 Exeter Place, St. Paul, Minn.
COLMENARES, Gaspar Vizoso (*A* 1938) Vice-Pres. and Gen. Mgr., Refrig. and Air Cond., • Castel-Vizo-S.A., Obrapia 407, and Calle 10 No. 34, Miramar, Havana, Cuba.
COMO, Jack A. (*M* 1939) Mech. Engr., • Independent Plumbing Co., 171 Luckie St. N.W., and 2805 Elliot Circle N.E., Atlanta, Ga.
COMSTOCK, Glen M. (*A* 1926) Sales Engr., L. J. Wing Mfg. Co., 1310 Murdoch Rd. (17), Pittsburgh, Pa.
CONATY, Bernard M. (*M* 1935) Vice-Pres., • American District Steam Co., North Tonawanda, and P.O. Box 342, Eden, N. Y.
CONKLIN, Robert M. (*J* 1940) Engr., 848 South 14th St., Newark, N. J.
CONNELL, Richard F. (*M* 1916) Mgr., Capitol Testing Lab., • U. S. Radiator Corp., 1056 National Bank Bldg., and 2970 Burlingame, Detroit, Mich.
CONNER, Raymond M. (*M* 1931) Dir. Labs., • American Gas Assoc., 1032 East 62nd St., and 2459 Dysart Rd., Cleveland, Ohio.
CONNORS, Edward C. (*A* 1940) Engr.-Custodian, Chicago Board of Education, 5500 Madison St., and • 6556 Ponchartrain Blvd., Chicago, Ill.
CONRAD, Roy (*M* 1935) Sales Engr., Carrier Corp., 1500 S. Santa Fe, Los Angeles, Calif, and • 3416 Colfax "B", Denver, Colo.
CONSTANT, Earl S. (*A* 1942; *J* 1935) Engr., Buffalo Forge Co., 490 Broadway, and • 242 N. Park Ave., Buffalo, N. Y.
CONVERSE, Thornton J. (*M* 1941) Engr., • Office of Douglas Orr, Archt., 96 Grove St., New Haven, and Stony Creek, Conn.
COOK, Benjamin F. (*M* 1920) Prop., Benjamin F. Cook, Consulting Engr., 114 W. Tenth St. Bldg., Kansas City, and • 1720 Overton Ave., Independence, Mo.
COOK, Henry Dale (*A* 1938) Sales Engr., • General Controls Co., 450 E. Ohio St., Chicago, Ill., and 73 E. Tenth St., Holland, Mich.
COOK, Ralph P. (*M* 1930) Asst. Supt., Engrg. and Maintenance Dept. in Charge of Engrg. Div., • Eastman Kodak Co., Kodak Park, and 663 Seneca Parkway, Rochester, N. Y.
COOKE, Thomas C. (*A* 1937) Htg. and Air Cond. Engr., • Tomlinson Co., Inc., 400-402 E. Peabody St., P.O. Box 217, and 1118½ Eighth St., Durham, N. C.
COOLEY, Edgerton C. (*M* 1938) Mfrs. Agent, Owner, • E. C. Cooley Co., 625 Market St., San Francisco, and Box 789 B, Route 1, Los Altos, Calif.
COOMBE, James (*A* 1932) Pres., • William Powell Co., 2525 Spring Grove Ave., and 2363 Grandin Rd., Cincinnati, Ohio.
COON, Thurlow E. (*M* 1916) Pres., • The Coon-DeVisser Co., Inc., 2051 W. Lafayette, and 826 Edison Ave., Detroit, Mich.
COOPER, Dale S. (*M* 1938; *A* 1937) Consulting Engr., 216 E. Cowan Drive, Houston, Tex.
COOPER, Donald E. (*J* 1939) Partner, • D. E. Cooper & Son, 540 Hood St., and 1665 Olive St., Salem, Ore.
COOPER, John W. (*M* 1932; *A* 1925; *J* 1921) Repr., • Buffalo Forge Co., 1598 Arcade Bldg., St. Louis, and 612 Hawbrook Drive, Kirkwood, Mo.
COOPER, William B. (*A* 1942; *J* 1937) Application Engr., Home Htg. and Air Cond. & Commercial Refrig., • Westinghouse Electric & Manufacturing Co., 653 Page Blvd., Springfield, and Bartlett Ave., North Wilbraham, Mass.
COOPERMAN, Edward (*S* 1940) Student, Carnegie Institute of Technology, and • 3120 Avalon St., Pittsburgh, Pa.
COPPERUD, Edmund R. (*A* 1942; *J* 1933) Asst. Mgr., Minneapolis Plumbing Co., 1420 Nicollet Ave., and • 17 West 25th St., Minneapolis, Minn.
CORNWALL, Charles C. (*A* 1942; *J* 1935) Research Engr., The Bahnson Co., 1001 S. Marshall St., and • 473 Carolina Circle, Winston-Salem, N. C.
CORNWALL, George I. (*M* 1919) Sales Engr., Burnham Boiler Corp., 701 Spring St., Elizabeth, N. J.
CORRAO, Joseph (*A* 1936; *J* 1933) Engr., Dept. of Works, Engrg. Dept., City Hall, and • 854-31st Ave., San Francisco, Calif.
CORRIGAN, James A. (*A* 1940; *J* 1935; *S* 1930) Treas., • Corrigan Co., 2501 W. St. Louis Ave., and 7128 Washington Ave., St. Louis, Mo.
COST, George W. (*J* 1939; *S* 1938) Engrg. Draftsman, Westinghouse Electric & Manufacturing Co., Union Bank Bldg., Pittsburgh, and • First and Foster Ave., North Irwin, Pa.
COTT, William B. (*M* 1940) Air Cond. Sales Engr., The Doermann-Roehrer Co., 450 East Pearl St., and • 3001 Bellewood Ave., Cincinnati, Ohio.

COTTRELL, W. H. (*A* 1940) Gen. Mgr., • Home Owners Heating Equipment Co., 2919 Nicollet, Minneapolis, and 3148 Webster Ave., St. Louis Park, Minn.

COVER, E. B. (*M* 1937) Sales Engr., York Ice Machinery Corp., 115 S. 11th St., St. Louis, Mo., and • 3252 Waverly, East St. Louis, Ill.

COVER, Richard R. (*A* 1936) Engr., • Mehring & Hanson Co., 12 H St. N.E., Washington, D. C., and 1914 N. Upton St., Arlington, Va.

COWARD, Charles W. (*M* 1935) Pres., • Coward Engineering Co., 411 Cooper St., Camden, and Cherry Lane, Riverton, N. J.

COX, Harrison F. (*A* 1930) Htg. and Air Cond., 243 Carroll St., Paterson, N. J.

COX, Samuel F. (*M* 1939) Tech. Dir., • Double Glazing Div., Pittsburgh Plate Glass Co., 2200 Grant Bldg., and 6049 Bunkerhill St., Pittsburgh, Pa.

COX, Vernon G. (*A* 1939) Dist. Sales Mgr., • Century Electric Co., 514 Texas Bank Bldg., and 207 Yarmouth St., Dallas, Tex.

COX, William W. (*Life Member*; *M* 1923) Pres. and Mgr., • Heating Service Co., Inc., 326 Columbia St., and 6232-31st Ave. N.E., Seattle, Wash.

CRAIG, Joseph A. (*J* 1940) Sales, • Trane Co., 850 Cromwell Ave., St. Paul, and 829 Eighth St. S., Minneapolis, Minn.

CRANE, Robert S. (*M* 1938) Dist. Engr., • Air Cond. Frigidaire Div., General Motors Sales Corp., 4 Cummins Station, and 3514 Harding Rd., Nashville, Tenn.

CRARY, James O. (*A* 1940) Mgr. Comm. Air Cond. Dept., • Frigidaire Div., General Motors Sales Corp., 4436 Toulouse St., New Orleans, and 107 Ridgewood Drive, Metairie, New Orleans, La.

CRAWFORD, Arthur C. (*A* 1938) Commercial Engr., Potomac Electric Power Co., Tenth and E Sts. N.W., and • 429 Butternut St., N.W. Washington, D. C.

CRAWFORD, John H., Jr. (*A* 1936; *J* 1930) 289 Reynolds Terrace, Orange, N. J.

CRESSY, L. Villere (*M* 1940) Partner, • L. Villere Cressy & Lewis S. Alcus, Consulting Engrs., 916 Union St., and 3217 DeSoto St., New Orleans, La.

CREW, F. D. (*J* 1941) Pres., • The F. D. Crew Co., Schaff Bldg., Philadelphia, and 753 Concord Ave., Drexel Hill, Pa.

CRIQUI, Albert A.* (*M* 1919) Chief Engr., Htg. and Vtg. Dept., Buffalo Forge Co., 490 Broadway, Buffalo, and • 39 St. Johns Ave., Kenmore, N. Y.

CROFT, Huber O. (*M* 1941) Head, Dept. Mech. Engrg., University of Iowa, 107 Engrg. Bldg., Iowa City, Iowa.

CROLEY, Jack G. (*J* 1940) 1st Lt., • 76th Coast Artillery, U. S. Army, Fort Bragg, N. C., and 406 East Temple Ave., College Park, Ga.

CROMBIE, James (*A* 1939) Sales Engr., American Radiator & Standard Sanitary Corp., Elyria, and • 3901 Oak St., Mariemont, Cincinnati, Ohio.

CRONE, Charles E. (*M* 1922) Pres., • Charles E. Crone Co., 1656 N. Ogden Ave., Chicago, and R.R. 2, Prairie View, Ill.

CRONE, Thomas E. (*Life Member*; *M* 1920) Retired, • 164th and Chapin Pkwy., Jamaica, L. I., N. Y.

CRONEY, P. Alfred (*M* 1938) Mech. Engr., Region I, U. S. Housing Authority, Old Interior Bldg., Washington, D. C., and • 112 Granville Drive, Silver Spring, Md.

CROPPER, Robert O. (*M* 1938) Supt. of Refrigeration Plants & Htg. Equip., War Dept., c/o Post Utilities, Fort Knox, and • Vine Grove, Ky.

CROSBY, Edward L. (*M* 1936) Pres., • Henry Adams, Inc., 1015-1023 Calvert Bldg., and 700 Brookwood Rd., Baltimore, Md.

CROSS, Freeman G. (*M* 1936) General Sales Mgr., • Fulton Sylphon Co., and 31 Nokomis Circle, Knoxville, Tenn.

CROSS, Robert C.* (*M* 1937) Sr. Combustion Engr., Testing Laboratories, • Sears, Roebuck & Co., Dept. 817, 925 S. Homan Ave., Chicago, and 334 Northwood Road, Riverside, Ill.

CROSS, Robert E. (*M* 1938; *A* 1931) Dist. Mgr., Minneapolis-Honeywell Regulator Co., 172 Chestnut St., and • 68 Kimberly Ave., Springfield, Mass.

CROUT, Marvin M. (*M* 1939; *A* 1938) Southeastern Mgr., • York Ice Machinery Corp., 412 Houston St., P.O. Box 2210, and 22 Brighton Rd., Atlanta, Ga.

CRUMLEY, Mearl T. (*M* 1941) Mgr., Mech. Engrg. Dept., Robert & Co., and • 1722 Polaris St., Jacksonville, Fla.

CRUMP, Alvin L. (*M* 1937) Sales Engr., • Powers Regulator Co., 2720 Greenview Ave., Chicago, and 2701 Payne St., Evanston, Ill.

CUCCI, Victor J. (*M* 1930) Consulting Engr., • 30 Church St., New York, and 451 55th St., Brooklyn, N. Y.

CULBERT, William P. (*A* 1929) Partner • Culbert-Whitby Co., 2019 Rittenhouse St., Philadelphia, and 929 Alexander Ave., Drexel Hill, Pa.

CULLEN, A. G. (*M* 1939; *A* 1930) Executive, • Cullen Co., 20 L St. S.W., Washington, D. C., and 6826 North 19th Rd., East Falls Church, Va.

CULLIN, William W. (*M* 1938) Chief Engr., Home Insulation Div., Johns-Manville Sales Corp., 22 East 40th St., New York, and • 35 Wildwood Ave., Mt. Vernon, N. Y.

CUMMING, Ford J. (*M* 1930) Contracting Engr., • Beecher-Cumming, Inc., 8 S. Ninth St., Minneapolis, and 120 Interlachen Rd., Hopkins, Minn.

CUMMING, Robert W. (*M* 1928) Sales Executive, Sarco Co., Inc., 475 Fifth Ave., New York, and • 81 Alkamont Ave., Scardsale, N. Y.

CUMMINGS, Carl H. (*A* 1927; *J* 1926) Pres., • Industrial Appliance Co. of New England, 110 Arlington St., Boston, and 41 Edgehill Rd., Chestnut Hill, Mass.

CUMMINGS, G. J. (*M* 1923) Vice-Pres. and Secy., • The Scott Co., 113 Tenth St., and 851 Trestle Glen Rd., Oakland, Calif.

CUMMINGS, Robert J. (*J* 1940) Engr. and Estimator, • Franck & Fric Co., 9334 Kinsman Rd., Cleveland, and 18109 Mapleboro Ave., Bedford, Ohio.

CUMMINS, George H. (*M* 1919) Dist. Mgr., Aerofin Corp., 1116 United Artists Bldg., Detroit, Mich.

CUMMISKEY, Jerome F. (*A* 1940) Sales, Minneapolis-Honeywell Regulator Co., 2405 N. Maryland, and • 4433 N. Cramer St., Milwaukee, Wis.

CUMNOCK, H. (*A* 1938) Pres., • Little Rock Refrigeration Co., 417 W. Capitol Ave., Little Rock, Ark.

CUNNINGHAM, John S. (*A* 1941; *J* 1937; *S* 1935) Chief Engr., Dowagiac Steel Furnace Co., and • 205 Spruce St., Dowagiac, Mich.

CUNNINGHAM, Thomas M. (*M* 1931; *J* 1930) Production Mgr., • Carrier Corp., 7-122 Merchandise Mart, Chicago, Ill.

CURL, Robert S. (*A* 1941) Mgr., Air Cond. Dept., Carle Electric Construction Co., 425 Ohio Bldg., and • 321 Hillwood Drive, Akron, Ohio.

CURLEY, Ellis I. (*A* 1941) U. S. Naval Reserve, • Lehigh University, Bethlehem, and 69 E. Lancaster Ave., Ardmore, Pa.

CURRY, Roger F. (*J* 1940; *S* 1938) Curtiss-Wright Corp., and • Central Y.M.C.A., 45 W. Mohawk, Buffalo, N. Y.

CURTICE, Jean M. (*A* 1936) Dist. Mgr., • Citizens Utilities Co., 15 W. Fourth St., and 906 Lincoln Ave., La Junta, Colo.

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CUSHING, C. F. (*M* 1938) Sales Promotion Mgr., • The Bryant Heater Co., 17825 St. Clair Ave., and 13415 S. Woodland Ave., Cleveland, Ohio.

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DAHLGREN, Gustave E. (*A* 1940) Insulation Mgr., • Thorkelsson Ltd., 1331 Spruce St., and 492 Sprague St., Winnipeg, Man., Canada.

DAHLSTROM, Godfrey A. (*A* 1927) Asst. Engr., Smith, Hinchman & Grylls, Inc. and Toltz, King & Day, Inc., Twin City Ordnance Plant, St. Paul, and • 3721-47th Ave. S., Minneapolis, Minn.

DAITSH, Abe (*J* 1938) Estimating Engr., • Henderson-Smart (Pty.), Ltd., 143-145 Annan House, 86 Commissioner St., and 106/7 Annan House, 86 Commissioner St., Johannesburg, South Africa.

DALY, Robert E. (*M* 1931) • American Radiator & Standard Sanitary Corp., Bessemer Bldg., Pittsburgh, and 271 Kenforest Drive, Mt. Lebanon, Pa.

D'AMBLY A. Ernest (*A* 1924; *J* 1921) Owner, • A. Ernest D'Ambly, 2101 Architects Bldg., and 1835 DeLancey St., Philadelphia, Pa.

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DANIELSON, Wilmot A.* (*M* 1935) Brig. Gen., U. S. Army, c/o Memphis General Depot, Memphis, Tenn.

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DAUBER, Oscar W. (*M* 1937) Consulting Engr., • 224 S. Michigan Ave., Chicago, and 366 Winnetka Ave., Winnetka, Ill.

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DAVEY, Geoffrey I. (*M* 1937) Consulting Engr., • Haskins, Davey & A. G. Gutteridge, 60 Hunter St., and "Netherby," Bangalla St., Warrawee, Sydney, Australia.

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DAVIES, Reginald H. (*M* 1930) Gas. Htg. Adviser, The Colonial Gas Assn., Ltd., 300 Collins St., Melbourne, and • "Harelands," 3 Willsmere Rd., Kew, Victoria, Australia.

DAVIS, Arthur F. (*M* 1934) Pres., • Johnson & Davis Plumbing & Heating Co., 2235 Arapahoe St., and 1901 Ivanhoe St., Denver, Colo.

DAVIS, Bert C. (*Life Member*; *M* 1904) (Council, 1917) Pres.-Treas., American Warming & Ventilating Co., 317 Pennsylvania Ave., Elmira, and • Big Flats, N. Y.

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DAVIS, Donald W., Jr. (*J* 1939) Dist. Mgr., • B. F. Sturtevant Co., 832 Empire Bldg., and 1803 W. Wisconsin Ave., Apt. 28, Milwaukee, Wis.

DAVIS, Edward James (*J* 1938) Sales Engr., Gurney Foundry Co., Ltd., 4 Junction Rd., and • 224 St. Clements Ave., Toronto, Ont., Canada.

DAVIS, George C. (*M* 1939; *A* 1939; *J* 1936) Vice-Pres., • Northern Public Service Corp., Ltd., 307 Power Bldg., and 366 Ash St., Winnipeg, Man., Canada.

DAVIS, George L., Jr. (*A* 1938) Estimator, R. L. Spitzley Heating Co., 1200 W. Fort St., Detroit, and • 1300 Wayburn St., Grosse Pointe Park, Mich.

DAVIS, Joseph (*M* 1927; *A* 1926) Owner-Pres., Engr. and Contractor, • Davis Refrigeration Co., Inc., 70 W. Chippewa St., and 166 Huntington Ave., Buffalo, N. Y.

DAVIS, Keith T. (*M* 1937) Chief Engr., • L. J. Mueller Furnace Co., Milwaukee, and 1500 E. Marion, Shorewood, Wis.

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DAVISON, Robert L. (*M* 1934) Dir. Housing Research, • John B. Pierce Foundation, 40 West 40th St., New York, and Meadow Glen Rd., Fort Salonga, L. I., N. Y.

DAWSON, Eugene F. (*M* 1934) Prof. of Mech. Engrg., • University of Oklahoma, and 229 E. Frank St., Norman, Okla.

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DAY, Irving M. (*A* 1936) Sales Engr., • Binks Manufacturing Co., 718 Mills Bldg., Washington, D. C., and 405 Cumberland Ave., Chevy Chase, Md.

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DEAN, Frank J., Jr. (*A* 1942; *J* 1935; *S* 1934) Pres., Dean-Hagny Corp., 14th & Magee St., and • 6028 Walnut St., Kansas City, Mo.

DEAN, Marshall H. (*J* 1938; *S* 1936) Chief Engr., Hotel President, 14th and Baltimore, and • 1030 West 55th St., Kansas City, Mo.

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DEE, Leo H. (*J* 1937) Engr., L. P. Graner (Consulting Engr.), 40 East 40th St., New York, and • 94 S. Highland Ave., Ossining, N. Y.

DEEVES, Edward W. (*J* 1940) Partner, • Fred Deeves & Sons, 1422A--17th Ave. W., and 2103 33rd St. W., Calgary, Alta., Canada.

DeLAND, Charles W. (*M* 1924; *J* 1923) Secy.-Treas., • C. W. Johnson, Inc., 211 N. Desplaines St., and 2021 Estes Ave., Chicago, Ill.

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DEMETER, Julius (*A* 1939) c/o Julio Donoso D, Calle Catedral 1472, Santiago, Chile.

DEMING, Roy E. (*M* 1941; *A* 1939) Chief Engr., Premier Furnace Co., and •107 Jay St., Dowagiac, Mich.

DEMPSEY, Stephen J. (*A* 1938) Stephen J. Dempsey Co., 79 Harvard St., Battle Creek, Mich.

DENHAM, Howard S. (*M* 1939) Consulting Engr., Cleverdon, Varney & Pike, 46 Cornhill St., Boston, and •80 Dexter St., Malden, Mass.

DENNY, Harold R. (*A* 1934) Eastern Merchandise Mgr., •American Blower Corp., 50 West 40th St., New York, N. Y., and 429 Edgewood Ave., Westfield, N. J.

DEPPMANN, Ray L. (*A* 1937) Owner, •R. L. Deppmann Co., 5853 Hamilton Ave., and 13201 Cloverlawn Ave., Detroit, Mich.

DERER, Bernard (*A* 1940) Designer and Estimator, Sheet Metal Construction, New Brunswick Roofing & Cornice Works, 8-10 Jelin St., New Brunswick, N. J., and •1548 East 31st St., Brooklyn, N. Y.

DeROO, William C. (*A* 1939) Research & Design Engr., •Hart & Cooley Manufacturing Co., and 567 Central Ave., Holland, Mich.

DeSALES, Monteiro, Jr. (*M* 1939) Chief Engr., •Isnard & Co., Rua de Lavradie 67 1°, and Rua Senador Vergueire 193 2°, Rio de Janeiro, Brazil.

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DETERLING, W. C. (*A* 1937) Section Head, Industrial Dept., •General Electric Co., 570 Lexington Ave., New York, and 32 W. Milton St., Freeport, L. I., N. Y.

DEVER, Henry F. (*M* 1936; *A* 1935) Vice-Pres. in charge of Engrg., Minneapolis-Honeywell Regulator Co., and •4609 Edina Blvd., Minneapolis, Minn.

DeVILBISS, Parker T. (*A* 1937) Engr., •R. E. Griffith Theatres, Inc., 7th Floor, Tower Petroleum Bldg., Dallas, Tex., and Hobbs, N. M.

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DEVORE, Angus B. (*A* 1937) Sales Engr., •James A. Messer Co., Inc., 1206 K St. N.W., and 4817-36th St. N.W., Washington, D. C.

DEWEY, Ritchie P. (*M* 1934) Mgr., Aircraft Products Div., •Barber-Colman Co., and 2301 Oxford St., Rockford, Ill.

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DIBBLE, Samuel E.* (*M* 1917) (*Presidential Member*) (Pres., 1925; 1st Vice-Pres., 1924; 2nd Vice-Pres., 1922; Council, 1921-26) Supt., Thomas Ranken Patton School, Elizabethtown, Pa.

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DICKASON, Gray D. (*M* 1938) Pres.-Treas., •Genesee Heating Service, Inc., 950 Mercantile Bldg., and 140 Windemere Rd., Rochester, N. Y.

DICKENS, Lester A. (*A* 1941) Treas., •Dickens Scheufler Burens, Inc., 3959 Mayfield Rd., and 3710 Grosvenor Rd., Cleveland Heights, Ohio.

DICKENSON, Malcolm E. (*M* 1930) Pres. and Gen. Mgr., •Livingston Stoker Co., Ltd., 33 Sanford Ave. S., and 964 Cumberland Ave., Hamilton, Ont., Canada.

DICKEY, Arthur J. (*M* 1921) Vice-Pres. and Gen. Mgr., C. A. Dunham Co., Ltd., 1523 Davenport Rd., and •9 Mossom Place, Toronto, Ont., Canada.

DICKINSON, Robert P., Jr. (*J* 1938) Warehouse Mgr., •Burnham Boiler Corp. of Ohio, 301 Brushton Ave., Pittsburgh, and 219 Meade St., Wilkinsburg, Pa.

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DICKSON, George P. (*M* 1919) Vice-Pres., •B. F. Sturtevant Co. of Canada, Ltd., 137 Wellington St. W., and 790 Eglinton W., Toronto, Ont., Canada.

DICKSON, Robert B. (*M* 1919) Pres., •Kewanee Boiler Corp., and 145 E. Division St., Kewanee, Ill.

DICKSON, Robert W., Jr. (*J* 1938) 1st Lt. Infantry, •132nd Service Unit, Morale Dept., Fort Geo. G. Meade, Md., and 1841 Oliver Bldg., Pittsburgh, Pa.

DIETER, George H. (*M* 1941) Sales Engr., •The Fluor Corp., Ltd., 1501 Mellie Esperson Bldg., and 3104 Lafayette St., Houston, Tex.

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DILLENDER, Eugene A. (*M* 1939) Asst. Engr., United States Engineer Office, and •1152 Menlo Ave., Los Angeles, Calif.

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DISNEY, Melvin A. (*A* 1934) Assoc. Engr., Htg. and Vtg., U. S. Engineer, Galveston, and •612 Tenth Ave. N., Texas City, Tex.

DISTEL, Robert E. (*M* 1941; *A* 1941; *J* 1938) Gen. Mgr., Distel Heating Equipment Co., 404-406 Kalamazoo Plaza, Lansing, and 547 Bailey St., East Lansing, and •P. O. Box 133, Lansing, Mich.

DIVER, M. L. (*M* 1925) Consulting Engr., P. O. Box 1016, San Antonio, Tex.

DIXON, Arthur G. (*M* 1928) Sales Mgr., •Modine Mfg. Co., and 442 Wolff St., Racine, Wis.

DODDS, Forrest F. (*M* 1920) Mgr., •American Radiator & Standard Sanitary Corp., 1023 Grand Ave., and 4600 Mill Creek, Kansas City, Mo.

DODGE, Harry A. (*M* 1936) Elec. Engr., S. H. Kress & Co., 114 Fifth Ave., and •514 West End Ave., New York, N. Y.

DOERING, Frank L. (*M* 1919) Sales Repr., American Radiator & Standard Sanitary Corp., 238 Boston Ave., Lynchburg, Va.

DOLAN, Raymond G. (*M* 1926; *J* 1922) Secy.-Treas., •Tom Dolan Heating Co., Inc., 614-16 W. Grand, and 708 Northwest 40th, Oklahoma City, Okla.

DOLAN, William H. (*A* 1941; *J* 1927) Pres. and Treas., •The Jennison Co., 17 Putnam St., and 65 Highland Ave., Fitchburg, Mass.

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DONNELLY, James A.* (*Life Member*; *M* 1904) (Treas., 1912-14; Board of Governors, 1913; Council, 1914) •Largent, W. Va.

DONNELLY, Russell (*M* 1923) Sales Engr., Nash Engineering Co., Graybar Bldg., 420 Lexington Ave., New York, N. Y.

DONOHOE, Charles F. (*M* 1941) Engr., Central Htg. Dept., • The Detroit Edison Co., 2000 Second Ave., Detroit, and 10065 Lincoln Drive W., Huntington Woods, Mich.

DONOHOE, John B. (*A* 1937; *J* 1935) Engr. and Estimator, • B. F. Donohoe Co., 51 Albany St., Boston, and 23 Primrose St., Roslindale, Mass.

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DORFAN, Morton I. (*M* 1929) Dust Control Specialist, Pangborn Corp., 604 Chamber of Commerce Bldg., and • 1217 Malvern Ave., Pittsburgh, Pa.

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DORSEY, Francis C. (*M* 1920) Pres., • Francis C. Dorsey, Inc., 4520 Schenley Rd., and 212 Gittings Ave., Baltimore, Md.

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DOUGHTY, Charles J. (*M* 1925) Mgr., • C. J. Doughty & Co., 30 Brenan Rd., and 1202 Ave. Joffre, Shanghai, China.

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DOWNES, Nate W. (*M* 1917) (Council, 1928-30) Asst. Supt. in Charge of Bldgs. and Grounds, • School Dist. of Kansas City, Mo., 317 Finance Bldg., and 2119 East 68th St., Kansas City, Mo.

DOWNING, Clarence B. (*A* 1938) Secy.-Treas., • N. B. Downing Co., Jefferson Ave. & Foster St., and Clark Ave., Milford, Del.

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DOWNS, Sewell H. (*M* 1931) (Council, 1936-41) Chief Engr., Clarage Fan Co., and • 1562 Spruce Drive, Kalamazoo, Mich.

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DUGAN, Thomas M. (*M* 1920) Sanitary Htg. Engr., National Tube Co., Fourth Ave. and Locust St., and • 1308 Freemont St., McKeesport, Pa.

DULLE, Willfred L. (*A* 1942; *J* 1936) Asst. Secy., E. E. Souther Iron Co., 1952 Kienlen Ave., St. Louis, and • 2910 Lincoln Ave., Normandy, Mo.

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DUPLANT, Jean L. (*A* 1940) • Westinghouse Electric Co. of India, 294 A Bazargate St., Bombay, India, and 137-43 219th St., Springfield Gardens, L. I., N. Y.

DUTCHER, Harvey S. (*A* 1938) Air & Refrigeration, Inc., 475 Fifth Ave., New York, and • 3420 Clarendon Rd., Brooklyn, N. Y.

DWYER, Thomas F. (*M* 1923) Chief, Htg. and Vtg. Div., • Board of Education, 49 Flatbush Ave. Ext., Brooklyn, and 82 Iris Ave., Floral Park, L. I., N. Y.

DYER, Wilfrid S. (*A* 1939) Partner, • H. W. Dyer & Son, 92 Byron St., Battle Creek, Mich.

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EARLE, Frederic E. (*M* 1937) Owner, • Frederic E. Earle Co., 898 Norman St., Bridgeport, and 1536 Main St., Stratford, Conn.

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EASTWOOD, E. O. (*M* 1921) (1st Vice-Pres., 1941; 2nd Vice-Pres., 1940; Council, 1931-33; 1937-41) Prof. of Mech. Engrg., • University of Washington, and 4702-12th Ave N.E., Seattle, Wash.
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EATON, Byron K. (*M* 1920) Sales Mgr., Delco Heat Div., Omaha Appliance Co., 18 and St. Mary's Ave., and • 817 South 38th St., Omaha, Nebr.
EATON, William G. M. (*M* 1942; *A* 1934) Sales Engr., Pease Foundry Co., Ltd., 227 Victoria St., and • 300 Wellesley St., Toronto, Ont., Canada.
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EDGE, Alfred J. (*M* 1938) Engr. in Charge Htg. and Air Cond., • John F. Reynolds, Consulting Engr., Duval Bldg., Room 316, and 2864 Olga Place, Jacksonville, Fla.
EDWARDS, Arthur W. (*M* 1936) Dist. Mgr., The Trane Co., 626 Broadway, and • 3423 Paxton Ave, Cincinnati, Ohio.
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EHRENZELLER, Adolph (*M* 1924) Consulting Htg. Engr., Owner, • A. Ehrenzeller, 329 Washington St., Dorchester, and 23 Parklawn Rd., West Roxbury, Mass.
EHRLICH, M. William* (*M* 1916) Chief Engr., Commodore Heaters Corp., 11 West 42nd St., New York, N. Y., and • 56 Ridge Rd., Lyndhurst, N. J.
EICHER, Hubert C. (*M* 1922) Chief School Plant Div., Dept. of Public Instruction, and • 207 North 30th St., Harrisburg, Pa.
EISELE, Dudley E. (*A* 1938) Owner, • Eisele Engineering Co., 427 W. College Ave., and 1735 N. Morrison St., Appleton, Wis.
EISS, Robert M. (*M* 1933; *J* 1930) Mech. Engr., Kimberly-Clark Corp., P. O. Box 31, and • Route 1, Adella Beach, Neenah, Wis.
EKINGS, Robert M., Jr. (*M* 1938) Engr., General Electric Co., 5 Lawrence St., Bloomfield, and • Old Chester Rd., Essex Fells, N. J.
EKLUND, Karl G. (*M* 1938) Consulting Engr., • Karl G. Eklunds Ingeniorsbyra A.B., Brunkebergstorg 15, Stockholm, and Storangen, Parkvagen 19, Sweden.
ELIZARDI, Ralph (*J* 1940) Design Engr., Leo S. Weil & Walter B. Moses, Cons. Engrs., 425 S. Peters St., and • 2222 Dublin St., New Orleans, La.
ELLINGWOOD, E. L. (*M* 1909) Consulting Engr., • 124 W. Fourth St., Room 700, Los Angeles, and 210 S. Los Robles Ave., Pasadena, Calif.
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ELLIOTT, Irwin (*A* 1937) Chief Engr., Universal Oven Co., 271 Broadway, New York, and • 103 Penfield Ave., Croton, N. Y.
ELLIOTT, Norton B. (*A* 1934) Sales Engr., • American Blower Corp., 632 Fisher Bldg., Detroit, Mich.
ELLIS, Fred E. (*M* 1923) Sales Mgr., • Imperial Iron Corp., Ltd., 30 Jefferson Ave., and 9 Princeton Rd., Kingsway P.O., Toronto, Ont., Canada.
ELLIS, Frederic R. (*M* 1913) Buerkel & Co., Inc., 18-24 Union Park St., Boston, and • 131 Beacon St., Hyde Park, Mass.
ELLIS, G. P. (*M* 1935) Dist. Mgr., Combustion Engineering Co., Inc., Gwynne Bldg., and • 1118 Delta Ave., Cincinnati, Ohio.
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ENGLISH, Harrold (*M* 1935; *A* 1930) Pres., • English & Lauer, Inc., 1978 S. Los Angeles St., and 515 S. Norton Ave., Los Angeles, Calif.
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ERICSSON, Eric B. (*M* 1933) Engr.-Custodian, Board of Education, and • 6720 Cregier Ave., Chicago, Ill.
ERIKSON, Harald A. (*M* 1939) Vice-Pres., AB. Svenska Flaktfabriken, Kungsgatan 16-18, Stockholm, and • Nockebyvagen 61, Nockeby, Sweden.
ERISMAN, Percival H., Jr. (*M* 1936) Vice-Pres., • Washington Refrigeration Co., 1733-14th St. N.W., Washington, D. C., and 4 Waltonway Rd., Belle Haven, Alexandria, Va.
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ESTEP, L. G. (*M* 1936) Chief Engr., United Wall Paper Factories, Inc., 3330 W. Fillmore St., Chicago, and • 506 S. Kensington Ave., LaGrange, Ill.
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EVANS, Edwin C. (*M* 1919) Mgr., Buffalo Office, • B. F. Sturtevant Co., 602 Jackson Bldg., 220 Delaware Ave., and Mayflower Apts., Apt. 2J, 66 Summer St., Buffalo, N. Y.

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EVEREST, R. Harry (*M* 1935) Engrg. and Sales, Sheldons, Ltd., Galt, and • 235 Waterloo St. S., Preston, Ont., Canada.

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FAGIN, Daniel J. (*M* 1932) Mgr., Sales Engrg. Div., • The Laclede Gas Light Co., 1017 Olive St., and 5846 Lindenwood Ave., St. Louis, Mo.

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FARNES, Bert W. (*A* 1938) Special Repr., Minneapolis-Honeywell Regulator Co., 122 N.E. Broadway, and • 3019 Northeast 26th Ave., Portland, Ore.

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FARRAR, Cecil W. (*M* 1920; *A* 1918) (Treas., 1930; Council, 1930) Vice-Pres., W. A. Case & Son Manufacturing Co., 31 Main St., and • 29 Oakland Place, Buffalo, N. Y.

FARRINGTON, S. Edward (*M* 1940) Designer, • Moody & Hutchison, 1701 Architects Bldg., and 3123 Rawle St., Philadelphia, Pa.

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FELTWELL, Robert H. (*Life Member*; *M* 1905) Htg. Engr., U. S. Radiator Corp., 2321 Fourth St. N.E., and • 1370 Oak St. N.W., Washington, D. C.

FENNER, N. Paul (*A* 1928) Dist. Office Mgr., • Hoffman Specialty Co., 130 N. Wells St., Chicago, and 227 Berteau Ave., Elmhurst, Ill.

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FERDERBER, Murray B., M.D.* (*M* 1938) Fellow of Dept. of Industrial Hygiene, University of Pittsburgh, Medical School, and • 5722 Fifth Ave., Pittsburgh, Pa.

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FIEDLER, Harry W. (*M* 1923) Owner, • Air Conditioning Utilities, 8 West 40th St., New York, and 16 Dobbs Terrace, Scarsdale, N. Y.

FIFE, G. Donald (*M* 1937; *A* 1931; *J* 1929) Air Cond. Engr., Architect of the Capitol, Washington, D. C., and • 4137 N. Henderson Rd., Arlington, Va.

FIGGIS, Thomas G. (*A* 1937; *J* 1936) Tech. Sales Engr., J. & E. Hall, Ltd., Dartford Ironworks, Kent, England.

FINDLEY, Howard N. (*A* 1941) Mgr., Merchandise Sales, • Avery Engineering Co., 1906 Euclid Ave., and 1762 East 89th St., Cleveland, Ohio.

FINERAN, Edward V. (*A* 1940; *J* 1935) 1st Lt. Signal Corps School, U. S. Army, and • Deal Apts., Deal, N. J.

FINNEY, Brandon (*M* 1937) Engr., Southern Heater Co., Compton, and • 721 Via de La Paz, Pacific Palisades, Calif.

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FIRESTONE, Maurice T. (*M* 1939) Mgr. of Dealers Northeastern Dist., • Carrier Corp., 405 Lexington Ave., and 65 Park Terrace East, New York, N. Y.

FISCHER, Frank P. (*A* 1940) Prop., Frank P. Fischer Engineering Co., 412 Dryades St., New Orleans, La.

FISCHER, Lawrence W. (*J* 1937) Plant Mgr., • Anemostat Corp. of America, 1031 New Britain Ave., Hartford, Conn., and 91 Islip Ave., Islip, L. I., N. Y.

FISHER, John T. (*A* 1942; *J* 1936) Project Engr., American Airlines, Inc., N. Y. Municipal Airport, Jackson Heights, and • 168-09 Crocheron Ave., Flushing, L. I., N. Y.

FITTS, Joseph C. (*M* 1930) Secy., Heating, Piping & Air Conditioning Contractors National Assn., 1250 Sixth Ave., New York, N. Y., and • 215 Kenilworth Rd., Ridgewood, N. J.

FITZ, Jean Chandler (*M* 1924) Engr.-Estimator, J. L. Murphy, Inc., 340 East 44th St., New York, and • 405 Webster Ave., New Rochelle, N. Y.

FITZGERALD, Matthew J. (*M* 1934) Pres., Standard Asbestos Manufacturing Co., 820 W. Lake St., Chicago, and • 1117 N. Linden Ave., Oak Park, Ill.

FITZGERALD, William E. (*J* 1936; *S* 1935) Pres.-Gen. Mgr., • Fitzgerald Plumbing & Heating Co., Inc., 939-41 Louisiana Ave., and 1905 Gilbert St., Shreveport, La.

FITZSIMONS, J. P. (*M* 1941; *A* 1940; *J* 1934; *S* 1932) Mgr., Air Cond. Dept., • Trane Co. of Canada, Ltd., 4 Mowat Ave., and 61 Burnaby Blvd., Toronto, Ont., Canada.

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FORSLUND, Oliver A. (*M* 1936) Gen. Mgr., Forslund Pump & Machinery Co., 1717-19 Main St., and • 108th St. and State Line, Kansas City, Mo.

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FOSTER, John G. (*J* 1938) U. S. Air Corps, and • 2635 Sedgwick Ave., New York, N. Y.

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FOULDS, P. A. L. (*M* 1910) Partner, • Hubbard Rickerd & Blakeley, Cons. Engrs., 110 State St., Boston, and 72 Whitin Ave., Point of Pines, Revere, Mass.

FOWLES, Harry H. (*A* 1940; *J* 1934) Htg.-Vtg. Engr., • Carman-Thompson Co., 12-14 Lincoln St., Lewiston, and 4 Haskell St., Auburn, Me.

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FRANKEL, Gilbert S. (*M* 1926) Mgr., Federal and Marine Dept., •Buffalo Forge Co., 512 Woodward Bldg., Washington, D. C., and 5800 Kirkside Drive, Chevy Chase, Md.

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FRENTZEL, Herman G. (*M* 1936) Chief Engr., The Heil Co., and •4363 N. Wildwood Ave., Milwaukee, Wis.

FRIEDLER, Joseph J., Jr. (*M* 1940) Southern Dist. Mgr., •Ilg Electric Ventilating Co., 304 Natchez Bldg., and 3411 Fontainbleau Drive, New Orleans, La.

FRIEDLINE, James M. (*A* 1942; *J* 1937) Engr., The Rath Packing Co., and •326 Western Ave., Apt. 4, Waterloo, Iowa.

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GIFFORD, Robert L. (*Life Member*; *M* 1908) Pres., Illinois Engineering Co., Cor. 21st St. and Racine Ave., Chicago, Ill., and • 1231 S. El Molino Ave., Pasadena, Calif.

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HALE, John F. (*Life Member*; *M* 1902) (*Presidential Member*) (Pres., 1913; 1st Vice-Pres., 1912; Board of Governors, 1908; 1910, 1912-13; Council 1914) Dist. Mgr., •Aerofin Corp., Room 544, 111 W. Washington St., Chicago, and 615 W. Elm Ave., LaGrange, Ill.

HALEY, Harry S.* (*M* 1914) Consulting Engr. and Partner, •Leland & Haley, 58 Sutter St., and 735–21st Ave., San Francisco, Calif.

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HALL, Cortice H. (*M* 1927) Chief Engr., Stoker Div., • Fairbanks, Morse & Co., and 1004 N. Main St., Three Rivers, Mich.

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HALL, John R. (*M* 1937; *J* 1932) Exec. Engr., DeVilbiss Co., and • 2902 W. Central Ave., Toledo, Ohio.

HALL, Mora S. (*M* 1934) • Paul Rosenthal & Associates, 600 Newton Place N.W., Washington, D. C., and 38 and Utah, Brentwood, Md.

HALL, Norman H. (*A* 1939) Supvr. House Htg., • East Ohio Gas Co., East 62nd St., North of St. Clair, Cleveland, and 147 Beachview Rd., Willoughby, Ohio.

HALL, Truman A. (*A* 1940) Field Engr., Dygert Distributing Co., 1 Ionia Ave. S.W., Grand Rapids, Mich.

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HAMIG, Louis L. (*M* 1941; *A* 1940; *J* 1935) Engr., John D. Falvey, Cons. Engr., 316 N. Eighth St., and • 3514 Utah St., St. Louis, Mo.

HAMILTON, Howard S. (*A* 1940) Owner, Air Comfort Co., 1214 N. Astor St., and • 1214 N. Astor St., Milwaukee, Wis.

HAMLET, F. Aylmer (*A* 1936) Branch Mgr., Sales Office, • C. A. Dunham Co., Ltd., Dominion Square Bldg., Room 832, 1010 St. Catherine St., and 3550 Shuter St., Montreal, Que., Canada.

HAMLIN, James B., Jr. (*A* 1937) Htg. Engr., Crane Co., 14 W. Broad St., and • 1530 East 51st St., Savannah, Ga.

HAMPLE, Henry J. (*S* 1939) Student, Carnegie Institute of Technology, and • 4921 Forbes St., Pittsburgh, Pa.

HANBURGER, Fred W. (*M* 1930) Consulting Engr., 252 West 76th St., New York, N. Y.

HANLEIN, Joseph H. (*M* 1937) Vice-Pres., • Wilberding Co., Inc., 1822 Eye St. N.W., and 5420 Connecticut Ave. N.W., Washington, D. C.

HANLEY, Edward V. (*A* 1933) Pres., • S. V. Hanley Co., 1653 N. Farwell Ave., Milwaukee, and 844 E. Birch Ave., Whitefish Bay, Wis.

HANLEY, T. F., Jr. (*M* 1933) Pres., Hanley & Co., 1503 S. Michigan Ave., and • 1640 East 50th St., Chicago, Ill.

HANNIGAN, William (*M* 1940) Bldg. Supt., • Acacia Mutual Life Insurance Co., 51 Louisiana Ave., Washington, D. C., and Route 2, Silver Spring, Md.

HANSEN, John T. (*A* 1941) The Herman Nelson Corp., Moline, Ill., and • 721-34th St., Des Moines, Iowa.

HANSLER, John E. (*M* 1937) 723 Glen Ave., Westfield, N. J.

HANSON, Leon C. (*A* 1918) Secy.-Mgr., • Bjorkman Bros. Co., 712 Tenth St. S., and 4713 Townes Rd., Minneapolis, Minn.

HANSON, Leslie P. (*M* 1937; *A* 1936; *J* 1935; *S* 1933) Engr., U. S. Air Conditioning Corp., 2101 Kennedy N.E., and • 5027 Nokomis Ave. S., Minneapolis, Minn.

HANTHORN, Walter (*J* 1939) Engr., • Kleenair Furnace Co., 5320 N.E. Sandy Blvd., and 2946 Northeast 54th Ave., Portland, Ore.

HARBERGER, G. L. (*A* 1939) Sales Mgr., Peerless Heating Div., The Eastern Foundry Co., Spring & Schaeffer Sts., Boyertown, and • 641 King St., Pottstown, Pa.

HARBIN, Frank, Jr. (*M* 1941) Htg. Engr., • Home Furnace Co., Sixth St. and P. M. Ry., and 181 West 21st St., Holland, Mich.

HARBORDT, Otto E. (*A* 1936) Sales Mgr., • U. S. Supply Co., 1315 West 12th St., and 1620 West 51, Kansas City, Mo.

HARD, Amos L. (*A* 1938) Chief Engr., Thos. Emery & Sons Co., Carew Tower, and • 910 Kreis Lane, Cincinnati, Ohio.

HARDEN, J. Clinton (*M* 1938) 106 Courtland St., Dowagiac, Mich.

HARDING, Edward R. (*M* 1936) N. C. State Sales Engr., • Kewanee Boiler Corp., P.O. Box 536, 1003 Jefferson St. Bldg., Greensboro, and Guilford College, N. C.

HARDING, Louis A.* (*Life Member*; *M* 1911) (*Presidential Member*) (Pres., 1930; 1st Vice-Pres., 1929; 2nd Vice-Pres., 1928; Council, 1922; 1924; 1928-31) 85 Cleveland Ave., Buffalo, N. Y.

HARDY, Frank L. (*A* 1941; *J* 1937) 2619 Arlington Ave. S., Birmingham, Ala.

HARE, W. Almon (*M* 1930) Exec. Partner, Sawyer-Hare Furnace Co., 5736 12th St., Detroit, Mich., and • 833 Kildare Crescent, Windsor, Ont., Canada.

HARMONAY, William L. (*A* 1935) Pres., • Michael Harmonay Corp., 124 Elm St., and 1346 Midland Ave., Yonkers, N. Y.

HARRIGAN, Edward M. (*M* 1915) Gen. Mgr., Harrigan & Reid Co., 1365 Bagley Ave., Detroit, Mich.

HARRIGAN, Edward R. (*M* 1939; *J* 1930) Sales Engr.-Dist. Repr., • General Electric Co., 570 Lexington Ave., New York, N. Y., and 19 Erwin Place, Caldwell, N. J.

HARRINGTON, Elliott* (*M* 1932; *A* 1930) In charge of Induction Sales, Industrial Dept., Building 2, General Electric Co., Schenectady, N. Y.

HARRINGTON, Larry J. (*M* 1941) Pres., • Larry Harrington Co., 633 S.E. Morrison St., and 3530 S.E. Lambert St., Portland, Ore.

HARRIS, Albert M. (*M* 1938) • Harris & Beeman, Inc., 509 E. Third St., and 3207 Denman St., Fort Worth, Tex.

HARRIS, Jesse B. (*M* 1918) Co-Partner, • Rose & Harris Engineers, 416 Essex Bldg., and 3620 Colfax Ave. S., Minneapolis, Minn.

HARROWER, William C. (*A* 1937) Air Cond. Engr., Timken Silent Automatic Div., 100 Clark St., Detroit, and • 12561 Third Ave., Highland Park, Mich.

HART, Harry M.* (*M* 1912) (*Presidential Member*) (Pres., 1916; 1st Vice-Pres., 1915; Council, 1914-17) Pres., • L. H. Prentice Co., 1048 Van Buren St., and 3730 Lakeshore Drive, Chicago, Ill.

HART, Stanley (*M* 1938) Vice-Pres., Tuttle & Bailey, Inc., New Britain, Conn.

HART, Theodore S. (*M* 1938) Engr., • Tuttle & Bailey, Inc., and 530 Lincoln St., New Britain, Conn.

HARTIN, William R., Jr. (*J* 1935) Vice-Pres.-Secy., W. R. Hartin & Son, Inc., 2123 Green St., and • 2744 Trenholm Rd., Columbia, S. C.

HARTMAN, John M. (*M* 1927) Engr., • Kewanee Boiler Corp., and 618 Elliott St., Kewanee, Ill.

HARTON, A. J. (*A* 1935) Sales Engr., St. Joseph Railway, Light Heat & Power Co., 520 Francis St., and • 730 E. Hyde Park Ave., St. Joseph, Mo.

HARTSOOK, Granville S., Jr. (*A* 1939) Owner, • G. S. Hartsook, Jr., Plbg. and Htg. Contr., P.O. Box 361, and 112 E. Fourth St., Front Royal, Va.

HARTWEIN, C. E. (*M* 1933) Supvr., House Heating and Service Div., • St. Louis County Gas Co., 231 W. Lockwood Ave., Webster Groves, and 135 Peeke Ave., Kirkwood, Mo.

HARTWELL, Joseph C. (*M* 1922) Pres. and Treas., • Hartwell Co., Inc., 87 Weybosset St., and 16 Freeman Parkway, Providence, R. I.

HARVEY, Alexander D. (*A* 1928; *J* 1925) Sales Mgr., • Kimberly-Clark Corp., and 819 E. Forest Ave., Neenah, Wis.

HARVEY, Lyle C. (*M* 1928) Pres., The Bryant Heater Co., 17825 St. Clair Ave., Cleveland, Ohio.

HASHAGEN, John B. (*M* 1930) P. O. Box 274, LaPorte, Ind.

HATCH, George (*A* 1941) Pres., Air Conditioning Engineering Co., 1104 Bay St., and • Park Plaza, Toronto, Ont., Canada.

HATCH, O. J. (*A* 1941) Mgr., • Clare Bros. Western Ltd., 179 Notre Dame Ave. E., Winnipeg., and 267 Berry St., St. James, Winnipeg, Man., Canada.

HATTIS, Robert E. (*M* 1926) Consulting Engr., • Board of Trade Bldg., and 1454 W. Fargo Ave., Chicago, Ill.

HAUAN, Merlin J. (*M* 1933) Consulting Engr., 3412–16th S., Seattle, Wash.

HAUCK, Elden L. (*A* 1940; *J* 1936) Mgr., • Hauck Bros., 232 S. Center St., Springfield, and 1010 S. Main St., Dayton, Ohio.

HAUER, Fred (*A* 1938) Owner, • Fred Hauer & Co., 111 N. Water St., and 315 Elmhurst Ave., Peoria, Ill.

HAUS, Irvin J. (*A* 1937; *J* 1935) Engr., Everett Smith Automatic Temperatures, Inc., 789 N. Water St., and • 5410 W. Center St., Milwaukee, Wis.

HAUSMAN, Louis M. (*M* 1935) Pres., L. M. Hausman & Co., P. O. Box 1729, Manila, P. I.

HAUSS, Charles F.* (*Life Member*; *M* 1922) Ball Heights, California, Ky.

HAWES, Harold D. (*J* 1942; *S* 1940) Master Sergeant, U. S. Army, 208th C.A. (AA), Hq. Btry., Quincy, Mass., and • 454 Toilsome Hill Rd , Bridgeport, Conn.

HAWISHER, Harold H. (*A* 1938) Mech. Engr., • Automatic Heating & Engineering Co., 416 N. Main St., and 411 S. Jamison Ave., Lima, Ohio.

HAWK, Joseph K. (*M* 1939; *J* 1936) Sales Engr., • W. A. Case & Son Manufacturing Co., 31 Main St., and 203 Woodward Ave., Buffalo, N. Y.

HAYES, James J. (*M* 1920) Vice-Pres., • Stannard Power Equipment Co., 53 W. Jackson Blvd., and 7443 Jeffrey Ave., Chicago, Ill.

HAYES, Joseph G. (*Life Member*; *M* 1908) Pres. and Engr., • Hayes Brothers, Inc., 236 W. Vermont St., and 2849 N. Capital Ave., Indianapolis, Ind.

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HAYNES, Charles V. (*Life Member*; *M* 1917) (*Presidential Member*) (Pres., 1934; 1st Vice-Pres., 1933; 2nd Vice-Pres., 1932; Council, 1926-29; 1932-35) 263 E. Wynnewood Ave., Merion, Pa.

HAZLEHURST, H. D. (*A* 1939) Southern California Gas Co., 810 S. Flower St., Los Angeles, and • 130 E. Garvey Ave., El Monte, Calif.

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HEALY, Charles T. (*J* 1940) Dist. Mgr., Minneapolis-Honeywell Regulator Co., 713 Maritime Bldg., New Orleans, La.

HEATH, William R. (*M* 1931) Asst. Chief Engr., Buffalo Forge Co., 490 Broadway, and • 119 Wingate Ave., Buffalo, N. Y.

HEBLEY, Henry F. (*M* 1934) Prod. Control Mgr., • Pittsburgh Coal Co., Office No. 3, Library, and 210 Jefferson Drive, Mt. Lebanon, Pittsburgh, Pa.

HECHT, Frank H. (*M* 1930) Sales Engr., • B. F. Sturtevant Co., 2635 Koppers Bldg., and 1467 Barnesdale St., Pittsburgh, Pa.

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HECKEL, Edmund P., Jr. (*J* 1941) Engr., • Buffalo Forge Co., 490 Broadway, and 226 Linwood Ave., Buffalo, N. Y.

HECKLER, Samuel (*J* 1937) Engr., • Westchester Square Plumbing Supply Co., Inc., 4617 White Plains Ave., and 2800 Creston Ave., New York, N. Y.

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HEDGES, H. Berkley (*M* 1919) Mgr., Industrial Sales, • John J. Nesbitt, Inc., Holmesburg, Philadelphia, and 114 Waverly Rd., Wyncote, Pa.

HEDLEY, Park S. (*M* 1923) Mfr. Repr., • Park S. Hedley Co., 361 Delaware Ave., Buffalo, and 31 Westgate Rd., Kenmore, N. Y.

HEEBNER, Walter M. (*M* 1922) Sales Engr., Warren Webster & Co., 20 Washington Place, Newark, and • 282 Highwood Ave., Teaneck, N. J.

HEIBEL, Walter E. (*M* 1917) Dist. Mgr., • Aerofin Corp., 11 West 42nd St., New York, N. Y., and Old Greenwich, Conn.

HEILMAN, Russell H.* (*M* 1923) Sr. Industrial Fellow, • Mellon Institute, and 2303 Beechwood Blvd., Pittsburgh, Pa.

HEINKEL, Charles E. (*J* 1938) Branch Mgr., • The Trane Co., 310 Postal Bldg., and 1956 N.W. Raleigh, Portland, Ore.

HEISTERKAMP, Herbert W. (*A* 1940; *J* 1937) Mgr., Air Cond. Sales, Bryant Heater Co., 17825 St. Clair Ave., and • 18103 East Park Drive, Cleveland, Ohio.

HELBURN, I. B. (*M* 1929; *J* 1927) Jr. Assoc., • Wyman Engineering, 1306 Chamber of Commerce Bldg., and 3815 Winding Way, Cincinnati, Ohio.

HELLER, Joseph A. (*A* 1938) Sales, • Air Conditioning Utilities Co., 8 West 40th St., and 150 West 79th St., New York, N. Y.

HELLMERS, Charles C., Jr. (*A* 1942; *J* 1937) Gas Htg. Engr., • Iowa-Nebraska Light & Power Co., 1401 "O" St., and 2554 Woodsdale Blvd., Lincoln, Nebr.

HELLSTROM, John (*A* 1929) Vice-Pres., • American Air Filter Co., Inc., 215 Central Ave., and 423 Lightfoot Rd., Louisville, Ky.

HELSTROM, Clifford W. (*M* 1938) Mgr. Htg. and Plbg. Dept., Globe Machinery & Supply Co., E. First and Court Ave., and • 1614 Thompson Des Moines, Iowa.

HELSTROM, Herman G. (*M* 1928) Firebox Boiler and Stoker Div., William Bros. Boiler & Manufacturing Co., Nicollet Island, and • 4608 Arden Ave. S., Minneapolis, Minn.

HELWICK, Numa John (*M* 1910; *A* 1940) Secy. and Mech. Engr., • American Heating & Plumbing Co., Inc., 829 Baronne St., and 809 Greenwood St., New Orleans, La.

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HENRY, Alexander S., Jr. (*M* 1930) Mech. Engr., R. K. O., 1270 Sixth Ave., and • 300 Central Park West, New York, N. Y.

HENRY, Ernest C. (*M* 1938) Owner, E. C. Henry Co., 1317 S. Water St., and • 1115 Park Ave., Bay City, Mich.

HEPBURN, E. M. (*A* 1940) Branch Mgr., • Empire Brass Manufacturing Co., Ltd., 74 Princess St., and 1045 McMillan Ave., Winnipeg, Man., Canada.

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HERBERT, Richard M. (*J* 1938) Lt., A.C., Brooks Field, and •110 Harding Place, San Antonio, Tex.

HERING, Alfred (*M* 1935) Pres., Hering Heating Co., Inc., 203 East 88th St., and •1830 Tenbroeck Ave., New York, N. Y.

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HERMAN, Neil B. (*J* 1937; *S* 1936) Engr., Riggs, Distler & Co., Inc., 516 Fifth Ave., New York, N. Y., and •4217 Garfield Ave. S., Minneapolis, Minn.

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HERTY, Frank B. (*M* 1933) The Gas Engine & Electric Co., Inc., 280 Meeting St., and •294 Congress St., Charleston, S. C.

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HESS, Arthur J. (*M* 1937) Engr., •English & Lauer, Inc., 1978 S. Los Angeles St., Los Angeles, and 897 Linda Vista, Pasadena, Calif.

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HEYDON, Charles G. (*A* 1923) Mgr., Sales of Western Div., Wright Austin Co., 315 W. Woodbridge St., and •2081 Nebraska, Detroit, Mich.

HEYMSFIELD, Herbert R. (*M* 1941; *A* 1926) Automatic Htg. Instructor, Brooklyn Technical High School, and •478 Third St., Brooklyn, N. Y.

HIBBS, Frank C. (*M* 1917) Htg. Engr., The H. B. Smith Co., Inc., 2209 Chestnut St., and •840 North 65th St., Philadelphia, Pa.

HICKEY, Daniel W. (*A* 1931) Pres., •D. W. Hickey & Co., Inc., 1841 University Ave., and 1874 Highland Pkwy., St. Paul, Minn.

HICKMAN, Herbert V. (*A* 1938) Sales Engr., •Anemostat Corp. of America, 1129 Folsom St., San Francisco, and 433 San Diego Ave., Daly City, Calif.

HIERS, Charles R. (*M* 1929; *J* 1927) Modine Manufacturing Co., 101 Park Ave., New York, and •19 Westminster Rd., Great Neck, L. I., N.Y.

HIGH, John M. (*M* 1940; *A* 1938) Mgr., Insulation Div., •The Ruberoid Co., 500 Fifth Ave., and 306 Lexington Ave., New York, N. Y.

HILDER, Frederick L. (*M* 1937) Chief Engr., Electric Furnace-Man, Inc., Emmaus, and •2604 Walnut St., Allentown, Pa.

HILDRETH, Egbert S. (*A* 1936) Air Cond. Engr., •Indianapolis Power & Light Co., 17 N. Meridian St., and 5626 E. Michigan St., Indianapolis, Ind.

HILL, Charles F. (*A* 1942; *J* 1936) 52 Grace St., Malden, Mass.

HILL, Edward, Jr. (*A* 1942; *J* 1939) 1st Lt., United States Army, Camp Roberts, and •102 12th St., Paso Robles, Calif.

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HILL, H. G. (*J* 1938) Mgr., Furnace & Air Cond. Div., Gurney Foundry Co., Ltd., Junction Rd., and •7 Armadale Ave., Toronto, Ont., Canada.

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HILL, Jared A. (*M* 1938) Htg. and Air Cond. Engr., Pacific Gas & Electric Co., 245 Market St., San Francisco, and •715 Laurel Ave., Burlingame, Calif.

HILL, Vaughn H. (*J* 1938) Htg. Engr., Motor Wheel Corp., and •2111 Colvin Court, Lansing, Mich.

HILL, Walter W. (*J* 1940) Engr., United Clay Products Co., 931 Investment Bldg., and •3415 Fessenden St. N., Washington, D. C.

HILLS, Arthur H. (*M* 1924) Mgr., •Sarco Canada, Ltd., 85 Richmond St. W., and 21 Nealon Ave., Toronto, Ont., Canada.

HINES, Guy M. (*A* 1937) Chief Engr., Texas Agricultural & Mechanical College, Power Dept., and •P.O. Box 248, College Station, Tex.

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HINRICHSEN, Arthur F. (*M* 1928) Pres., •A. F. Hinrichsen, Inc., 50 Church St., New York, N.Y., and Mountain Lakes, N. J.

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HITCHCOCK, Paul C. (*M* 1931) Pres., •Hitchcock & Estabrook, Inc., 521 Sexton Bldg., and 5130 Harriet Ave., Minneapolis, Minn.

HITT, John C. (*A* 1936) Branch Mgr., •Holland Furnace Co., 55 18th St., and Howard Place, Wheeling, W. Va.

HOBBIE, E. H. (*A* 1937) Mgr., Sales Promotion, •Mississippi Glass Co., 220 Fifth Ave., New York, N. Y., and Ridgedale Ave., Florham Park, N. J.

HOBBS, J. G. (*M* 1920) Vice-Pres., •Diamond Alkali Co., P.O. Box 430, and 60 Wood St., Painesville, Ohio.

HOBBS, William S. (*A* 1936) •P.O. Box 269, and 327 Park Ave., Swarthmore, Pa.

HOCKENSMITH, Francis E. (*M* 1936) Chief Engr., •Lennox Furnace Co., Inc., 400 N. Midler Ave., and 458 Plymouth Drive, Syracuse, N. Y.

HODGE, William B. (*M* 1934) Vice-Pres., •Parks-Cramer Co., P.O. Box 23, and 2600 Roswell Ave., Charlotte, N. C.

HOECKER, George F., Jr. (*A* 1941) Sales Mgr., Sears, Roebuck & Co., Union City, and •520 Burton Ave., Hasbrouck Heights, N. J.

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HOFFMAN, Harry (*M* 1939) Branch Mgr., •Johnson Service Co., 105 Piedmont Bldg., Greensboro, and Route 1, Guilford College, N. C.

HOFFMANN, Angelo (*A* 1938) Vice-Pres., •Louis Hoffman Co., 117 W. Pittsburgh Ave., and 4850 N. Oakland Ave., Milwaukee, Wis.

HOGAN, Edward Lawrence* (*M* 1911) Gen. Consulting Engr., American Blower Corp., •P. O. Box 58, Roosevelt Park Station, and Alden Park Manor, 8100 E. Jefferson Ave., Detroit, Mich.

HOGUE, William M. (*A* 1935) Sales Engr., •United States Electrical Motors, Inc., 200 E. Slauson Ave., and 4839 Keniston Ave., Los Angeles, Calif.

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LEEK, Walter (*Life Member*; *M* 1903) Pres., •Leek & Co., Ltd., 1111 Homer St., and 4769 W. Second Ave., Vancouver, B.C., Canada.

LEFEBVRE, Eugene J. (*M* 1937) Engr., Warden King, Ltd., 2104 Bennett Ave., Montreal, and •378 Wood Ave., Westmount, Que., Canada.

LEFFEL, Paul C. (*A* 1941) Owner, The Leffel Co., 3323 Main, and •316 East 75th St., Kansas City, Mo.

LEGLER, Frederick W. (*M* 1935; *A* 1933) Pres., The Waterbury Co., 17 West 28th St., and •2919 Johnson St. N.E., Minneapolis, Minn.

LEHMAN, M. G. (*A* 1937) Owner, •720 O St., and 2011 Worthington, Lincoln, Nebr.

LEIBY, R. S. (*A* 1940) Owner, •Robert S. Leiby, 212 N. Grant Ave., Columbus, and 6870 Havens Corners Rd., Blacklick, Ohio.

LEICHNITZ, Robert W. (*J* 1936) Mechanic, U. S. Navy, Puget Sound Navy Yard, Bremerton, and •203 Belmont N., Seattle, Wash.

LEILICH, Roger L. (*M* 1922) Pres., •The Wallace Stebbins Co., 100 S. Charles St., Baltimore, and Ruxton, Baltimore Co., Md.

LEINROTH, J. P. (*M* 1929) Gen. Industrial Fuel Repr., •Public Service Electric & Gas Co., 80 Park Place, Newark, N. J.

LEITCH, Arthur S. (*M* 1908) Pres. and Managing Dir., The Arthur S. Leitch Co., Ltd., 1123 Bay St., and •421 Russell Hill Rd., Toronto, Ont., Canada.

LEITGABEL, Kenneth A. (*J* 1942; *S* 1939) Jr. Engr., •Modine Manufacturing Co., 1243 Arthur Ave., Racine, and 2320 North 58th St., Milwaukee, Wis.

LELAND, Warren B. (*M* 1929) Sales Engr., •The H. B. Smith Co., Inc., 57 Main St., Westfield, and 159 Sumner Ave., Springfield, Mass.

LELAND, William E. (*Life Member*; *M* 1915) Partner, •Leland & Haley, 58 Sutter St., San Francisco, and 704 The Alameda, Berkeley, Calif.

LENIHAN, William O. (*A* 1936) Vice-Pres., •Laverack & Haines, Inc., White Bldg., and 703 W. Ferry St., Buffalo, N. Y.

LENONE, Jose M. (*M* 1919) Designing Engr., •Wilson & Co., Inc., 4100 S. Ashland Ave., and 1017 East 46th St., Chicago, Ill.

LEONARD, Lorcan C. G. (*J* 1937) Tech. Mgr., •McCann Jeffreys, Ltd., 19-20 Ellis' Quay, and 265 Clontarf Rd., Dollymount, Dublin, Ireland.

LEONHARD, Lee W. (*M* 1936) Supvr., Eastman Kodak Co., and •1075 Winona Blvd., Rochester, N. Y.

LEONARD, R. R. (*A* 1942; *J* 1941) Sales Engr., A. M. Byers Co., 1409 Girard Trust Bldg., Philadelphia, Pa.

LEOPOLD, Charles S. (*M* 1934) Consulting Engr., •213 S. Broad St., Philadelphia, and 7600 West Ave., Elkins Park, Pa.

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LESSINGER, Edgar F. (*A* 1941) Mgr., •Lessinger Plumbing & Heating Co., 221 S. 10th St., and 814 North 18th St., Boise, Idaho.

LEUPOLD, George L. (*A* 1937) Sales Engr., Minneapolis-Honeywell Regulator Co., 561 Reading Rd., and •1630 Northwood Dr., Cincinnati, Ohio.

LEUTHESSER, Fred W., Jr. (*M* 1937) Secy., •National Metal Products Corp., 21 N. Loomis St., Chicago, and 1715 North 77th Court, Elmwood Park, Ill.

LEVINE, Charles (*J* 1939) Air Cond. Service Engr., Alfred L. Hart, Inc., 315 Vanderbilt Ave., and •1171 Ocean Parkway, Brooklyn, N. Y.

LEVINE, Lawrence J. (*J* 1940) Asst. Engr., Paragon Oil Burner Corp., 75 Bridgewater St., and •1378 East 12th St., Brooklyn, N. Y.

LEVITT, Leroy L. (*A* 1942; *J* 1940) Air Cond. Engr., Calvert Distilling Co., Relay, and •3326 Gwynns Falls Pkwy., Baltimore, Md.

LEVY, Marion I. (*M* 1938; *A* 1936; *J* 1931) Pres., •Viking Air Conditioning Corp., 5606 Walworth Ave., and 3156 Ludlow Rd., Cleveland, Ohio.

LEWIS, Carroll E. (*M* 1930) Sales Mgr., Delco-Appliance Div., General Motors Sales Corp., 391 Lyell Ave., and •9 Shelwood Rd., Rochester, N. Y.

LEWIS, Harry E. (*A* 1942; *J* 1939) Engr., Eagle Picher Lead Co., Temple Bar Bldg., and •220 Atkinson, Apt. 2, Cincinnati, Ohio.

LEWIS, H. Frederick (*M* 1940; *A* 1937) Vice-Pres. and Gen. Mgr., •Harvey A. Dwight Oil Heat and Supply Co., Inc., 147 Dongan Ave., Albany, and Sweet's Crossing, Nassau, N. Y.

LEWIS, Kenneth C. (*A* 1938) 306 General Motors Research Bldg., Detroit, Mich.

LEWIS, L. L.* (*M* 1918) Vice-Pres. and Chief Engr., •Carrier Corp., S. Geddes St., and 207 Sedgwick Drive, Syracuse, N. Y.

LEWIS, Samuel R.* (*M* 1905) (*Presidential Member*) (Pres., 1914; 2nd Vice-Pres., 1910; Board of Governors, 1909-10; 1912; Council, 1914-15) Consulting Mech. Engr., •407 S. Dearborn St., and 4737 Kimbark Ave., Chicago, Ill.

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LEWIS, W. Warner (*A* 1939) Mgr., Refrig. and Air Cond., •Major Appliance Dept., Beeson Hardware Co., P.O. Box 1390, and 604 Woodrow Ave., High Point, N. C.

LEY, Ralph B. (*A* 1940) Htg.-Vtg. Engr., Architect's Office, District Government, Washington, D. C., and •Pine Whiff Beach, Edgewater, Md.

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LLOYD, Edward C. (*M* 1927) •Armstrong Cork Co., and R. D. 5, Old Philadelphia Pike, Lancaster, Pa.
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LOH, Nan-Shee (*M* 1933; *A* 1931; *J* 1927) Mgr., New Shanghai Heating & Plumbing Co., 330 National Commerce Bank Bldg., 406 Kiangse Rd., Shanghai, China.
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LONG, Edward J. (*S* 1939) Chief Plbg. Engr., Carr & J. E. Greiner Co., Marine Barracks, Jacksonville, and •Box 232A, R. R. No. 1, Newport, N. C.
LONG, Herbert P. (*M* 1938) Sales Engr., •Buffalo Forge Co., 490 Broadway, Buffalo, N. Y.
LONG, Wayne E. (*M* 1935) Prof. Mech. Engrg., •Texas Agricultural & Mechanical College, College Station, Tex.
LONGCOY, Grant B. (*M* 1933) Engr., Joseph Breslove, Cons. Engr., Leader Bldg., Cleveland, and •1215 Ramona Ave., Lakewood, Ohio.
LOO, Ping Yok (*M* 1933) Gen. Mgr., •China Engineering Co., 30 Brenan Rd., Shanghai, and 271-73 Dunbarton Rd., Tientsin, China.
LOUCKS, D. W. (*A* 1930) Supvr., Commercial Elec. & Steam Sales, •Duquesne Light Co., 435 Sixth Ave., and 1049 Osage Drive, Wilkinsburg, Pittsburgh, Pa.
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LOVE, Clarence H. (*M* 1919) Mfrs. Agent, Nash Engineering Co., 421 Chamber of Commerce, and •289 Norwalk Ave., Buffalo, N. Y.
LOWE, Robert A. (*J* 1938) Engr., United States Government, and •4015½ Country Club Drive, Los Angeles, Calif.
LOWE, Walter (*J* 1940; *S* 1938) Htg. Engr., The Peoples Natural Gas Co., 545 Wm. Penn Way, and •443 Althea St., Pittsburgh, Pa.
LOWNSBERY, B. F. (*M* 1920) Project Engr., Benjamin F. Shaw Co., 201 E. Lombard St., and •21 S. Sycamore St., Wilmington, Del.
LUCK, Alexander W.* (*Life Member*; *M* 1919) Htg. Engr.-Consultant, 532 Woodward St., Reading, and •P.O. Box 34, Reiffton, Pa.
LUCKE, Charles E. (*M* 1924) Stevens Prof. Emeritus Mech. Engrg., •Columbia University, Pupin Bldg., and 186 Riverside Drive, New York, N. Y.
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LUTY, Donald J. (*M* 1933) Asst. Gen. Mgr., Air Cond. Div., •Gar Wood Industries, Inc., 7924 Riopelle St., and 13661 Cloverlawn Ave., Detroit, Mich.
LYFORD, Robert G. (*J* 1939) Branch Mgr., •The Powers Regulator Co., 1634 Allen Bldg., and 2717 E. Amherst Ave., Dallas, Tex.
LYLE, J. I.* (*M* 1911) (*Presidential Member*) (Pres., 1917; Council, 1917-18) Pres., •Carrier Corp., and Orchard Rd., Syracuse, N. Y.
LYMAN, Samuel E. (*A* 1924) Buensod-Stacey Air Conditioning, Inc., 60 East 42nd St., New York, N. Y., and •865 Hueston St., Union, N. J.
LYNCH, James R. (*A* 1940) Owner, •Lynch Furnace Co., 1804 N.E. Union, and 2952 N.E. Edgehill Place, Portland, Ore.
LYNCH, William L. (*M* 1928) Pres., •Rome-Turney Radiator Co., and 1205 N. George St., Rome, N. Y.
LYNN, Frederick E. (*M* 1938) Refrig. Engr., Electric Products Corp., 5624 Penn Ave., Pittsburgh, and •312 Moyhend St., Springdale, Pa.
LYON, Douglas McClure (*A* 1941) Partner, •Douglas McClure Lyon, 317 State Tower Bldg., and 1826 James St., Syracuse, N. Y.
LYON, P. S. (*M* 1929) Pres., •Cochrane Corp., 17th St. and Allegheny Ave., and 3416 Warden Drive, Philadelphia, Pa.
LYONS, Cornelius J. (*A* 1932) Sales Engr., Nash Engineering Co., Wilson Ave., and •5 Olmstead Place, South Norwalk, Conn.

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MABLEY, Louis C. (*M* 1937) Lt., U. S. N. R., Executive Officer, U. S. Naval Section Base, Sabine Pass, Tex.
MABLEY, T. Hollister (*M* 1939) Chief Engr., •Mechanical Heat & Cold, Inc., 7704 Woodward Ave., Detroit, and 2323 Yorkshire Rd., Birmingham, Mich.
MABON, James E. (*J* 1942; *S* 1939) Engr., The Glenn L. Martin Co., 409 Southway, Baltimore, Md., and •340 N. Walnut St., Blairsville, Pa.
MACCUBBIN, Howard A. (*M* 1934) Buyer, Htg. Equip., Montgomery Ward & Co., and •4914 N. Mason Ave., Chicago, Ill.

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MacDONALD, Douglas J. (*M* 1935) Vice-Pres., Htg. Div., • Standard Sanitary Dominion Radiator Co., Ltd., Royce and Lansdowne Aves., and 96 Hudson Drive, Toronto, Ont., Canada.
MacEACHIN, Graham C. (*M* 1938) Capt., Corps of Engrs., U. S. Army, Co. C., 35th Engineer Tng. Battalion, Fort Leonard Wood, Mo., and • 5112 Byers Ave., Fort Worth, Tex.
MacGREGOR, Cecil M. (*A* 1939) Capt., F. A., U. S. Army, Field Artillery School, Fort Sill, and • Box 1193, Lawton, Okla.
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MACHEREL, Ferdinand (*M* 1939) Mgr.-Owner, INCO, "Industries et Coufert," 50 Rue Daquerre, Algiers, French North Africa.
MACHIN, Donald W. (*J* 1935) Fuel Service Engr., Pittsburgh & Midway Coal Mining Co., 610 Dwight Bldg., Kansas City, Mo., and • 2112 Vermont St., Lawrence, Kan.
MACK, Emil H. (*A* 1938) Asst. Sales Mgr., The Vilter Manufacturing Co., 2217 S. First St., and • 2225 N. Booth St., Milwaukee, Wis.
MACK, Ludwig (*M* 1935) Dist. Mgr., Cooling & Air Cond. Div., B. F. Sturtevant Co., Cresmont and Haddon Aves., Camden, N. J., and • 412 W. Hortter St., Philadelphia, Pa.
MacLACHLAN, Victor D. (*A* 1939; *J* 1938) Flight Lt., R.A.F.V.R., • Honeywell-Brown, Ltd., Wadsworth Rd., Perivale, Greenford, Middlesex, and R.A.F. Station, Digby, Lincoln, England.
MacLEAN, H. A. (*M* 1939) Mgr., • The MacLean Plumbing Service, P.O. Box 400, and 89 Tremoy Rd., Noranda, Que., Canada.
MacMILLAN, Alexander R. (*M* 1936) Mgr., Delco Appliance Div., • General Motors Sales Corp., 2-160 General Motors Bldg., and 2455 Longfellow Ave., Detroit, Mich.
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MacWATT, Donald A. (*M* 1938) Sales Engr., Powers Regulator Co., 231 East 46th St., New York, and • Plandome, L. I., N. Y.
MADDUX, O. Lloyd (*M* 1935; *A* 1933) Owner, O. Lloyd Maddux, 53 Park Place, New York, N. Y., and • 17 Tallmadge Ave., Chatham, N. J.
MADELY, Frederick J. (*A* 1930) Chief Estimator, Eastern Steel Products, Ltd., 1335 Delorimier Ave., and • 6370 Louis-Hemon St., Montreal, Que., Canada.
MADISON, Richard D. (*M* 1926) Research Engr., • Buffalo Forge Co., 490 Broadway, Buffalo, and 218 Brantwood Rd., Snyder, N. Y.
MAEHLING, Leon S. (*M* 1932) Supt. of Service, Equitable Gas Co., 6304 Penn Ave., and • 778 Country Club Drive, Pittsburgh, Pa.
MaGIRL, Willis J. (*M* 1934; *A* 1931; *J* 1927) Chief Engr., • P. H. MaGirl Foundry & Furnace Works, 401-13 E. Oakland Ave., and 1119 E. Monroe St., Bloomington, Ill.
MAGNUSSON, Nicholas (*A* 1938) Estimator-Designer-Sales, Montgomery Ward & Co., 150-15 Jamaica Ave., and • 138-05 Linden Blvd., Jamaica, L. I., N. Y.
MAHON, B. B. (*M* 1935) Principal of Air Cond., • International Correspondence School, Wyoming Ave. and Ash St., and 433 Fig St., Scranton, Pa.
MAHON, Clarence A. (*A* 1938) Public Works Dept., • City Hall, 18th Floor, and 6123 Kenwood Ave., Kansas City, Mo.
MAHON, Frank B. (*M* 1937) Industrial Sales Promotion, • Duquesne Light Co., 435 Sixth Ave., and 290 LeMoyne Ave., Pittsburgh, Pa.
MAHONEY, David J. (*M* 1930; *A* 1926) Branch Mgr., • Johnson Service Co., 503 Franklin St., and 140 Linwood Ave., Buffalo, N. Y.
MAIER, George M. (*M* 1921) • American Radiator & Standard Sanitary Corp., Bessemer Bldg., Room 1226, Pittsburgh, and 135 Longue Vue Dr., Mt. Lebanon, Pa.
MAIER, Herman F. (*M* 1926) Chief Engr.-Secy., • New York Blower Co., 3155 Shields Ave., and 7124 S. Morgan St., Chicago, Ill.
MAKIN, Henry T., Jr. (*M* 1939) Engr. and Archts. Repr., American Radiator & Standard Sanitary Corp., 2212 Walnut St., and • 301 Wadsworth Ave., Philadelphia, Pa.
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MALLY, Chester F. (*M* 1940; *A* 1938) Gen. Mgr., • Mally & Co., 307 Boulevard Bldg., Detroit, and 292 W. Woodland Ave., Ferndale, Mich.
MALONE, Dayle G. (*M* 1929; *A* 1925) Branch Mgr., • Petroleum Heat & Power Co., 3301 S. California Ave., and 7337 Merrill Ave., Chicago, Ill.
MALONE, James S. (*A* 1936) Dist. Repr., • Hoffman Specialty Co., 4 N. Eighth St., and 7124 Waterman Ave., St. Louis, Mo.
MALVIN, Ray C. (*M* 1929) Pres., • Malvin & May, Inc., 2015 S. Michigan Ave., and 8220 Dante Ave., Chicago, Ill.
MANDELL, Thomas P. (*A* 1937) Sales, Carrier Corp., 704 Statler Office Bldg., Boston, and • Walnut Rd., South Hamilton, Mass.
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MANN, Walter N. (*M* 1939) Gen. Mgr., Brockhouse Heater Co., Ltd., Victoria Works, West Bromwich, Staffs., and • "Moneymore," Canwell, Sutton Coldfield, Warwickshire, England.
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MARTYN, Henry J. (*A* 1937) Pres., • Martyn Bros., Inc., 911 Camp St., and 5306 Ridgedale St., Dallas, Tex.

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MARZORATI, Giuseppe (*M* 1938) Consulting Engr., • S.i. N.C. Giacomo Jucker, Mauro Macchi 28, and Baldissera 9, Milano, Italy.

MASON, Ray B. (*M* 1925) Sales Engr., • Kewanee Boiler Corp., 2014 Wyandotte St., and 121 East 70th Terrace, Kansas City, Mo.

MAST, Clyde M. (*A* 1940) Heating & Air Conditioning Supply, Inc., 263 Sierra St., and • 536 Nixon St., Reno, Nev.

MATCHETT, James C. (*M* 1923) Vice-Pres. and Gen. Mgr., • Illinois Engineering Co., Racine Ave. and 21st St., and 9936 S. Winchester Ave., Chicago, Ill.

MATHEKA, Charles R. (*S* 1939) U.S.S. Tattnall, • c/o Postmaster General, New York, N. Y., and 1506 Summit Ave., Union City, N. J.

MATHER, Harry H. (*A* 1929) Treas., Mather Paper Co., 611 S. Front St., Philadelphia, and • 373 Lakeview Ave., Drexel Hill, Pa.

MATHEWSON, M. E. (*M* 1937) Secy., • A. M. Kinney, Inc., 1301 Enquirer Bldg., and 3569 Erie Ave., Cincinnati, Ohio.

MATHIS, Eugene* (*M* 1922) Vice-Pres. and Treas., • The New York Blower Co., 32nd St. and Shields Ave., Armour P.O. Station, and 9151 S. Hoyne Ave., Chicago, Ill.

MATHIS, Henry (*M* 1921) The New York Blower Co., 32nd St. and Shields Ave., Armour P. O. Station, and • 11246 Longwood Dr., Chicago, Ill.

MATHIS, John (*A* 1938) Engr., • Standard Furnace & Supply Co., 407 S. Tenth St., and 109 South 42nd St., Omaha, Nebr.

MATHIS, Julien W. (*A* 1921) • New York Blower Co., 3145-55 Shields Ave., and 7929 Bishop St., Chicago, Ill.

MATHISON, Russell St. Clair (*A* 1938) Asst. Mgr., Weathermakers (Canada), Ltd., 593 Adelaide St. W., and • 44 Strathgowan Ave., Toronto, Ont., Canada.

MATOUSEK, A. G. (*M* 1937) Mgr., Gamble Store, Schuyler, Nebr.

MATTHEWS, John E. (*M* 1934) Sales Engr., B. F. Sturtevant Co., Crestmont & Haddon Ave., Camden, and • 300 Chestnut St., Haddonfield, N. J.

MATTHIES, Leo A. (*S* 1941) Ensign, U.S.N.R., • B. O. 2, Naval Air Station, Seattle, Wash., and 4433 18th Ave. S., Minneapolis, Minn.

MATTINGLY, Maurice F. (*A* 1939) Sales Engr., • Johnson Service Co., 1355 Washington Blvd., and 8028 Ingleside Ave., Chicago, Ill.

MATZ, George N. (*M* 1938) Mech. Engr., A. Ernest D'Ambly, 2101 Architects Bldg., Philadelphia, and • 649 Ferne Ave., Drexel Hill, Pa.

MATZEN, Harry B. (*M* 1919) Consulting Mech. Engr., 185 Madison Ave., New York, and • 16 Addison Place, Rockville Centre, L. I., N. Y.

MAURER, Lester (*M* 1941) Asst. Engr., Navy Dept., and • 2027 Massachusetts Ave. N.W., Washington, D. C.

MAVES, G. D. (*A* 1939) Sales Engr., • Minneapolis-Honeywell Regulator Co., 1305 Capitol Ave., and 2340 Wroxton Rd., Houston, Tex.

MAWBY, Pensyl (*M* 1934) Dist. Sales Mgr., Lehigh Navigation Coal Co., 123 S. Broad St., Philadelphia, and • 15 E. Ridley Ave., Ridley Park, Pa.

MAXWELL, George W. (*M* 1935; *S* 1932) Engr., Kenealy & Maxwell, Main St., and • Lower County Rd., Harwich Port, Mass.

MAXWELL, R. Shierlaw (*M* 1937) Gen. Mgr., • Bennett & Wright, Ltd., 72 Queen St. E., and 107 Cheltenham Ave., Toronto, Ont., Canada.

MAY, Arthur O. (*A* 1938; *J* 1928) Secy., • Stannard Power Equipment Co., 53 W. Jackson Blvd., Room 925, and 5736 N. Bernard St., Chicago, Ill.

MAY, C. W. (*M* 1933) Consulting Engr., • 1503 Smith Tower, and 6056 Fourth N.E., Seattle, Wash.

MAY, Edward M. (*M* 1931) Branch Mgr. and Combustion Engr., Steel Products Engineering Co., 1601 S. Michigan Ave., Chicago, and • 848 N. Ridgeland Ave., Oak Park, Ill.

MAY, George Elmer* (*M* 1933) Utilization Engr., New Orleans Public Service, Inc., 317 Baronne St., New Orleans, La.

MAY, James W. (*M* 1938; *J* 1935) Assoc. Prof. of Htg. and Vtg., • College of Engrg., University of Kentucky, and 1046 Fontaine Rd., Lexington, Ky.

MAY, Maxwell F. (*M* 1929) Vice-Pres., • Young Radiator Co., Racine, Wis., and Palos Park, Ill.

MAYETTE, Charles E. (*M* 1926) Sales Engr., E. B. Badger & Sons Co., 75 Pitts St., and • Engineers Club, Boston, Mass.

MAYNARD, J. Earle (*M* 1931) Chief Htg. Engr., Sales Engrg. Dept., American Radiator & Standard Sanitary Corp., and • 324 Fifth St., Elyria, Ohio.

MAYNE, Walter L. (*M* 1938) Vice-Pres.-Sales Mgr., • Marsh Valve Co., Brigham Rd. at Fourth St., and 741 Park Ave., Dunkirk, N. Y.

McBRIDE, J. Nevins (*A* 1941) Vice-Pres., • Frank A. McBride Co., 158 Ward St., and 228 Derrom Ave., Paterson, N. J.

McCAFFERTY, Joseph E. (*A* 1937) Dist. Engr., Petroleum Heat & Power Co., 419 Boylston St., Boston, and • 747 Front St., Weymouth, Mass.

McCAFFRAY, Charles E. (*M* 1938) Sales Engr., • 100 West 22nd St., Baltimore, and 5700 Cross Country Blvd., Mt. Washington, Md.

McCAIN, H. King (*M* 1939; *A* 1938; *J* 1937) Consulting Engr., • Newcomb & Boyd, 615 Trust Co. of Ga. Bldg., and 85 Old Ivy Rd., Atlanta, Ga.

McCALLUM, Chester E. (*M* 1941) Asst. Chief Engr., • The Bahnson Co., and Chatham Apts., Winston-Salem, N. C.

McCANN, Frank D. (*A* 1939) Supvr. Air Cond. Sales, • Westinghouse Electric & Manufacturing Co., 40 Wall St., New York, and 378 Scarsdale Rd., Crestwood, Yonkers, N. Y.

McCARTHY, John J. (*A* 1937) Htg.-Vtg. Engr., • Providence Public School Dept., 20 Summer St., and 318 Academy Ave., Providence, R. I.

McCARTHY, Thomas F. (*M* 1938) Sales Engr., Kroeschell Engineering Co., 215 W. Ontario St., and • 6948 Calumet Ave., Chicago, Ill.

McCAULEY, James H. (*M* 1921) Pres., • J. H. McCauley & Son, 5620 West 65th St., Chicago, and 707 William St., River Forest, Ill.

McCLANAHAN, L. C. (*M* 1930) Dist. Mgr., • Aerofin Corp., 603 Great National Life Bldg., and 811 S. Tyler, Dallas, Tex.

McCLELLAN, James E. (*M* 1922) Office Mgr., • American Blower Corp., 228 N. LaSalle St., Chicago, and 738 Marion Ave., Highland Park, Ill.

McCLINTOCK, William (*M* 1935) Consulting Engr., Board of Education, 34½ East 12th St., and • 647 East 232nd St., New York, N. Y.

McCLOSKEY, John H. (*A* 1940) Owner, J. H. McCloskey, 109 North St., and • 304 Elkton Blvd., Elkton, Md.

McCLUNG, Tom H. (*A* 1942; *J* 1939) Sales Engr., Brod & McClung, Lewis Bldg., and • 6626 N.E. Alameda, Portland, Ore.

McCONACHIE, L. L. (*A* 1928) Owner, • L. L. McConachie Co., 1003 Maryland Ave., Detroit, and 1415 Harvard Rd., Grosse Pointe Park, Mich.

McCONNER, Charles R. (*A* 1925; *J* 1922) Gen. Sales Mgr., • Clarage Fan Co., and 1904 Waite Ave., Kalamazoo, Mich.

McCORMACK, Denis (*M* 1933) Mgr., Commercial Instruments & Controls Dept., Julien P. Friez & Sons, Div. of Bendix Aviation Corp., 4 N. Central Ave., and • Ruxton Post Office, Baltimore, Md.

McCORMICK, George W., Jr. (*A* 1941) Sales Engr., American Blower Corp., 307 Essex Bldg., and • 802 Mt. Curve Ave., Minneapolis, Minn.

McCOY, C. E. (*M* 1936) Partner, • Turner-McCoy, 315 W. Second St., and 5117 Sherwood Rd., Little Rock, Ark.

McCOY, Thomas F. (*M* 1924) Mgr., • The Powers Regulator Co., 125 St. Botolph St., Boston, and Glen Rd., Wellesley Farms, Mass.

McCREA, Joseph B. (*M* 1937) Htg. and Vtg., 2919 Drexel Ave., Detroit, Mich.

McCREA, Lester W. (*M* 1920) Prop., • McCrea Sales Co., 19 N. Carrollton Ave., and 564 W. University Pkwy., Baltimore, Md.

McCULLEY, D. E. (*A* 1917) Mfrs. Repr., • 1101 Jackson St., and 1102 Park Ave., No. 3, Omaha, Nebr.

McCULLOUGH, Henry G. (*M* 1936) Vice-Pres., S. S. Fretz, Jr., Inc., 1902 Chestnut St., and • 7042 Lincoln Dr., Mt. Airy, Philadelphia, Pa.

McCULLOUGH, John L. (*M* 1930) • Harry Dougherty & Son, Freeport, and 105 Roycroft Ave., (16), Pittsburgh, Pa.

McCUNE, B. V. (*M* 1928) Owner, • McCune's Air Conditioning, 807 W. Yakima Ave., Yakima, Wash.

McCUSKER, James P. (*S* 1940) Student, Catholic University of America, and • 1445 Evarts St. N.E., Washington, D. C.

McDERMOTT, John P. (*A* 1942; *J* 1939) • Engr. Corps, Fort Belvoir, Va., and 3534 S.E. Claybourne, Portland, Ore.

McDONALD, Ivan (*A* 1938) Branch Mgr., • Minneapolis-Honeywell Regulator Co., 44 Princess St., and 132 Kingston Row, Winnipeg, Man., Canada.

McDONALD, Thomas (*A* 1931) Vice-Pres., • Minneapolis-Honeywell Regulator Co., and 4618 Drexel Ave., Minneapolis, Minn.

McDONNELL, Everett N. (*M* 1923) (Council, 1940-41) Pres., • McDonnell & Miller, 400 N. Michigan Ave., and Drake Hotel, Chicago, Ill.

McDONNELL, John E. (*A* 1936) Sales Engr., • McDonnell & Miller, 400 N. Michigan Ave., Chicago, and 2299 Lakeside Place, Highland Park, Ill.

McDOWELL, Harry L. (*J* 1939) Sales Repr., • U. S. Radiator Corp., 1128 Belt Line Blvd., Columbia, S. C.

McELGIN, John W.* (*A* 1937; *J* 1931) Engr., J. J. Nesbitt, Inc., Holmesburg, Philadelphia, and • Limekiln and Butler Pikes, Ambler, Pa.

McENTEE, Francis M. (*M* 1940) Asst. Supervising Air Cond. Engr., • Office of the Architect, U. S. Capitol, and 718 Somerset Place, Washington, D. C.

McGEORGE, Richard H. (*M* 1927) Mgr., Htg.-Air Cond. Dept., McCord Radiator & Manufacturing Co., 2587 E. Grand Blvd., and • 14565 Glastonbury Rd., Detroit, Mich.

McGINN, George F. (*A* 1941) Maj., Infantry, • Reception Center, Fort George G. Meade, Md., and 106 Beechwood Ave., Catonsville, Md.

McGINNIS, Frank L. (*M* 1940) Mech. Engr., Williamsburg Restorations, Inc., and • 332 N. Henry St., Williamsburg, Va.

McGONAGLE, Arthur (*M* 1932) Consulting Engr., • 1013 Fulton Bldg., Pittsburgh, and 6815 Prospect Ave., Ben Avon, Pa.

McGOWN, Frederick H., Jr. (*J* 1941; *S* 1939) Instructor, • Wyoming Seminary, Kingston, Pa., and P.O. Box 105, Cooperstown, N. Y.

McGRAIL, Thomas E. (*M* 1926) Local Repr., • Canadian Sirocco Co., Ltd., 63 Sparks St., Ottawa, Ont., Canada.

McILVAINE, John H.* (*M* 1929) Pres., Landwehr Heating Corp., Sixth and Cuyuga Sts., Philadelphia, and • 601 Pembroke Rd., Bryn Mawr, Pa.

McINDOE, James F. (*M* 1939; *A* 1931) Sales, American Radiator & Standard Sanitary Corp., 1001 Pacific Bldg., and • 1863 N.W. Aspen St., Portland, Ore.

McINTIRE, James F.* (*M* 1915; *A* 1914) (*Presidential Member*) (Pres., 1939; 1st Vice-Pres., 1938; 2nd Vice-Pres., 1937; Council, 1926-28; 1932-40) Vice-Pres., • U. S. Radiator Corp., 1056 First National Bank Bldg., and 3261 Sherbourne Rd., Detroit, Mich.

McINTOSH, Fabian C. (*M* 1921; *J* 1917) (Council, 1929-31; 1933-35) Branch Mgr., • Johnson Service Co., 1238 Brighton Rd., and 3650 Perrysville Ave., Pittsburgh, Pa.

McKEE, J. Wilson (*A* 1938) Branch Mgr., • U. S. Radiator Corp., 439 N. Plankinton Ave., and 6713 W. Bluemound Rd., Milwaukee, Wis.

McKEEMAN, Clyde A.* (*M* 1936) Assoc. Prof. Mech. Engrg., • Case School of Applied Science, 10900 Euclid Ave., Cleveland, and 4035 Bluestone Rd., Cleveland Heights, Ohio.

McKENZIE, Charles Murdock, Jr. (*M* 1938) Htg. Engr., Southern California Gas Co., 810 S. Flower St., and • 3806 Boyce Ave., Los Angeles, Calif.

McKERLIE, Jardine (*M* 1938) Managine Dir., • Industrial Training Systems, Ltd., 67 Carlton St., and 15 Glen Arden Rd., Toronto, Ont., Canada.

McKINLEY, Carroll B. (*A* 1942; *J* 1936; *S* 1934) Sr. Engr., • Kohlenberger Engineering Corp., Fullerton, and 523 S. Lemon, Anaheim, Calif.

McKINNEY, Carl A. (*A* 1939; *J* 1937) Air Cond. Engr., • United Gas Corp., 708 United Gas Bldg., and 1918 Park St., Houston, Tex.

McKINNEY, William J. (*M* 1938; *A* 1934) Dist. Mgr., • American Blower Corp., 714-101 Marietta St. Bldg., and 3363 Mathieson Dr. N.W., Atlanta, Ga.

McKITRICK, Walter D. (*M* 1936) Mech. Engr., Robert & Co., Inc., 51 East 42nd St., and • 680 Riverside Drive, New York, N. Y.

McKITTRICK, Percy A. (*A* 1934) Treas.-Gen. Mgr., Parks-Cramer Co., P. O. Box 444, and 219 Blossom St., Fitchburg, Mass.

McLAREN, T. H. (*A* 1938) Gen. Sales Mgr., • The James Morrison Brass Manufacturing Co., Ltd., 276 King St. W., and 2084 Girard St. E., Toronto, Ont., Canada.

McLARNEY, Harry W. (*M* 1933) Air Cond. Div., • Union Electric Co. of Missouri, 315 North 12th Blvd., St. Louis, and 807 Hawbrook Rd., Glendale, Mo.

McLEAN, Dermid (*M* 1917) Mech. Engr., • Snyder & McLean, 2214 Penobscot Bldg., and 12651 Birwood Ave., Detroit, Mich.

McLEISH, William S. (*A* 1932; *J* 1928) • The Ric-wiL Co., 1563 Union Commerce Bldg., Cleveland, Ohio, and 3101 Sunset Dr., Morehead City, N. C.

McLENEGAN, D. W.* (*M* 1933) Engr. in Charge, Air Cond. and Commercial Refrig. Dept., • General Electric Co., 5 Lawrence St., Bloomfield, and 73 Arlington Ave., Caldwell, N. J.

McLOUTH, Bruce F. (*M* 1936; *J* 1934) Pres. and Gen. Mgr., • McLouth Air Conditioning Corp., 2400 E. Michigan Ave., Lansing, and 135 Gunson, East Lansing, Mich.

McMAHON, Thomas W. (*M* 1928) Dist. Mgr., • American Blower Corp., 1711 Railway Exchange Bldg., St. Louis, Mo.

McMULLEN, E. W. (*M* 1938) Dir. of Research, • The Eagle-Picher Lead Co., and 900 Richmond Rd., Joplin, Mo.

McNAMARA, William (*A* 1930) Mgr., • The Trane Co., 850 Cromwell Ave., and 1355 Como Ave. W., St. Paul, Minn.

McNAMEE, Earl W. (*M* 1940) Air Cond. Engr., • B. & J. Jacobs Co., 1729 John St., and 2627 Ocosta Ave., Cincinnati, Ohio.

McNEVIN, Joseph E. (*M* 1937) Owner, • Colorado Heating Co., 950 Cherokee St., and 215 East Bayaud, Apt. 4, Denver, Colo.

McPHERSON, William A. (*M* 1929) Chief, Htg.-Vtg. Div., Dept. of School Bldgs., 26 Norman St., Boston, and • 86 Dwinnell St., West Roxbury, Mass.

McQUAID, Dan J. (*M* 1934) Owner, • Dan J. McQuaid Engineering Service, 1742-46 Arapahoe St., and 1565 Milwaukee St., Denver, Colo.

McRAE, M. W. (*M* 1939) Research Engr., • Crane Co., 836 S. Michigan Ave., Chicago, and 816 Fairview Ave., Park Ridge, Ill.

MEAD, E. A. (*M* 1926) Sales Mgr., • Nash Engineering Co., South Norwalk, Conn.

MEAD, George E. (*A* 1941) Owner, • George E. Mead Co., Seattle Construction Center, Frye Hotel Bldg., and 4729 36th Ave. N.E., Seattle, Wash.

MEAD, H. K. (*A* 1939) Htg. and Vtg. Equipment, • 1100 Guardian Bldg., Portland, and Jennings Lodge, Ore.

MEAGHER, Arthur T. (*M* 1938) Dir. and Sales Mgr., Plbg. and Htg. Dept., Wm. Stairs, Son & Morrow, Ltd., 174-190 Lower Water St., and • 83 Seymour St., Halifax, Nova Scotia, Canada.

MEDOW, Jules (*A* 1941; *J* 1937) Designing Engr., Ilg Electric Ventilating Co., 2850 N. Crawford Ave., Chicago, Ill., and • 650 S. Cloverdale Ave., Los Angeles, Calif.

MEHMKEN, Herman O. (*A* 1941; *J* 1940) Engr., York Ice Machinery Corp., and • 400 W. Park, Edwardsville, Ill.

MEHNE, Carl A. (*M* 1929) Htg.-Vtg. Expert, C. A. Mehne, 101 Park Ave., Room 821, New York, and • 35 Livingston St., Valhalla, N. Y.

MEILLER, Daniel V. (*A* 1941) Gas Testing Engr., • Public Service Company of Northern Illinois, 1001 S. Taylor Ave., Oak Park, and 1101 S. Fifth Ave., Maywood, Ill.

MEINHOLTZ, Herbert W. (*M* 1936) 608 Mayo Bldg., Tulsa, Okla.

MEINKE, Howard G. (*M* 1933) Div. Engr., • Consolidated Edison Co. of New York, Inc., 4 Irving Place, Room 1500, New York, and 41 Harte St., Baldwin, L. I., N. Y.

MELLON, James T. J. (*M* 1911) (Council, 1915) Pres., • Mellon Co., 4419 Ludlow St., and 431 North 63rd St., Philadelphia, Pa.

MELNICK, Nicholas A. (*M* 1941) Engr., G. M. Simonson, and • 279 Fifth Ave., San Francisco, Calif.

MELONEY, Edward J. (*M* 1930) Vice-Pres.-Secy., • Bowers Bros. Co., 2015 Sansom St., Philadelphia, and 100 E. Stewart Ave., Lansdowne, Pa.

MENDEN, Peter J. (*M* 1935) Htg. Engr., Advance Heating Co., 910 Herrick Ave., and • 2200 Taylor Ave., Racine, Wis.

MENSING, Frederick D. (*M* 1920) (Treas., 1931-32), (Council, 1931-32), Consulting Engr., Mensing & Co., 2845 Frankford Ave., Philadelphia, Pa.

MERENS, Seymour H. (*A* 1939) Vice-Pres., Max Miller & Co., 823 N. California Ave., and • 4935 N. Whipple St., Chicago, Ill.

MERGARDT, Albert P. (*A* 1940) Pres., • American Heating Engineering Co., Inc., 1005 New York Ave. N.W., Washington, D. C., and 3905 N. Fifth St., Arlington, Va.

MERRILL, Carle J. (*M* 1919) Treas., • C. J. Merrill, Inc., 54 St. John St., and 15 Longfellow St., Portland, Me.

MERRILL, Frank A. (*M* 1934) Consulting Engr., • Office of Hollis French, 210 South St., Boston, and 19 Auburndale Rd., Marblehead, Mass.

MERZ, Robert A. (*S* 1940) Student, Michigan State College, and • 810 West Grand River Ave., East Lansing, Mich.

MERTZ, Walter A. (*M* 1919) Secy., • The Kehm Bros. Co., 51 E. Grand Ave., and 3753 N. Keeler Ave., Chicago, Ill.

MERWIN, Gile E. (*M* 1924; *J* 1923) Dist. Mgr., The Trane Co., 5012 Parker St., Omaha, Nebr.

METCALFE, Curtis (*A* 1937) Engr., House Htg. Dept., Michigan Consolidated Gas Co., and • 14433 Faust, Detroit, Mich.

METZGER, A. F. (*M* 1940) Supvr. of Steam Utilization Div., • Allegheny County Steam Heating Co., 435 Sixth Ave., and 3421 Horne St., Pittsburgh, Pa.

METZGER, H. J. (*A* 1937) Pres., Wheeler-Blaney Co., 137 E. Water St., Kalamazoo, Mich.

MEYER, Charles L. (*M* 1930) L. J. Wing Manufacturing Co., 154 West 14th St., New York, and • American Welfare League, 86-60 Palo Alto Ave., Hollis, L. I., N. Y.

MEYER, Frank L.* (*M* 1932; *J* 1928) Pres., • The Meyer Furnace Co., and 9 Cole Court, Peoria, Ill.

MEYER, Henry C., Jr.* (*Life Member*; *M* 1898) (Council, 1915-16) Pres., • Meyer, Strong & Jones, Inc., 101 Park Ave., New York, N. Y., and 25 Highland Ave., Montclair, N. J.

MEYER, Karl A. (*M* 1938) Design Engr., New York Blower Co., 171 Factory St., and • 109 Woodward Ave., La Porte, Ind.

MICHIE, D. Fraser (*M* 1938; *A* 1930) Htg. Sales Engr., • Crane, Ltd., 93 Lombard St., and 176 Green Ave., Winnipeg, Man., Canada.

MIDEKE, Joseph M. (*A* 1938) Vice-Pres., Mideke Supply Co., 100 E. Main St., and • 2505 Northwest 19th St., Oklahoma City, Okla.

MILENER, Eugene D. (*M* 1936) Secy., Industrial and Commercial Gas Section, • American Gas Association, 420 Lexington Ave., Suite 550, New York, N. Y.

MILES, Clarence N. (*A* 1938) Foreman, Assembly Dept., Kohlenberger Engineering Corp., 805 S. Spadra Rd., and • Route 1, Box 174-A, Fullerton, Calif.

MILLARD, Junius W. (*M* 1929) 204 Glendale Ave., Alexandria, Va.

MILLER, Archibald T. (*M* 1938) Mgr., Insulation Sales, The Barrett Div., Allied Chemical & Dye Corp., 40 Rector St., New York, N. Y., and • 125 Godwin Ave., Ridgewood, N. J.

MILLER, Bruce R. (*M* 1935; *A* 1930) Mech. Engr., R. K. Werner, Cons. Engr., 816 W. T. Waggoner Bldg., Fort Worth, Tex.

MILLER, Charles A. (*A* 1917) Sales, The H. B. Smith Co., Inc., 331 Madison Ave., New York, N. Y.

MILLER, Charles W. (*M* 1919; *J* 1908) Pres., •The Rado Co., 759 N. Milwaukee St., Room 405, Milwaukee, and R-1, Box 42, Menomonee Falls, Wis.
MILLER, Edgar R. (*A* 1935) Chief Engr., •Winnipeg Cold Storage Co., Ltd., Salter and Jarvis Ave., and P.O. Box 1384, Winnipeg, Man., Canada.
MILLER, Floyd A. (*M* 1911) Inspection Engr., •Federal Works Agency, U. S. Government, 377 U. S. Court House, and 944 Montrose Ave., Chicago, Ill.
MILLER, George F. (*M* 1936) Sales Engr., •George F. Miller, 1614 K St. N.W., Washington, D. C., and 5608 Grove St., Chevy Chase, Md.
MILLER, Glen* (*A* 1937) Research Asst., California Institute of Technology, and •772 S. Lake Ave., Pasadena, Calif.
MILLER, Jacob (*M* 1936) Pres., •Hy-Grade Construction Co., Inc., 342 West 14th St., New York, and 20 East 58th St., Brooklyn, N. Y.
MILLER, John W. (*M* 1941) Engr., Motor Wheel Corp., and •R. 2, Box 602, Lansing, Mich.
MILLER, Leo B. (*M* 1926) Sales Exec., •Perfex Corp., 500 W. Oklahoma Ave., and 3481 N. Hackett Ave., Milwaukee, Wis.
MILLER, Lorin G.* (*M* 1933) Head, Mech. Engrg. Dept., •Michigan State College, R. E. Olds Hall of Engrg., and 232 University Drive, East Lansing, Mich.
MILLER, Robert A.* (*M* 1931) Tech. Sales Engr., •Pittsburgh Plate Glass Co., 2200 Grant Bldg., Pittsburgh, and 1211 Carlisle St., Tarentum, Pa.
MILLER, Robert T. (*A* 1927) Chief Engr., Sales Dept., •Masonite Corp., 111 W. Washington St., Chicago, and 1412 Schilling St., Chicago Heights, Ill.
MILLER, William T. (*M* 1938) Prof. Htg.-Vtg., •Purdue University, and 525 Hayes St., West Lafayette, Ind.
MILLHAM, Franklyn B. (*M* 1938) Installation Mgr., S. S. Fretz, Jr., Inc., 1902 Chestnut St., and •532 Ellet St., Philadelphia, Pa.
MILLIGAN, Donald G. (*A* 1941) Draftsman, Vtg. Dept., G. G. Sharp, 30 Church St., and •155 East 39th St., New York, N. Y.
MILLIKEN, J. H.* (*M* 1923) Repr., •American Air Filter Co., Inc., 228 N. LaSalle St., Chicago, and 1021 Ridge Court, Evanston, Ill.
MILLIS, Linn W. (*Life Member*; *M* 1918) Secy., Security Manufacturing Co., 1630 Oakland Ave., and •3534 Wabash Ave., Kansas City, Mo.
MILLS, D. M. (*A* 1940) Mgr., Houston Div., F. J. Evans Engineering Co., 3223 Milam St., and 711 W. Alabama, Houston, Tex.
MILLS, Hartzell C. (*A* 1935) Sales Engr., Minneapolis Gas Light Co., Eighth and Marquette, and •4137 Tenth Ave. S., Minneapolis, Minn.
MILNE, A. H. (*M* 1938) Dir., Dept. of Bldgs., •Protestant Board of School Commissioners, City of Montreal, 3460 McTavish St., and 4786 Grosvenor Ave., Montreal, Que., Canada.
MILWARD, Robert K. (*A* 1920) Branch Mgr., •U. S. Radiator Corp., 127 Campbell Ave., and 2441 Calvert Ave., Detroit, Mich.
MINER, H. Harvey (*A* 1940) Partner, •Miner Supply Co., 129 W. Front St., Red Bank, and 71 Silverton Ave., Little Silver, N. J.
MINKLER, William A. (*M* 1940) Asst. Sales Mgr., Young Radiator Co., and •916 W. Lawn Ave., Racine, Wis.
MIRABILE, J. James (*A* 1938) Engr., •Thomas Shipley, Inc., 161 Roosevelt Ave., and 1525 Third Ave., York, Pa.
MITCHELL, Alva E. (*M* 1939) Mech. Engr.-Htg., Office of Price Administration, U. S. Government, Fourth and D St. N.W., Washington, D. C., and •6501 Sligo Pkwy., Hyattsville, Md.
MITCHELL, A. J. (*M* 1938; *J* 1930) Vice-Pres., Air Conditioning Co., 3215 McKinney Ave., Houston, Tex.
MITCHELL, John A. (*A* 1940; *J* 1938) Owner, •Air Conditioning & Refrigeration Systems, 202 Waterloo Bldg., and Sherwood Park, Waterloo, Iowa.
MITCHELL, John G. (*J* 1937; *S* 1936) Sales Engr., •Fairbanks Morse & Co., 417 S. Fourth St., and 704 Delaware S.E., Minneapolis, Minn.
MITTENDORFF, E. M. (*M* 1932) Engr., Sarco Co., Inc., Merchandise Mart, Chicago, and •956 Greenwood Ave., Winnetka, Ill.
MODIANO, Rene (*M* 1925) Managing Dir., Carrier Continentale, 4 Rue d'Aguesseau, Paris (8s), and •55 Blvd. Beaus'jour, Paris (16s), France.
MOESEL, F. Albert (*A* 1939) Asst. Mgr., •W. A. Case & Son Manufacturing Co., 31 Main St., Buffalo, and 382 Argonne Drive, Kenmore, N.Y.
MOFFAT, Ormond George (*M* 1940; *A* 1937) Application Engr., Canadian Westinghouse Co., Ltd., Sanford Ave., and •141 George St., Hamilton, Ont., Canada.
MOHN, H. Leroy (*M* 1937) Chief Engr., Milton Manufacturing Co., and •705 Hepburn St., Milton, Pa.
MOHRFELD, Herbert H. (*J* 1935) Engr. and Vice-Pres., •C. P. Mohrfeld, Inc., 24 Lees Ave., Collingswood, and 670 Station Ave., Haddonfield, N. J.
MOHSIN, Shawki (*J* 1940) Air Cond. Engr., Egyptian Government, and •Carrier Corp., C.I.D., Syracuse, N. Y.
MOLFINO, Philip (*M* 1938) Mech. Engr., •Leland & Haley, 58 Sutter St., and 125 Clayton St., San Francisco, Calif.
MOLLANDER, Eric D. (*A* 1940) Dir., Register & Grille Manufacturing Co., Inc., 70 Berry St., Brooklyn, and •1564 Unionport Rd., Parkchester, New York, N. Y.
MOLLENBERG, Harold J. (*M* 1936) Vice-Pres., Mollenberg-Betz Machinery Co., 22 Henry St., Buffalo, and •172 Westgate Rd., Kenmore, N.Y.
MOLONEY, Roger R. (*M* 1937) 26 Bonner Ave., Manly, Sydney, Australia.
MONICK, Fred R. (*A* 1936) Mgr., •American Radiator & Standard Sanitary Corp., and 1114 S. Sixth Ave., Sioux Falls, S. D.
MONTGOMERY, Edward G. (*A* 1938) Special Repr., Steel Co. of Canada, Ltd., 525 Dominion St., Montreal, and •20 Finchley Rd., Hampstead, Que., Canada.
MONTGOMERY, John R. (*A* 1937) Mgr., Standards and Research, •Truscon Steel Co., Albert St., and 296 Granada Ave., Youngstown, Ohio.
MOODY, Lawrence E. (*M* 1919) •Moody & Hutchison, 1701 Architects Bldg., Philadelphia, Pa., and 237 Jefferson Ave., Haddonfield, N. J.
MOON, L. Walter (*M* 1915) (Council, 1933-36) Vice-Pres., St. Louis Industrial Truck Co., 7700 E. Railroad Ave., and •1137A Hornsby Ave., St. Louis, Mo.
MOORE, Bryant W. (*A* 1939) Mfrs. Agent, 36 S.W. Third St., and •7910 Southeast 30th Ave., Portland, Ore.
MOORE, Frank C. (*A* 1938) Canadian Mgr., Aerofin Corp., 67 Yonge St., and •44 Lola Rd., Toronto, Ont., Canada.
MOORE, H. Carlton* (*M* 1935) Engr.-Design, Shreve Lamb & Harmon, Fay, Spofford & Thorndyke, 11 Beacon St., Boston, and •145 Beaumont Ave., Newtonville, Mass.
MOORE, H. Lee (*M* 1919) (Council, 1927-28) Repr., •Buffalo Forge Co., 431 Fulton Bldg., and Fluccus Rd., Ben Avon, Pittsburgh, Pa.
MOORE, Henry W. (*M* 1935) Mgr., Air Cond. Engrg. Dept., The Bimel Co., 305 Walnut St., and •1406 Myrtle Ave., Cincinnati, Ohio.
MOORE, Herbert S. (*A* 1923) Dist. Repr., Iron Fireman Manufacturing Co. of Canada, Ltd., 602 King St., and •107 Clendenan Ave., Toronto, Ont., Canada.
MOORE, MacDonell (*A* 1940) Pres.-Gen. Mgr., •The Moore Fuel Corp., 23 Rose St., and 14 Fairview Ave., Danbury, Conn.

MOORE, R. E. (*A* 1928) Vice-Pres. in Charge of Sales, • Bell & Gossett Co., 8200 N. Austin Ave., Morton Grove, and 425 Merrill Ave., Park Ridge, Ill.

MOORE, Wesley Robert (*M* 1937) Branch Mgr., • Minneapolis-Honeywell Regulator Co., 4501 Prospect Ave., Cleveland, and 14211 Ashwood Rd., Shaker Heights, Ohio.

MORAWECK, Alvin H., Jr. (*J* 1941) Secy.-Sales Engr., Abbott Engineering Co., 49 Liberty St., Newark, and • 370 Central Ave., Orange, N. J.

MOREHOUSE, H. Preston (*M* 1933) Gen. Htg. and Air Cond. Repr., Public Service Electric & Gas Co., 80 Park Place, Newark, N. J.

MOREHOUSE, J. Stanley (*M* 1938) Dean of Engrg., • Villanova College, Villanova, and 102 Llandoff Rd., Upper Darby, Pa.

MORGAN, Arthur S. (*M* 1938) Mgr., Fess Oil Burners of Canada, Ltd., 85 King St. W., and • 156 Glenmanow Drive, Toronto, Ont., Canada.

MORGAN, Glenn C. (*M* 1911) Vice-Pres. and Secy., • Morgan-Gerrish Co., 307 Essex Bldg., and 4308 Fremont Ave. S., Minneapolis, Minn.

MORGAN, Robert C. (*M* 1915) Pres., • Stewart A. Jellett Co., 1200 Locust St., and 314 W. Seymour St., Philadelphia, Pa.

MORGAN, Robert W. (*M* 1938) Engr., Fedders Manufacturing Co., 57 Tonawanda St., Buffalo, and • 71 Highland Dr., Williamsville, N. Y.

MORIARTY, John M. (*M* 1937) Owner, • Consolidated Heating & Ventilating Co., 1709 W. Eighth St., Los Angeles, and 1616 Baldwin Ave., Arcadia, Calif.

MORIN, A. R. (*A* 1938) Mgr., Refrigeration Dept., Macklanburg Brass & Copper Products, Inc., 111 Northwest 23rd, and • 2115 Sherman, Oklahoma City, Okla.

MORRIS, C. Raymond (*M* 1921) Pres., Power & Heating Equipment Sales, Inc., 14 Burnett Place, Nutley, N. J.

MORRIS, John A. (*A* 1939; *J* 1936) Htg. Dept., James Robertson Co., Ltd., 946 William St., and • 4134 Marlowe Ave., Montreal, Que., Canada.

MORRISON, Chester B. (*M* 1931) Managing Dir., • York Shipley, Ltd., North Circular Rd., Hendon, London, and 40 Berkeley Sq., London, W. 1, England.

MORRISON, Walter B. (*A* 1942; *J* 1939) Engr., Htg. Dept., Meier & Frank Co., and • 1805 Northeast 27th, Portland, Ore.

MORRISON, Wayne L. (*A* 1938) Owner, • W. L. Morrison Co., 1908 Broadway St., and P.O. Box 697, Great Bend, Kan.

MORRO, John J. (*M* 1940) Engr., Paragon Oil Co., 75 Bridgewater St., Brooklyn, and • 28 East 29th St., New York, N. Y.

MORROW, J. DeWitt (*A* 1938) Secy.-Treas. and Mgr., • The Warren Co., Inc., 614 Walker Ave., and 5503 LaBranch St., Houston, Tex.

MORSE, Clark T. (*M* 1913) Pres., • American Blower Corp., 6000 Russell St., and 16222 Shaftsbury Rd., Detroit, Mich.

MORSE, Floyd W. (*A* 1934) Vice-Pres., Chamberlin Metal Weather Strip Co., 15 Oak St., and 132 Villa St., Mt. Vernon, N. Y.

MORSE, Louis S., Jr. (*M* 1938; *J* 1936) Mgr., • Spee-D Chemical Service, 415 Brainard St., Detroit, and Lone Pine Rd., Bloomfield Hills, Mich.

MORSE, Robert D. (*M* 1936) Mfrs. Repr., • 4404 White Bldg., and 4316 East 43rd St., Seattle, Wash.

MORTON, Charles H. (*A* 1931) Local Repr., • Warren Webster & Co., 228 Ottawa Ave. N.W., and 1106 Sherman St. S.E., Grand Rapids, Mich.

MORTON, Harold S. (*M* 1931) Maj., Ordnance Dept., U. S. Army, Office of the Chief of Ordnance, Washington, D. C., and • 524 Ritchie Ave., Silver Spring, Md.

MORTON, Paul S. (*J* 1939) Htg.-Plbg. Sales & Engrg., 609 Bangor Rd., Lawrence, Mich.

MOSES, Walter B., Jr. (*J* 1940; *S* 1936) Engr., Leo S. Weil & Walter B. Moses, 425 S. Peters St., and • 8330 Spruce St., New Orleans, La.

MOSHER, Clarence H. (*A* 1919) Owner, C. H. Mosher Co., 423 Ashland Ave., Buffalo, N. Y.

MOTZ, O. Wayne (*M* 1932) Consulting Engr., • 234 Paramount Bldg., and 2605 Briarcliff, Cincinnati, Ohio.

MOULD, Delmar E. (*M* 1936) Mgr., • J. W. Mould & Son, Ltd., 10642–102nd Ave., and 8619–108 A St., Edmonton, Alta., Canada.

MOULD, Harry W., Jr. (*J* 1941) Student Engr., Buffalo Forge Co., Buffalo, and • 251 Parkwood Ave., Kenmore, N. Y.

MUCKLE, James (*M* 1939) Dept. Chief, • Andersen, Meyer & Co., Ltd., 21 Yuen Ming Yuen Rd., Box 265, and 11 Park Rd., Apt. 8, Shanghai, China.

MUELLER, Harald C. (*M* 1936; *A* 1930) Mgr., • Powers Regulator Co., 2720 Greenview Ave., Chicago, and 1277 Forest Glen Dr. N., Winnetka, Ill.

MUELLER, Harold P. (*M* 1936) Pres.-Treas., • L. J. Mueller Furnace Co., 2005 W. Oklahoma Ave., and 511 E. Monrovia Ave., Milwaukee, Wis.

MUELLER, John E. (*M* 1937) Mgr. of Commercial Sales, • West Penn Power Co., 14 Wood St., Pittsburgh, and 570 Crystal Drive, Mt. Lebanon, Pa.

MUENZENMAYER, Willard R. (*A* 1941) Owner-Mgr., Muenzenmayer Sheet Metal Co., Junction City, Kan.

MUESSIG, James W. (*M* 1938) Sales Engr., • Clarage Fan Co., 333 N. Michigan Ave., Chicago, and 405 Lodge Lane, Lombard, Ill.

MUIRHEID, John G. (*A* 1940; *J* 1937) Nash-Kelvinator Corp., and • 1621 Queens Rd., Charlotte, N. C.

MULCEY, Paul A.* (*A* 1939; *J* 1938) Asst. Dir., Anthracite Industries Laboratory, Primos, Del. Co., and • 300 Springfield Rd., Aldan, Del. Co., Pa.

MULLEN, Thomas J., Jr. (*A* 1942; *J* 1935) Industrial Mgr., • B. F. Sturtevant Co., Western Div., Inc., 602 Wrigley Bldg., Chicago, Ill.

MUMFORD, Albert R. (*M* 1940) Sr. Research Assoc., Consolidated Edison Co. of New York, 4 Irving Place, New York, N. Y., and • 107 Palisade Ave., Bogota, N. J.

MUMFORD, William W. (*J* 1940; *S* 1939) Machinist, Oil Well Improvement Co., and • 1920 N. Delaware Place, Tulsa, Okla.

MUNIER, Leon L. (*M* 1919; *J* 1913) Pres., • Wolff & Munier, Inc., 222 East 41st St., New York, and 63 Columbia Ave., Hartsdale, N. Y.

MUNKELT, Frederick H. (*M* 1938) Vice-Pres., • W. B. Connor Engineering Corp., 114 East 32nd St., New York, and 317 East 17th St., Brooklyn, N. Y.

MUNN, E. Fitz (*M* 1935) • 1111 McArthur Bldg., Winnipeg, and 65 Berrydale Ave. E., St. Vital, Man., Canada.

MURDOCH, John P. (*M* 1937) Pres., • John P. Murdoch Co., S.W. Cor. 30th and Oakford Sts., Philadelphia, and 735 Beechwood Drive, Beechwood, Pa.

MURHARD, Erroll A. (*M* 1930) Pres.-Engr. (Civil and Mech.), Muirhead & Murhard Co., 338 S.W. Ninth Ave., and • 2136 N.W. Upshur St., Portland, Ore.

MURNIN, Edward A., Jr. (*A* 1938) Supt. of Development and Assembly, Sarco Manufacturing Co., Clewell and Itaska Sts., and • 802 Broadway, Bethlehem, Pa.

MURPHREE, Robert L. (*A* 1940; *J* 1936) Engr., • Rogers Plumbing & Heating Co., 2127 Eighth St., and 914–35th Ave., Tuscaloosa, Ala.

MURPHY, Daniel C. (*A* 1940) Sales Repr., • C. A. Dunham Co., 214 Old Colony Bldg., and 3900 Grand Ave., Des Moines, Iowa.

MURPHY, Delacour I. (*M* 1941) Sales Engr., E. C. Cooley Co., 625 Market St., San Francisco, and • 3027 Millsbrae Ave., Oakland, Calif.

MURPHY, Edward T.* (*M* 1915) Vice-Pres., • Carrier Corp., and 1055 James St., Syracuse, N. Y.

MURPHY, Howard C.* (*M* 1923) Vice-Pres., • American Air Filter Co., 215 Central Ave., and 495 Lightfoot Rd., Louisville, Ky.

MURPHY, Joseph R. (*M* 1934; *A* 1925) Vice-Pres., Taco Heaters, Inc., 342 Madison Ave., New York, N. Y., and • Terrace Ave., Riverside, Conn.
MURPHY, William W. (*M* 1930) Treas., • W. W. Murphy Co., 424 Worthington St., and 25 Mansfield St., Springfield, Mass.
MURRAY, H. G. S. (*A* 1941; *J* 1936) Sales Engr., • Canadian Comstock Co., Ltd., Room 2206, 80 King St. W., and 19 Elora Rd., Toronto, Ont., Canada.
MURRAY, Thomas F. (*M* 1923) State Archt., • 14 S. Lake Ave., Albany, N. Y.
MURSINNA, Gilbert P. (*A* 1939) Htg.-Air Cond. Contractor, • Gilbert P. Mursinna, 411 Poplar St., and 3057 Boudinot Ave., Cincinnati, O.
MUSGRAVE, Merrill N. (*A* 1935) Pres., Harrison Sales Co., Inc., 2019 Third Ave., and • 1005 E. Roy St., Apt. 13, Seattle, Wash.
MYER, Haydn (*A* 1920) Pres., • Haydn Myer Co., Inc., 2224 Comer Bldg., and 1411 Avon Circle, Birmingham, Ala.
MYERS, George W. F. (*M* 1930; *A* 1928; *J* 1923) Myers Engineering Equipment Co., 3736 W. Pine Blvd., St. Louis, and • 476 Pasadena Ave., Webster Groves, Mo.
MYLER, William M., Jr.* (*M* 1937) Chief Engr., • Janitrol Engrg. Dept., Surface Combustion Corp., 400 Dublin Ave., and 1340 Glenn Ave., Columbus, Ohio.
MYTINGER, Kenneth L. (*M* 1936) Sales Repr., Cooper & Cooper, Inc., Pittsfield, Mass., and • 714 Madison Ave., Albany, N. Y.

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NACHMAN, George P. (*M* 1938) Treas., • The Spohn Heating & Ventilating Co., 1775 East 45th St., and 2870 Meadowbrook Blvd., Cleveland, Ohio.
NAMAN, Israel A. (*J* 1940) Mech. Engr., Planning Div., Ventilation Group, Puget Sound Navy Yard, and • 1307–11th St., Bremerton, Wash.
NAROWETZ, Louis L., Jr. (*M* 1929; *A* 1912) Secy.-Gen. Mgr., • Narowetz Heating & Ventilating Co., 1711 Maypole Ave., Chicago, and 112 S. Northwest Highway, Park Ridge, Ill.
NASS, Arthur F. (*M* 1927) Pres., • McGinness Smith & McGinness Co., 527 First Ave., and 29 Elmhurst Rd., (Wabash Station), Pittsburgh, Pa.
NEAL, James P. (*A* 1939) Capt., Ord. Dept., • Walter Reed General Hospital, Washington, D. C., and 3282 Beredith Place, Cincinnati, Ohio.
NEARINGBURG, Arthur (*A* 1938) Sales Engr., • Sheldons, Ltd., 1221 Bay St., and 130 Floyd Ave., Toronto, Ont., Canada.
NEE, Raymond M. (*M* 1936) Steam Service Engr., • Boston Edison Co., 39 Boylston St., Boston, and 10 Orkney Rd., Brookline, Mass.
NEILER, Samuel G. (*Life Member*; *M* 1898) Owner, • Neiler Rich & Co., Cons. Mech. & Elec. Engrs., 431 S. Dearborn St., Chicago, and 737 N. Oak Park Ave., Oak Park, Ill.
NELSON, Alex A. (*A* 1942) Chief Engr., • Federal Reserve Bank Bldg., North Kansas City, R.F.D. 4, and Northcrest Addition, Clay Co., Mo.
NELSON, C. L. (*A* 1937; *J* 1929) Chief Air Cond. Engr., Sears & Piou, 814 S. Vandeventer St., St. Louis, and • 1015 Nolan Drive, Glendale, St. Louis Co., Mo.
NELSON, D. W.* (*M* 1928) Assoc. Prof. Mech. Engrg., • College of Engineering, University of Wisconsin, Mech. Engrg. Bldg., and 3906 Council Crest, Madison, Wis.
NELSON, George O. (*M* 1923) Engr., Carstens Bros., Ackley, Iowa.
NELSON, Harold M. (*M* 1937) Pres., • H. M. Nelson & Co., Inc., 1223 Connecticut Ave., Washington, D. C., and Falls Church, Va.
NELSON, Herman W. (*Life Member*; *M* 1909) Pres. and Gen. Mgr., • The Herman Nelson Corp., 1824 Third Ave., and 2615–12th St., Moline, Ill.
NELSON, Laurence K. (*M* 1940) Assoc. Engr., • James M. Todd, Consulting Engr., 624 Gravier St., and 2502 Palmer Ave., New Orleans, La.
NELSON, Richard H. (*A* 1933; *J* 1928) Secy.-Treas., The Herman Nelson Corp., 1824 Third Ave., and • 1303 30th St., Moline, Ill.
NELSON, Roy O. (*M* 1938) Sales Engr., • C. H. Bevington Co., 600 S. Michigan Ave., and 5927 N. Rockwell St., Chicago, Ill.
NESBITT, Albert J.* (*M* 1921) Secy.-Treas., • John J. Nesbitt, Inc., State Rd. and Rhawn St., Philadelphia, and Babylon and Davis Grove Rds., Hatboro, Pa.
NESMITH, Oliver E. (*A* 1928) Engr., Williams Oil-O-Matic Heating Corp., Bell and Hanna, and • 107 Warner, Bloomington, Ill.
NESS, William H. C. (*M* 1931) Gen. Mgr., • Master Fan Corp., 1323-35 Channing St., and 215 N. Kingsley Dr., Los Angeles, Calif.
NESSELL, C. W. (*M* 1937) Supvr., Federal Projects Div., • Minneapolis-Honeywell Regulator Co., 2753 Fourth Ave. S., Minneapolis, Minn.
NESSI, Andre (*M* 1930) Ingr. des Arts et Mfrs., Expert pres le Tribunal Civil de la Seine, 1 Avenue du President Wilson, Paris, XVI, France.
NEST, Richard E. (*M* 1936) Consultant, 5018 Morello Rd., Baltimore, Md.
NEUBAUER, Edwin W. (*M* 1939) Engr., • Cambell Norquist & Co., 1127 S. W. Morrison St., and 4804 N. E. Davis St., Portland, Ore.
NEWBY, Ira P. (*M* 1941) Charge of Htg. Sales., • American Radiator & Standard Sanitary Corp., 1512 Santa Fe Bldg., and 6325 Bryan Parkway, Dallas, Tex.
NEWMAN, Harold E. (*M* 1938) Asst. Mgr., • B. A. Newman Co., 320 North H St., and 419 Buckingham Way, Fresno, Calif.
NEWPORT, Charles F.* (*Life Member*; *M* 1906) Sales Engr., Weil-McLain Co., Michigan City, Ind., and • 10001 Longwood Dr., Chicago, Ill.
NEWTON, Alwin B.* (*M* 1938) Mgr. Refrigeration Controls Div., • Minneapolis-Honeywell Regulator Co., 2747 Fourth Ave. S., and 18 W. Rustic Lodge Ave., Minneapolis, Minn.
NICHOLLS, John M. (*M* 1939) • Robbins Gamwell Corp., 68 West St., and 136 Bartlett Ave., Pittsfield, Mass.
NICHOLLS, Percy* (*Life Member*; *M* 1920) Supervising Engr., Fuels Section, • U. S. Bureau of Mines, 4800 Forbes St., and 5251 Forbes St., Pittsburgh, Pa.
NICHOLSON, Sterling J. (*A* 1941) Pres., Nicholson, Inc., Box 317, Durham, N. C.
NICKLE, Arthur J. (*A* 1936) Sales Engr., • Darling Brothers, Ltd., 140 Prince St., and 4356 Marcil Ave., Montreal, Que., Canada.
NICOLL, Scott F.* (*M* 1939) Air Cond. Application Engr., • York Ice Machinery Corp., York, and 1433 First Ave., Elmwood, York, Pa.
NICOLS, J. A. (*M* 1941) Dist. Mgr., • B. F. Sturtevant Co., 1217 McKnight Bldg., and 3915 S. Tyrol Trail, Tyrol Hills, Minneapolis, Minn.
NIELSEN, Howard B. (*A* 1939) Mfrs. Repr., • 409 Couch Bldg., Portland, and Jennings Lodge, Ore.
NIESSE, Joseph H. (*M* 1938) Branch Mgr., Ilg Electric Ventilating Co., 836 Architects & Builders Bldg., and • 5837 Winthrop Ave., Indianapolis, Ind.
NIGHTINGALE, George F. (*A* 1931) Western Sales Mgr., • Tuttle & Bailey, Inc., 61 W. Kinzie St., Chicago, and 2136 W. Linneman St., Glenview, Ill.
NININGER, Christian H. (*A* 1938) Sales Engr., • Frigidaire Div., General Motors Sales Corp., 29 Franklin Rd., and Route 1, Box 198, Roanoke, Va.
NOBBS, Walter W. (*M* 1919) Consulting Engr., 26 Victoria St., London S.W. 1, and • 50 Fairhazel Gardens, London N.W. 6, England.
NOBIS, H. M. (*M* 1914) Owner, • Nobis Heater Co., 1935 Union Commerce Bldg., Cleveland, and 1827 Stanwood Rd., East Cleveland, Ohio.
NOBLE, James P. (*A* 1937) 1960 Beverly Rd., Columbus, Ohio.

NOBLE, Milner (*M* 1940; *A* 1929; *J* 1924) Gen. Mgr., Aerofin Corp., 410 S. Geddes, Syracuse, N. Y.
NOLAN, James J., Jr. (*M* 1939) Sr. Mech. Engr., War Dept., Office of Quartermaster General, and • 4024 Calvert St., N.W., Washington, D. C.
NOLL, William F. (*M* 1924) Prop., • Wm. F. Noll, 629 North 27th St., and 4823 W. Townsend St., Milwaukee, Wis.
NOORD, Donald F. (*J* 1941; *S* 1935) Mech. Engr., Giffels & Vallet, Arcade Bldg., St. Louis, and • 1361 McCutcheon, Richmond Heights, Mo.
NORAIR, Henry (*M* 1938) Pres., • Norair Engineering Corp., 1114–22nd St. N. W., and 5908 32nd St. N.W., Washington, D. C.
NORBY, Karl H. (*A* 1938) Mgr. Htg. Dept., • Tacoma Plumbing Supply Co., 315 South 23rd St., Tacoma, and Box 113, Parkland, Wash.
NORDINE, L. F. (*M* 1914) Office Mgr., The Trane Co., 2701 Ontario Rd. N.W., Washington, D. C., and • 812 Silver Springs Ave., Silver Spring, Md.
NORFOLK, Leslie W. (*A* 1941; *J* 1939) Consulting Engr., 42 Hampden St., Nottingham, England.
NORMAN, Roy A. (*M* 1937) Prof. of Mech. Engrg., Iowa State College, Mech Engrg. Dept., and • 715 Ridgewood Ave., Ames, Iowa.
NORRINGTON, Walter L. (*J* 1938) 1st Lt. Ordnance Dept., Office of Chief of Ordnance, Washington, D. C., and • 305 Barbara Fritchie, Beverly Plaza, Alexandria, Va.
NORRIS, William P. (*J* 1938) Sales Engr., Natkin & Co., 3920 Lindell Blvd., and • 410 N. Newstead Ave., St. Louis, Mo.
NORTH, William R. (*A* 1940) Sales Engr., • 27 S. Gay St., and 1521 Kingsway Rd., Baltimore, Md.
NORTON, John A. (*M* 1940) Mgr. Htg. Sales Div., Crane, Ltd., 306 Front St. W., Tortonto, and • 71 Donegall Dr., Leaside, Ont., Canada.
NORTON, L. Ivan (*A* 1941) Engr., • Eveready Plumbing & Heating Co., 211 S. Iowa Ave., and 110 E. Polk St., Washington, Iowa.
NOTTBERG, Gustav (*A* 1933) Vice-Pres., • U. S. Engineering Co., 914 Campbell St., and 1835 East 68th Terrace, Kansas City, Mo.
NOTTBERG, Henry J. (*M* 1919) Pres., • U. S. Engineering Co., 914 Campbell St., and 150 West 54th St., Kansas City, Mo.
NOTTBERG, Henry, Jr. (*J* 1937) Secy., • U. S. Engineering Co., 914 Campbell St., and 150 West 54th St., Kansas City, Mo.
NOVOTNEY, Thomas A. (*M* 1928) S. K. Wellman Co., 1381 East 49th St., and • University Club, Cleveland, Ohio.
NOWITZKY, Herman S. (*A* 1931) Supt. Construction Corps., Wilmer & Vincent, 1776 Broadway, New York, N. Y., and • 821 Llewellyn Ave., Norfolk, Va.
NOYES, Richard R. (*J* 1938) Sales Engr., • Canadian Sirocco Co., Ltd., 630 Dorchester St. W., and 2010 Mansfield, Apt. 12, Montreal, Que., Canada.
NUSBAUM, Lee* (*M* 1915) Owner, • Pennsylvania Engineering Co., 1119-21 N. Howard St., Philadelphia, and 315 Carpenter Lane, Germantown, Philadelphia, Pa.
NUSBAUM, S. Richard (*J* 1940) Mgr., • Pennsylvania Engineering Co., 1119 N. Howard St., Philadelphia, and 552 Montgomery Ave., Haverford, Pa.
NUTTING, Arthur* (*M* 1940) Chief Engr., • American Air Filter Co., 215 Central Ave., and 504 Wallace Ave., Louisville, Ky.
NYE, L. Bert, Jr. (*J* 1936) Laboratory Supvr., Washington Gas Light Co., 411 Tenth St., Washington, D. C., and • McLean, Va.
NYQUIST, John D. (*J* 1942; *S* 1941) Jr. Mech. Engr., Collins Radio Co., 35th St., and • 115–25th St. Drive S.E., Cedar Rapids, Iowa.

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OAKLEY, LeRoy W. (*M* 1937) Owner, • L. W. Oakley Sales Co., 408 W. Clinch Ave., and 2003 Laurel Ave., Knoxville, Tenn.
OAKS Orion O. (*M* 1917) Consulting Mech. Engr., 511 Fifth Ave., New York, N. Y., and • 119 Oakridge Ave., Summit, N. J.
O'BANNON, Lester S.* (*M* 1928) Research Engr., • Agricultural Experiment Station, University of Kentucky, and 123 State St., Lexington, Ky.
OBERG, H. C. (*A* 1933) Mgr. Engrg. Dept., Crane Co., Fifth and Broadway, and • 1362 W. Minnehaha St., St. Paul, Minn.
OBERLIN, James A. (*J* 1942; *S* 1940) Engr., • Allied Products Corp., Plant 3, and 211 Union St., Hillsdale, Mich.
OBERSCHULTE, Richard H. (*J* 1938) Sales Engr., • D. T. Randall & Co., 404 Blvd. Bldg., Detroit, and Box 76, Franklin, Mich.
O'DOWER, Hugh J. (*A* 1938) Sales Engr., Vilter Manufacturing Co., 400 No. 114 W. Tenth St., Kansas City, Mo.
ODUM, Ralph A. (*A* 1939) Chief Heating Inspector, Southern Engineering & Architectural Co., Panama City, and • 218 Broadway, Daytona Beach, Fla.
OELGOETZ, J. F. (*M* 1938) Owner, • J. F. Oelgoetz Co., 3365 N. High St., and 279 E. North Broadway, Columbus, Ohio.
OERTEL, Fritz H. E. (*M* 1939) Consulting Engr., R. F. D. 2, Greensboro, N. C.
OESTERLE, A. L. (*M* 1940) Engr., • Gulf Engineering Co., Inc., 916 S. Peters St., and 3420 Live Oak Pl., New Orleans, La.
OFFEN, Ben (*M* 1928) • B. Offen & Co., 608 S. Dearborn St., and 3740 Lake Shore Dr., Chicago, Ill.
OFFNER, Alfred J. *(*M* 1922), (Treasurer, 1935-38; Council, 1935-41), Consulting Engr., • 139 East 53rd St., New York, and 160-15 11th Ave., Beechhurst, L. I., N. Y.
O'FLAHERTY, J. G. (*M* 1937) Chief Engr., • Unifin Tube Co., 1109 York St., and 288 Central Ave., London, Ont., Canada.
O'GORMAN, J. S., Jr. (*A* 1934) Branch Mgr., • Johnson Service Co., 230 E. Alexandrine Ave., Detroit, and 147 Abbey Rd., Birmingham, Mich.
OLD, William H. (*M* 1937) Asst. Mgr., • Glanz & Killian, 1761 Forest Ave. W., Detroit, and 18245 Devonshire Rd., R. F. D. No. 4, Birmingham, Mich.
OLDES, Willard E. A. E. (*A* 1939; *J* 1936) 650 West 204th St., New York, N. Y.
OLSEN, Carlton F. (*A* 1925; *J* 1920) Engr. and Sales, Kewanee Boiler Corp., 1858 S. Western Ave., and • 1000 West 100th St., Chicago, Ill.
OLSEN, Gustav E. (*M* 1930) Sales Mgr., Fitzgibbons Boiler Co., Inc., 101 Park Ave., New York, and • 68-09 Beach Channel Dr., Arverne, L. I., N. Y.
OLSON, Barney (*A* 1929) • Barney Olson, Inc., 122 S. Michigan Ave., and 5724 N. Natoma Ave., Chicago, Ill.
OLSON, Gilbert E. (*M* 1930) Southern Mgr., • H. K. Ferguson Co., 1605 Commerce Bldg., and 8359 Park Place Blvd., Houston, Tex.
OLSON, Milton J. (*A* 1941; *J* 1937) Vice-Pres., Olson Bros., 2651-55 St. Mary's Ave., and • 5627 Williams St., Omaha, Nebr.
OLSON, Robert G. (*M* 1923) Eastern Mgr., • Hydraulic Coupling Div., American Blower Corp., 50 West 40th St., and 22 East 38th St., New York, N. Y.
OLVANY, William J. (*M* 1912) Pres., Wm. J. Olvany, Inc., 100 Charles St., New York, N. Y.
O'NEILL, James W. (*M* 1929; *A* 1927; *J* 1923) Chief Engr., Trane Co. of Canada, Ltd., 4 Mowat Ave., and • 53 Highview Crescent, Toronto, Ont., Canada.
OONK, W. J. (*M* 1937) Dist. Mgr., B. F. Sturtevant Co., 915 Olive St., and • 4548 Red Bud Ave., St. Louis, Mo.
OOSTEN, Louis S. (*J* 1938) Engr., Bell & Gossett Co., 8200 Austin Ave., Morton Grove, and • 1827 Berwyn Ave., Chicago, Ill.
O'REAR, L. R. (*M* 1934) Pres., • Midwest Plumbing & Heating Co., 2450 Blake St., and 825 S. Josephine St., Denver, Colo.

ORGELMAN, George H. (*J* 1942; *S* 1940) Construction Engr., Turner Construction Co., Newberry St., Boston, Mass., and • 10 Pearl St., Danbury, Conn.
ORMISTON, John B. (*A* 1940) Owner, • Ormiston Plumbing & Heating Co., 105 Manning Ave., and 1 Gramatan Drive, Yonkers, N. Y.
O'ROURKE, Hugh D., Jr. (*J* 1937; *S* 1936) Asst. Mgr., Unit Heater Dept., McCord Radiator & Manufacturing Co., 2587 E. Grand Blvd., and • 14535 Strathmoor Ave., Detroit, Mich.
ORR, George M. (*M* 1936) Pres., • G. M. Orr & Co., 542 Baker Arcade Bldg., and 1100 West 53rd St., Minneapolis, Minn.
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READER, Joseph T. (*A* 1938) Partner, •Kerr Machinery Co., 608 Kerr Bldg., and 8162 E. Jefferson Ave., Detroit, Mich.

REARDON, J. F. (*A* 1941) Contract Sales, Powers Regulator Co., 2341 Carnegie, Cleveland, and •26981 Mallard Ave., Euclid, Ohio.

RECK, William E. (*M* 1927) Civil Engr., •The Reck Heating Co., Ltd., Esromgade 15, Copenhagen, and Sundvej 16, Hellerup, Denmark.

REDRUP, Will D. (*M* 1936) Pres., •The Majestic Co., and 310 Randolph St., Huntington, Ind.

REDSTONE, A. L. (*M* 1931) Research Engr., Proctor & Schwartz, Seventh and Tabor Rd., and •1636 E. Duval, Philadelphia, Pa.

REED, Frederick J. (*M* 1939) Asst. Prof. Mech. Engrg., •Duke University, 263 College Station, and 2203 Englewood Ave., Durham, N. C.

REED, Van A., Jr. (*M* 1930) Mech. Engr., •Federal Engineering Co., 239 Fourth Ave., Pittsburgh, and 114 Water St., Elizabeth, Pa.

REED, Virgil C. (*M* 1938) Owner, •James H. Pinkerton Co., 640 Natoma St., and 1234 Second Ave., San Francisco, Calif.

REED, William H., III (*A* 1938) Sales Mgr., •Air Cond. Equipment, Dravo Corp., 302 Penn Ave., and 7458 Penyield Pl., Pittsburgh, Pa.

REESE, Henry L. (*M* 1923) Consulting Engr., •Reese & Co., 940 Spruce St., Reading, and R. D. 4, Coudersport, Pa.

REGER, Henry P. (*M* 1934) Pres., •H. P. Reger & Co., 1501 East 72nd Pl., and 6939 Bennett Ave., Chicago, Ill.

REICH, Julius G. (*A* 1941) Chief Engr., Sales Engr., •The Holland Furnace Co., 12700 Superior Ave., and 10703 Lee Ave., Cleveland, Ohio.

REID, Henry P. (*M* 1931; *A* 1927) Operating Engr., Universal Atlas Cement Co., 135 East 42nd St., Chrysler Bldg., New York, N. Y.

REID, Herbert F. (*A* 1932) Partner, Htg. Dept., Reid-Graff Co., 1417 Peck St., Muskegon Heights, Mich.

REIF, Allan F. (*M* 1937) Pres., •Reif-Rexoil, Inc., 37-43 Carroll St., Buffalo, and 10 Livingston Pkwy., Snyder, N. Y.

REIF, Charles A. (*M* 1937) Vice-Pres., •Reif-Rexoil, Inc., 37 Carroll St., Buffalo, and 77 Ruskin Rd., Eggertsville, N. Y.

REIFSCHNEIDER, Jake (*A* 1938) Chief Engr., •Eppley Hotels Co., 1802 Dodge St., and 4409 Poppleton, Omaha, Nebr.

REILLY, Bertram B. (*J* 1938) Engr., •Dravo Corp., 300 Penn Ave., Pittsburgh, and 220 Ridge Ave., Ben Avon, Pa.

REILLY, Philip Harry, Jr. (*A* 1941) Production Mgr., •Heating Equipment Co., 1123 Harrison St., San Francisco, and 1326 Columbus Ave., Burlingame, Calif.

REINKE, Alfred G. (*A* 1940; *J* 1933) Secy., Gus Reinke Machinery & Tool Co., 63 Dickerson St., Newark, and •321 Park Pl., Irvington, N. J.

REINKE, Louis F. (*A* 1937) Owner, •Reinke Sheet Metal Works, 534-38 S. Fifth St., and 1535 W. Walker St., Milwaukee, Wis.

REIS, Robert (*J* 1939) Mech. Engr., •Wigton Abbott Corp., 1225 South Ave., and 713 Watchung Ave., Plainfield, N. J.

REISBERG, Lester K. (*A* 1939) Vice-Pres., •Goodin Co., 707 N. Third St., and 2724 N.E. Hayes St., Minneapolis, Minn.

RENOUF, Edward Prince (*M* 1933) Air Cond. Supvr., Westinghouse Electric & Manufacturing Co., 1005 Insurance Bldg., and •9440 Waterview Rd., Dallas, Tex.

REPP, Harry L. (*M* 1922) Branch Mgr., •U. S. Radiator Corp., 3221 Carnegie Ave., and 15612 Braemar Dr., Cleveland, Ohio.

RESCH, Roy J. (*A* 1940) Pres., •McQuay, Inc., 1600 Broadway N.E., and Minneapolis Athletic Club, Minneapolis, Minn.

RETTEW, Harvey F. (*M* 1929) Chief Engr., Board of Education, School Dist. of Philadelphia, 21st and Parkway, and •4413 Walnut St., Philadelphia, Pa.

REYNOLDS, Thurlow W. (*M* 1922) Consulting Engr., 100 Pinecrest Dr., Hastings-on-Hudson, N. Y.

REYNOLDS, W. V. (*A* 1928) Pres., •Walter Reynolds, Inc., 861 Third Ave., and 444 East 52nd St., New York, N. Y.

RHINE, George R. (*A* 1938) Capt., Ordnance Dept., c/o San Antonio Arsenal, San Antonio, and • 1804 Vincent, Brownwood, Tex.
RHOTON, W. R. (*M* 1936) Pres., The W. R. Rhoton Co., 5915 Bonna Ave., Cleveland, and • 1728 Lee Rd., Cleveland Heights, Ohio.
RICE, C. J. (*A* 1923) Pres., Sterling, Inc., 3738 N. Holton St., and • Route 6, Box 374, Milwaukee, Wis.
RICE, Robert B. (*M* 1934) Prof. of Experimental Engrg., • North Carolina State College, North Carolina State College Station, and 2502 White Oak Rd., Raleigh, N. C.
RICHARD, Edwin J. (*M* 1933) Owner, • Richard Equipment Co., 2137 Reading Road, and 3147 Victoria Ave., Cincinnati, Ohio.
RICHARDS, Guy H. (*A* 1939) Mgr. Htg. Dept., Crane Co., 2 South 20th St., and • 7230 First Ave. S., Birmingham, Ala.
RICHARDS, Leslie V. (*M* 1941; *A* 1940) Owner-Mgr., • Richards Oil Burner Sales & Service, 97 Lawrence St., Malden, Mass.
RICHARDSON, Henry G. (*M* 1934) Pres., Williams-Richardson Co., and • 1433 Harvard Ave., Salt Lake City, Utah.
RICHARDSON, Laurence S. (*A* 1939) Mgr.-Engr., Htg. Dept., Stark-Davis Co., 110 S. W. Front St., and • 6929 S.E. Yamhill St., Portland, Ore.
RICHARDSON, R. Donald (*J* 1938) Active Service, Bombardier Royal Regiment Artillery, and • 85 Silhill Hall Rd., Solihull, Birmingham, England.
RICHFIELD, Nicholas H. (*M* 1937) Engr., Oil Burner Div., American Radiator & Standard Sanitary Corp., Bessemer Bldg., and • 1419 Milan Ave., Brookline, Pittsburgh, Pa.
RICHTMANN, William M.* (*A* 1932; *J* 1926) Assoc. Prof. of Mech. Engrg., • Texas College of Arts & Industries, and 600 W. Richards St., Kingsville, Tex.
RIDLEY, Walter H. (*M* 1939) Mgr., Dryer Div., Riggs & Lombard, Inc., Suffolk St., Lowell, and • 22 Westford St., Chelmsford, Mass.
RIES, Lester S. (*M* 1929) Supt., Dept. of Bldgs. and Grounds, • Oberlin College, 32 E. College St., and 291 Oak St., Oberlin, Ohio.
RIESECK, Wilbert L. (*S* 1941) Student, Carnegie Institute of Technology, Schenley Park, and • 3763 East St., Pittsburgh, Pa.
RIESMEYER, Edward H., Jr. (*A* 1936; *J* 1930) Htg. Engr., • Schaffer Heating Co., 231-33 Water St., and 4702 Stanton Ave., Pittsburgh, Pa.
RIETZ, Elmer W.* (*M* 1923) Mgr., Specialty Div., • The Powers Regulator Co., 2720 Greenview Ave., Chicago, and 2250 S. Sheridan Rd., Highland Park, Ill.
RISLEY, George H. (*A* 1941) Branch Mgr., • Cole Sullivan Engineering Co., 801 S.W. Stark St., and 1613 N. Benton Ave., Portland, Ore.
RITCHIE, A. G. (*M* 1933) Pres., • John Ritchie, Ltd., 102 Adelaide St. E., and 41 Garfield Ave., Toronto, Ont., Canada.
RITCHIE, Edmund J. (*M* 1923) Vice-Pres., Sales, Sarco Co., Inc., 475 Fifth Ave., New York, and • 2 Grace Court, Brooklyn, N. Y.
RITCHIE, William (*Life Member*; *M* 1909) 17 Van Reipen Ave., Jersey City, N. J.
RITTELMEYER, John M. (*M* 1941) Pres., • Rittelmeyer & Co., 179 Whitehall St. S.W., 2332 Woodward Way, Atlanta, Ga.
RITTENHOUSE, Owen R. (*S* 1941) Student, • Carnegie Institute of Technology, 5017 Forbes St., Pittsburgh, and 440 Richhill St., Waynesburg, Pa.
RITTER, Arthur (*M* 1911) Mgr., New York Office, • American Blower Corp., 50 West 40th St., New York, and 29 Edgemont Rd., Scarsdale, N. Y.
RIVARD, M. M. (*M* 1935) Mgr., Rivard Sales Co., 4550 Main St., and • 1805 West 49th St. Terrace, Kansas City, Mo.
ROACH, Edwin R. (*A* 1941) Htg.-Vtg. Engr., Baskervill & Son, Box 157, Fort Belvoir, and • 724 St. Asaph St., Apt. 108, Alexandria, Va.
ROBB, Joseph E. (*A* 1936) Sales Engr., Minneapolis-Honeywell Regulator Co., Minneapolis, Minn., and • 1717 Illinois St., Lawrence, Kan.
ROBERTS, Harry H. (*J* 1941) Air Cond. Engr., • Arkansas Natural Gas Corp., P. O. Box 1734, and Apt. D 30, 1901 Centenary Blvd., Shreveport, La.
ROBERTS, Henry L. (*M* 1916) • 228 North 16th St., Philadelphia, and 1014 Allston Rd., Brookline, Del. Co., (Upper Darby P. O.), Pa.
ROBERTS, Henry P. (*A* 1936) Secy., • Roberts-Hamilton Co., 713 S. Third St., and 3142 W. Lake Calhoun Blvd., Minneapolis, Minn.
ROBERTSON, J. A. M. (*A* 1936) Vice-Pres., • The James Robertson Co., Ltd., 946 William St., Montreal, and 109 Sunnyside Ave., Westmount, Que., Canada.
ROBIN, Richard C. (*A* 1941) Prop., • 700 Commonwealth Ave., Boston, and Hartford St., R. F. D. 1, Framingham, Mass.
ROBINSON, Arthur S. (*M* 1936) Engr., E. I. duPont deNemours Co., Wilmington, Del., and • 730 Ogden Ave., Swarthmore, Pa.
ROBINSON, Donald M. (*A* 1936) Sales Engr., • Buffalo Forge Co., 512 Woodward Bldg., Washington, D. C., and 5509 Lambeth Rd., Bethesda, Md.
ROBINSON, Edgar R. (*A* 1938) Sr. Draftsman, Vtg.-Htg., New York Shipbuilding Corp., Camden, and • 216 Carlton Ave., Westmont, N. J.
ROBINSON, George L. (*A* 1935) Designer, E. I. duPont deNemours Co., and • 136 West 34th St., Wilmington, Del.
ROBINSON, Jack A. (*A* 1940; *J* 1936) Air Cond. Engr., Australian Gas Light Co., Parker St., Box 481 AA GPO, Sydney, and • 595 New South Head Rd., Rose Bay, N.S.W., Australia.
ROBINSON, Kenneth E. (*J* 1941) Chief Engr., McLouth Air Conditioning Corp., 2400 E. Michigan Ave., and • 211 Smith Ave., Lansing, Mich.
ROCK, George A. (*M* 1937) Chief Inspector, • U. S. Navy, Cunningham Field, Cherry Point, and 30 S. Front St., New Bern, N. C.
ROCKWELL, Theodore F. (*M* 1933; *J* 1932) Asst. Prof. Mech. Engrg., • Carnegie Institute of Technology, Schenley Park, Pittsburgh, and Glenover Place, Aspinwall, Pittsburgh, Pa.
RODEE, E. J.* (*M* 1936) Office of Supvr. of Shipbuilding, U. S. Navy, Bethlehem Steel Corp., and • 18 Samoset Ave., Quincy, Mass.
RODEFFER, Edgar W. (*A* 1941) Engr., Supt., Burke & Son Co., 11225 Robertson Blvd., and • 668 West 70th St., Los Angeles, Calif.
RODENHEISER, George B. (*M* 1933) Asst. Dir., • David Ranken, Jr., School of Mech. Trades, 4431 Finney Ave., and 3639a Dover Pl., St. Louis, Mo.
RODGERS, Frederick A. (*A* 1934) Pres., Rodgers Engineering Co., 204 Thomas Bldg., and • 3913 Amherst, Dallas, Tex.
RODGERS, Joseph S. (*A* 1937; *J* 1934) Asst. Mech. Engr., Naval Academy, U. S. Navy, Annapolis, and • 16 Fourth Ave. S., Glen Burnie, Md.
RODMAN, R. W. (*M* 1922) Supt. of Custodians, • Board of Education, City of New York, 110 Livingston St., Brooklyn, and 175 West 73rd St., New York, N. Y.
ROEBUCK, William, Jr. (*M* 1917) Mfrs. Agent, • 220 Delaware Ave., and 1240 Delaware Ave., Buffalo, N. Y.
ROEDER, Winfield (*M* 1941) Branch Mgr., • American Blower Corp., 185 Church St., and 484 Whitney Ave., New Haven, Conn.
ROGERS, Charles S. (*A* 1940) Htg. Engr., L. P. Steuart & Bros., Inc., 138-12th St. N.E., Washington, D. C., and • 304 Maple Ave., Falls Church, Va.
ROGERS, Robert C. (*A* 1937) House Htg. Engr., Community Natural Gas Co., and • 5722 Worth, Dallas, Tex.
ROGERS, Thomas L. C. (*A* 1939) 158 Victoria St., London, Ont., Canada.

ROLLAND, Sverre L. (*A* 1934) Design Engr., • Oklahoma Gas & Electric Co., 321 N. Harvey, and 2207 Northwest 19th St., Oklahoma City, Okla.

RONSICK, Edward H. (*M* 1937) Supvr., Gas Heating Sales, • St. Louis County Gas Co., 231 W. Lockwood Ave., Webster Groves, and 7739 Stanford, University City, Mo.

ROOT, Edwin B. (*M* 1936) Superior Safety Furnace Pipe Co., 5816-44 Forsythe Ave., Detroit, and • 964 Pierce St., Birmingham, Mich.

ROPER, Richard F. (*A* 1940) Pres., Pleasantaire Corp., Tower Bldg., and • 2700 Tilden St. N.W., Washington, D. C.

ROSAS, Milton L. (*J* 1941) Sales Engr., • 514 Wesley Temple Bldg., Minneapolis, and 139 E. Congress, St. Paul, Minn.

ROSE, Harold J.* (*M* 1938) Sr. Industrial Fellow, • Mellon Institute, 4400 Fifth Ave., and 219 Lytton Ave., Pittsburgh, Pa.

ROSE, Howard J. (*M* 1934) Treas., Gen. Mgr., Suburban Air Conditioning Corp., 10 Brookdale Pl., Mt. Vernon, and • 100 Siebrecht Pl., New Rochelle, N. Y.

ROSE, Jerome C. (*M* 1937) Air Cond. Engr., Buensod-Stacey Air Conditioning, Inc., 60 East 42nd St., New York, and • 8031-213th St., Hollis, L. I., N. Y.

ROSEBROUGH, J. Stoddard (*A* 1937) Sales, L. J. Mueller Furnace Co., 4246 Forest Park, and • 5937 McPherson Ave., St. Louis, Mo.

ROSEBROUGH, Robert M. (*M* 1920) Branch Mgr., L. J. Mueller Furnace Co., 4246 Forest Park Blvd., St. Louis, and • 519 S. Gore Ave., Webster Groves, Mo.

ROSEBY, Thomas A. (*M* 1939) Chief Designing Engr., • Carrier Australasia, Ltd., 36-40 Bourke St., Sydney, N.S.W., and 11 Gillies St., North Sydney, N.S.W., Australia.

ROSELL, Axel F. (*M* 1935) Civil Engr., • A.-B. Svenska Flaktfabriken, Kungsgatan 18, and Kammakaregatan 25, Stockholm, Sweden.

ROSEN, Edmond J. (*A* 1939) Vice-Pres., • Parker Building Specialties, Inc., 991 Bryant St., San Francisco, and 3701 Redding St., Oakland, Calif.

ROSENBERG, Irwin (*J* 1942; *S* 1939) Jr. Naval Archt., Norfolk Navy Yard, and • 713 High St., Apt. 2, Portsmouth, Va.

ROSENBERG, Philip (*A* 1928) Secy.-Treas., Universal Fixture Corp., 135 West 23rd St., and • 250 West 104th St., New York, N. Y.

ROSENBLATT, Arthur M. (*M* 1938) Pres., • Rosenblatt & Hunt, Inc., P. O. Box 828, and 1250 Edgewood Dr., Charleston, W. Va.

ROSS, David S. (*S* 1941) Student, • Carnegie Institute of Technology, 4929 Forbes St., Pittsburgh, Pa., and 1131 West 22nd St., Lorain, Ohio.

ROSS, J. D. (*A* 1937) Sales, • Railway & Engineering Specialties, Ltd., 417 St. Peter St., and 4376 Earnscliffe Ave., Montreal, Que., Canada.

ROSS, John O.* (*M* 1920) Pres., • Ross Industries Corp., 350 Madison Ave., New York, and Lake George, N. Y.

ROSS, Roderick (*M* 1937) Consulting Engr., • Nicholas Bldg., 37 Swanston St., P. O. Box 1381 M, Melbourne, C. 1, and 5 Burns St., Elwood, Melbourne S. 3, Australia.

ROSSITER, I. J. (*A* 1939) Lt. U. S. Army, • Chanute Field, Rantoul, Ill., and 439 Harrison St., West Lafayette, Ind.

ROTH, Charles F. (*A* 1930) Pres., • International Exposition Co., Grand Central Palace, and 141 East 36th St., New York, N. Y.

ROTH, Harold R. (*M* 1935) Sales Engr., • Canadian Sirocco Co., Ltd., 57 Bloor St. W., and 5 Castleview, Toronto, Ont., Canada.

ROTHMANN, S. C. (*M* 1936) Chief Engr., • West Virginia Workmen's Compensation Commission, State Capitol Bldg., and 2008A Kanawha Blvd. E., Charleston, W. Va.

ROTTMAYER, Samuel I. (*A* 1933; *J* 1928) Mech. Engr., • Samuel R. Lewis, Cons. Engr., 407 S. Dearborn St., and 8222 S. Perry Ave., Chicago, Ill.

ROWE, William A.* (*M* 1921), (Council, 1929-31) Mech. Engr., • The Trane Co., LaCrosse, Wis., and 718 Longfellow Ave., Detroit, Mich.

ROWE, William M. (*J* 1936) Sales, • American Blower Corp., 1302 Swetland Bldg., Cleveland, and 151 Bradley Ave., Chagrin Falls, Ohio.

ROWLEY, Frank B.* (*M* 1918), (*Presidential Member*), (Pres., 1932; 1st Vice-Pres., 1931; 2nd Vice-Pres., 1930; Council, 1927-33), Prof. Mech. Engrg. and Dir. of Engineering Experiment Station, University of Minnesota, and • 4801 E. Lake Harriet Blvd., Minneapolis, Minn.

ROWLEY, Robert K. (*S* 1941) Student, University of Minnesota, and • 4801 E. Lake Harriet Blvd., Minneapolis, Minn.

ROY, Arthur C. (*A* 1937) Eastern Sales Mgr., Quaker Manufacturing Co., 617 Beverly Rd., Teaneck, N. J.

ROY, Leo (*A* 1937) Power Sales Engr., • Quebec Power Co., 229 St. Joseph St., and 41 Laurentide Ave., Quebec City, P. Q., Canada.

ROYER, Earl B. (*M* 1928) Mech. Engr., Fosdick & Hilmer, Consulting Engrs., 1703 Union Trust Bldg., and • 6635 Iris Ave., Cincinnati, Ohio.

RUDD, Dann J. (*M* 1937) Mech. Designer, Caribbean-Architect Engineer, 41 East 42nd St., New York, and • 367 Deer Park Ave., Babylon, L. I., N. Y.

RUEMMELE, Albert M. (*J* 1938) 1st Lt., Ordnance Section PRGD, Fort Buchanan, Puerto Rico; (after Feb. 1, 1942), Ashley, N. D.

RUFF, Adolph G. (*M* 1935) Supt. of Power, U. S. Playing Card Co., Park Ave., Norwood, and • 3824 Woodford Rd., Cincinnati, Ohio.

RUFF, D. C. (*M* 1922) Treas., • Healy-Ruff Co., 2255 University Ave., and 2211 St. Clair Ave., St. Paul, Minn.

RUGART, Karl (*A* 1924) Mfrs. Dist. Repr., • 26 South 20th St., Philadelphia, and 612 Bryn Mawr Ave., Narbeth P. O., Penn Valley, Pa.

RUGGLES, Robert F. (*M* 1936; *A* 1927; *J* 1926) Gen. Sales Mgr., • Autovent Fan & Blower Co., 1805 N. Kostner Ave., and 7041 Branch, N. Ozark, Edison Park, Chicago, Ill.

RUMBOLD, Allan H. (*M* 1941) Branch Mgr., • Clow Gasteam Heating Co., 427 Peachtree St., Atlanta, and Norcross, Ga.

RUMMEL, Adolph J.* (*M* 1937) Asst. Mgr., Commercial Dept., • San Antonio Public Service Co., 201 N. St. Mary's St., and 235 North Dr., San Antonio, Tex.

RUMSEY, John L. (*M* 1941) Chief Mech. Engr., Cory Joslin & Macusons, Utah Ordnance Plant, Ordnance, and • 1000 State St., Salt Lake City, Utah.

RUNGE, Carl H. (*M* 1941; *A* 1940) Consulting Engr., 944 E. Sylvan Ave., Milwaukee, Wis.

RUPPERT, C. Farrell (*S* 1939) Student, Purdue University, and • 400 Northwestern Ave., West Lafayette, Ind.

RUSSELL, Boyd A. (*A* 1941) Partner, Engr. in charge of Design-Installation, • Takoma Engineering Co., 923 Selim Rd., Silver Spring, and Hillandale, Silver Spring, Md.

RUSSELL, Edward A. (*M* 1936) Chief Engr., Vapor Car Heating Co., Inc., 1600 S. Kilbourn Ave., and • 8103 Dorchester Ave., Chicago, Ill.

RUSSELL, J. Nelson (*Life Member*; *M* 1899) Dir., Rosser & Russell, Ltd., 30 Conduit St., London, W. 1, and • Fernacres, Fulmer near Slough, Buckinghamshire, England.

RUSSELL, Wayne B. (*A* 1936) Gen. Mgr., Research Engineering Co., Second and Perry, and • 6064 E. Fourth, Rte. 8, Spokane, Wash.

RUSSELL, W. A. (*M* 1921), (Council, 1934-39, 1941) Southwestern Zone Mgr., • Hoffman Specialty Co., 1229 Union Ave., and 444 West 58th St., Kansas City, Mo.

RYAN, Harold J. (*M* 1940) Pres., • Harold J. Ryan, Inc., 122 East 42nd St., New York, and 461-81st St., Brooklyn, N. Y.

RYAN, Joseph B. (*M* 1938) Mech. Engr., Gentry & Voskamp, 13th and Main St., and • 20 West 74th St. Terrace, Kansas City, Mo.

RYAN, William F. (*M* 1940; *A* 1939; *J* 1933) Chief Engr., The Salina Supply Co., 302-304 N. Santa Fe., and • 310 W. Republic, Salina, Kan.

RYBOLT, Arthur L. (*A* 1938) Gen. Mgr., • The Rybolt Heater Co., Miller St., and 75 Samaritan Ave., Ashland, Ohio.
RYERSON, Herbert E. (*M* 1937) Sales Engr., Cardox Corp., 311 N. Michigan Ave., Chicago, and • 908 S. Wheaton Ave., Wheaton, Ill.

S

SABIN, Edward R. (*M* 1919) Pres., • E. R. Sabin & Co., 4710 Market St., Philadelphia, Pa., and 205 Page Ave., Allenhurst, N. J.
SABLE, Edward J. (*M* 1939) Treas., • The T. O. Murphy Co., 25-27 E. College St., and 246 W. Lorain St., Oberlin, Ohio.
SACHS, Sam* (*J* 1940) Research Asst. in Mech. Engrg., • University of Illinois, 213 Mech. Engrg. Lab., Urbana, and 725 S. Wright St., Champaign, Ill.
SADLER, C. Boone (*M* 1928) Associate Civil Engr., • Public Works Office, 11th Naval Dist., and 4828 Orchard Ave., San Diego, Calif.
SAENGER, Lester W. (*J* 1941) Htg. Engr., 4928 Holly Hills Ave., St. Louis, Mo.
SAGINOR, S. V. (*M* 1939) Gen. Mgr., • Davey Compressor Co., and 307 Woodard Ave., Kent, Ohio.
SAITO, Shozo (*M* 1923) • Saito Shozo Shoten, Ltd., Marunouchi Bldg., Opposite Tokyo Station, and 3, No. 4 3chome, Kamata-ku, Tokyo, Japan.
SALE, Francis B. (*A* 1939) Sales Engr., • Preferred Utilities Co., 2032 Belmont Rd., Washington, D. C.
SALINGER, Robert J. (*A* 1942; *J* 1937) Mech. Engr., The Austin Co., M. & M. Bldg., and • 2335 Swift, Houston, Tex.
SALTER, Ernest H. (*M* 1936) Engr., • Electrical Testing Labs., 2 East End Ave., New York, and 182 Cleveland Ave., Great Kills, S. I., N. Y.
SALZER, Alfred R., Jr. (*J* 1940) Mech. Engr., 1923 N. Claiborne Ave., New Orleans, La.
SAMPSON, Edwin T. (*A* 1938) Mgr., Acoustical Dept., • Atlas Asbestos Co., Ltd., 110 McGill St., Montreal, and 28 Finchley Rd., Hampstead, Que., Canada.
SAMPSON, Will D. (*M* 1940) Consulting Engr., • Will D. Sampson & Associates, 922 Ambassador Bldg., and 7108 Dale Ave., St. Louis, Mo.
SAMUELS, Sidney (*M* 1941; *A* 1928; *J* 1925) Pres., • Sidney Samuels, Inc., 146 West 99th St., and 245 West 107th St., New York, N. Y.
SANBERN, E. N.* (*M* 1923) Asst. Secy., Hoffman Specialty Co., Inc., 1001 York St., and • 5007 Boulevard Place, Indianapolis, Ind.
SANDER, Andy J. (*M* 1941) Engr., • Chicago Master Steamfitters Assn., 228 N. LaSalle, and 7401 Kingston Ave., Chicago, Ill.
SANDERS, Charles M., Jr. (*J* 1938) Precipitron Repr., • Westinghouse Electric & Manufacturing Co., 411 N. Seventh St., St. Louis, and 1115 North and South Rd., University, Mo.
SANDFORT, John F. (*A* 1942; *J* 1938) Instructor in Mech. Engrg., • Iowa State College, and 821½ Duff Ave., Ames, Iowa.
SANDS, Clive C. (*M* 1929) G. P. O. Box 601 F. F., Sydney, N.S.W., Australia.
SANFORD, A. L. (*M* 1915) Mech. Engr., C. H. Johnston, Archts. & Engrs., 715 Empire Bank Bldg., St. Paul, and • 127 Banning Ave., White Bear Lake, Minn.
SANFORD, Sterling S.* (*M* 1930) Sales Engr., • The Detroit Edison Co., 2000 Second Ave., and 1503 Seyburn Ave., Detroit, Mich.
SAPP, Charles L. (*A* 1936) Sales Mgr., Farquhar Furnace Co., and • 620 N. Walnut St., Wilmington, Ohio.
SATLOW, Abraham (*J* 1941) Jr. Naval Archt., Philadelphia Navy Yard, and 4206 Chester Ave., Philadelphia, Pa., and • 1946 Bay Ridge Parkway, New York, N. Y.
SATTERLEE, H. A. (*J* 1940) Partner, • The Schulte Plumbing & Heating Co., 526 Joplin St., and 1006 N. Sergt., Joplin, Mo.
SAUNDERS, Lawrence P. (*M* 1933), (Council, 1941) Chief Engr., Research Engrg., • Harrison Radiator Div., General Motors Corp., and The Tuscarora Club, Lockport, N. Y.
SAURWEIN, George K. (*M* 1938) Supt., Engrg., Harvard University, Lehman Hall, Cambridge, Mass.
SAWDON, Will M.* (*M* 1920) Prof. of Experimental Engrg., • Cornell University, College of Engrg., and 1018 E. State St., Ithaca, N. Y.
SAWHILL, R. V. (*A* 1929) Domestic Engineering, • 110 East 42nd St., New York, and 115 Townsend Ave., Pelham Manor, N. Y.
SAXON, Ruben B. (*M* 1941) Operating Supt., University of Nebraska, 42 and Dewey Ave., and • 1206 South 43rd St., Omaha, Nebr.
SCALINGI, Ciro R. (*J* 1940; *S* 1938) Melchior, Armstrong, Dessau Co., 614 Memorial Dr., Cambridge, and • 94 Josephine Ave., Somerville, Mass.
SCANDRETT, Harold R. (*M* 1941) Estimator, • Pacific Gas & Electric Co., 445 Sutter St., and 458–17th Ave., San Francisco, Calif.
SCANLON, Edward L. (*A* 1934) Chief Htg. Engr., • Equitable Gas Co., 435 Sixth Ave., Pittsburgh, and 3310 Regan Ave., Brentwood, Pittsburgh, Pa.
SCHAD, Clifford A. (*A* 1938; *J* 1937) Engr., United States Air Conditioning Corp., 2101 N.E. Kennedy St., and • 4425–43rd Ave. S., Minneapolis, Minn.
SCHAFER, Harry C. (*M* 1937) Sales Mgr., Iroquois Gas Corp., 45 Church St., Buffalo, and • 197 Union St., Hamburg, N. Y.
SCHECHTER, Jack E. (*A* 1941; *J* 1937) Design Engr., Raisler Corp., 129 Amsterdam Ave., New York, and • 2520 Kings Highway, Brooklyn, N.Y.
SCHEIDECKER, Daniel B. (*A* 1919) Pres., • Hunter-Clark Ventilating System Co., 2800 Cottage Grove Ave., and 4626 N. Kilbourn Ave., Chicago, Ill.
SCHERNBECK, Fred H. (*A* 1930) Sales, • William Bros. Boiler & Manufacturing Co., Nicollet Island, and 5045 Portland Ave., Minneapolis, Minn.
SCHERRER, Leon B. (*A* 1941; *J* 1936) Sales Engr., St. Louis Cooperage Co., 101 Arsenal St., and • 6112 Simpson Terrace, St. Louis, Mo.
SCHIMM, Ellis L. (*A* 1941) 7215 Coronada, Dallas, Tex.
SCHLICHTER, Charles F. (*M* 1938) Dist. Mgr., Surface Combustion Corp., 600 W St. N.E., Washington, D. C., and • 6740 Fairfax Rd., Bethesda, Md.
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STAHL, Walter A. (*M* 1938) Operating Mgr., • Merchandise Mart, 222 Bank Dr., Chicago, and 2504 Harrison St., Evanston, Ill.

STAMMER, Edward L. (*Life Member*; *M* 1919) Supt. Htg. and Vtg. Repairs, St. Louis Board of Education, Board of Education Bldg., and • 4430 Tennessee Ave., St. Louis, Mo.

STANDRING, Ronald A. (*A* 1942; *J* 1938) Htg. Engr., • Gurney Foundry Co., Ltd., 100 Principal St., Ville St. Laurent de Montreal, and 2358 Leclaire St., Maisonneuve, Montreal, Que., Canada.

STANGER, R. B. (*M* 1920) Prop., • Robinson & Stanger, Empire Bldg., Pittsburgh, and Middle Rd., Glenshaw, Pa.

STANGLE, William H. (*M* 1940) Mgr., New Products Div., Servel, Inc., 18th Fl., 51 East 42nd St., New York, and • 111-03–76th Ave., Forest Hills, L. I., N. Y.

STANLEY, Robert L. (*M* 1938) Engr., Factory Repr., • Pacific Gas Radiator Co., 7631 Roseberry Ave., Huntington Park, and 2518 Dearborn Dr., Los Angeles, Calif.

STANTON, Harold W. (*M* 1938) Commercial Sales Dir., • Iowa-Nebraska Light & Power Co., and 2100 Ryons St., Lincoln, Nebr.

STARK, W. E.* (*M* 1926) (Council, 1932-36) Mgr. Stoker Div., The Bryant Heater Co., 17825 St. Clair Ave., Cleveland, and • 1875 Rosemont Rd., East Cleveland, Ohio.

STASZESKY, Francis M. (*J* 1942; *S* 1938) Student (Co-Op), Boston Edison Co., • Mass. Inst. of Tech. Dorms, Cambridge, Mass. (from Oct. to June), and 10 Roselawn Ave., Roselle, Wilmington, Del.

STAUFFER, James E. (*S* 1940) Student, • Carnegie Institute of Technology, 624 Clyde St., Pittsburgh, Pa., and 2000 N. Maine St., Dayton, Ohio.

STECKHAN, Louis (*M* 1928; *J* 1926) Sales Engr., • Crane Co., 30 South 16th St., St. Louis, and 3240 Liberty St., St. Louis, Mo.

STEEL, R. Justin (*A* 1938) Lt. (jg) U. S. N. R., • Bureau of Ships, Navy Dept., Washington, D. C., and S. College and Kells Aves., Newark, Del.

STEELE, J. B. (*M* 1932) Chief Engr., • Winnipeg School Board, Ellen & William Ave., Winnipeg, and 184 Waterloo St., Riverheights, Winnipeg, Man., Canada.

STEFFNER, Edward F. (*A* 1937; *J* 1934) Htg. Engr., The Henry Furnace & Foundry Co., 3471 East 49th St., Cleveland, and • 1429 East 133rd St., East Cleveland, Ohio.

STEGGALL, Howard B. (*A* 1934) Branch Mgr., • United States Radiator Corp., 941 Behan St., and 1166 Murray Hill Ave., Pittsburgh, Pa.

STEIN, Jerome (*J* 1942; *S* 1940) Secy., Torrington Supply Co., Inc., 125 Maple St., and • 756 Waterville St., Waterbury, Conn.

STEINHORST, T. F. (*M* 1919) Pres., Emil Steinhorst & Sons, Inc., 612 South St., Utica, N. Y.

STEINKE, Bernard J. (*J* 1940; *S* 1937) Engr. and Estimator, 16 W. Palisade Ave., Englewood, and • Harrington Ave., Closter, N. J.

STEINMETZ, C. W. A. (*M* 1934) Office Mgr., • American Blower Corp., 249 High St., Newark, and 50 Oakwood Ave., Bogota, N. J.

STELLWAGEN, Frank G. (*A* 1937) Sales., • Fitzgibbons Boiler Co., Inc., 101 Park Ave., New York, and 8637–77th St., Woodhaven, L. I., N. Y.

STE-MARIE, Gaston P. (*M* 1930) Examiner Technician, • Provincial Government, 97 E. Notre Dame St., and 5329 Duquette Ave., N.D.G., Montreal, Que., Canada.

STENCEL, Reinhold Arthur (*M* 1938) Chief Engr., Canadian Ice Machine Co., Ltd., and • 540 Eglinton Ave. W., Toronto, Ont., Canada.

STENGEL, Frank J. (*A* 1935) Secy., • R. F. Stengel & Son, 76-80 Rosehill Pl., Irvington, and 55 Hughes St., Maplewood, N. J.

STEPHENSON, James R. (*A* 1940) Partner, • J. R. Stephenson & M. M. Perry, 311 Bank of Nova Scotia Bldg., and 647 Strathcona St., Winnipeg, Man., Canada.

STEPHENSON, K. A. (*M* 1941; *A* 1941) Secy.-Treas., • Stephenson Co., 14 Harris St. N.W., and 923 St. Charles Ave. N.E., Atlanta, Ga.

STEPHENSON, Lewis A. (*M* 1917) Dist. Mgr., • The Powers Regulator Co., 409 East 13th St., and 801 West 57th Terrace, Kansas City, Mo.

STERMER, Clarence J. (*M* 1936) Engr., Crane Co., 836 S. Michigan Ave., and • 7839 Clyde Ave., Chicago, Ill.

STERN, Edward J. (*A* 1941) Sales Engr., • Armstrong Cork Co., 1022 Santa Fe Bldg., and 223 E. Davis St., Dallas, Tex.

STERNBERG, Edwin (*A* 1932; *J* 1931) 16 East 98th St., New York, N. Y.

STERNE, C. M. (*A* 1934) Chief Engr., Metropolitan Refining Co., Inc., 50-23–23rd St., Long Island City, L. I., N. Y.

STERNER, Douglas S. (*M* 1941; *A* 1940; *J* 1938; *S* 1936) 1st Lt., • 67th Quartermaster Co., Camp Shelby, Miss., and 78 Baker St. N. W., Atlanta, Ga.

STETSON, Lawrence R. (*M* 1913) Engr., • McMurrer Co., 303 Congress St., Boston, and 35 Bradfield Ave., Roslindale, Mass.

STEVENS, Earl Knights (*A* 1940) Treas., • International Exposition Co., 480 Lexington Ave., New York, N. Y., and Fairfield Ave., Greenwich, Conn.

STEVENS, Harry L. (*M* 1934; *A* 1927; *J* 1924) Pres., • M. M. Stevens Plumbing & Heating Co., 108-110 W. Sherman St., and 320 West 20th St., Hutchinson, Kan.

STEVENS, Howard R. (*M* 1941) Mgr., Htg. and Air Cond. Dept., H. E. Saviers & Son, Inc., Cor. W. Second and West St., and • 530 E. Taylor St., Reno, Nev.

STEVENS, Judson E. (*A* 1941) In charge of Htg., Engrg. Dept., • National Coal Co., P. O. Box 318, and 520 Vassar St., Reno, Nev.

STEVENS, Kenneth M. (*J* 1936) Sales Engr., • Powers Regulator Co., 409 East 13th St., and 7336 Walnut, Kansas City, Mo.

STEVENS, Wayne H. (*A* 1939) Service Mgr. and Engr., • Shellenberger, Gregg & Co., 2203 N. Prospect Ave., and 2501 E. Stratford Court, Milwaukee, Wis.

STEVENS, William R. (*A* 1934) Htg. Contractor, • L. E. Stevens Co., 626 Broadway, Cincinnati, Ohio, and 30 Chalfonte Court, Fort Thomas, Ky.

STEVENSON, Mel. J. (*M* 1935) Htg. and Vtg. Engr., Consoer-Townsend & Quinlan & Battey & Childs, Archt.-Engr., Kansas Ordnance Plant, Parsons, Kan., and • 5321 Cornell Ave., Apt. 123, Chicago, Ill.

STEVENSON, W. W. (*M* 1928) Steam Htg. Engr., Allegheny Co. Steam Heating Co., 435 Sixth Ave., and 1125 Lancaster Ave., Pittsburgh, Pa.

STEWART, Charles W. (*M* 1919; *A* 1918) Asst. Gen. Sales Mgr., • Hoffman Specialty Co., 1001 York St., and 3111 No. Meridian St., Indianapolis, Ind.

STEWART, Duncan J.* (*M* 1936; *A* 1930) Mgr., Elec. Div., • Barber-Colman Co., and Hickory Lane, Rockford, Ill.

STEWART, James Pirtle (*A* 1940; *J* 1937) Engr., Carrier Corp., 12 South 12th St., Philadelphia, and • 126 Birch Ave., Bala, Pa.

STEWART, John N. (*A* 1939) Plan Examiner, Div. of Smoke Regulation and Boiler Inspection, District Bldg., and • 2019 N St. N.W., Washington, D. C.

STEWART, Robert S. (*A* 1941) Owner, • R. S. Stewart & Co., 7 N. Main St., and 512 Stanley St., Middletown, Ohio.

STEWART, Wesley O. (*A* 1938) Mgr., Los Angeles Office, • Johnson Service Co., 153 West Ave. 34, and 4100½ Los Feliz Blvd., Los Angeles, Calif.

STILES, Gordon S. (*A* 1941; *J* 1936) Instructor Engrg. Drawing, A. & M. College of Texas, College Station, and • 101 Dodge, R. F. D. 4, Bryan, Tex.

STILLER, Frederick W. (*A* 1942; *J* 1933) Estimator, • F. C. Stiller & Co., 129 S. Tenth St., and 138 West 49th St., Minneapolis, Minn.

STIRLING, Walter N. (*M* 1939) Refrig., Air Cond., 189 Park St., Calcutta, India.

STITES, Richard, Jr. (*J* 1937) Sales Engr., • Buffalo Forge Co., 2051 W. Lafayette Blvd., and 2170 E. Jefferson, Detroit, Mich.

STOCK, Charles S. (*M* 1936) Dist. Repr., The Herman Nelson Corp., Room 404, 1108–16th St. N.W., Washington, D. C., and • 6752 Fairfax Rd., Bethesda, Md.

STOCKWELL, William R. (*Life Member*; *M* 1903; *J* 1901) Gen. Mgr., Mfg. Div., Weil-McLain Co., Michigan City, Ind.

STOKES, Alvin D. (*M* 1936) Engr., • Riggs Distler & Co., Inc., 216 N. Calvert St., and 424 Winston Ave., Baltimore, Md.

STOKES, Arledge (*A* 1942; *J* 1936) Engr., Mehring & Hanson Co., 12 H St. N.E., Washington, D. C., and • 4908 North 17th St., Arlington, Va.

STONE, Frank M. (*M* 1940) Secy.-Treas., • Sam Stone, Jr. & Co., Inc., Archts., 1800 Masonic Temple, and 2214 Adams St., New Orleans, La.

STORMS, Robert M. (*M* 1930) Consulting Engr., Plbg.-Htg.-Vtg., • 1717 N. Vine St., Los Angeles, and 354 W. Wilson Ave., Glendale, Calif.

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STOTT, F. W. (*M* 1938) Sales, C. A. Dunham Co., Ltd., 1139 Bay St., Toronto, and • 86 Allan St., Oakville, Ont., Canada.

STOTZ, Robert B. (*J* 1938) Sales Engr., • Frigidaire Div., General Motors Sales Corp., 675 Greenwood Ave., and 1702 Harvard Rd., Atlanta, Ga.

STRAND, Charles A. (*A* 1940) Supt., • Bruce Wigle Plumbing & Heating Co., 9117 Hamilton Ave., and 6533 Barlum Ave., Detroit, Mich.

STRANSKY, Malcolm W. (*A* 1942; *J* 1941) Engr., Heating Equipment Co., 1123 Harrison St., and 316 Fulton St., San Francisco, Calif.

STRAUCH, Paul C. (*A* 1934) Sales Engr., The Henry Furnace & Foundry Co., 24 South 18th St., Pittsburgh, and • Sherwood Hall, Cambridge Court Apts., Edgewood, Pittsburgh, Pa.

STREATER, Edward C. (*A* 1939) Mgr., • L. E. Streater Lumber Co., Spring Park, and Mound, Minn.

STREVELL, R. P. (*M* 1934) Pres.-Treas., • The William R. Hogg Co., Inc., 900 Fourth Ave., Asbury Park, and Victor Pl. and State Highway, Neptune, N. J.

STROCK, Clifford (*M* 1937; *A* 1929) Editor, Heating & Ventilating, 148 Lafayette St., New York, N. Y.

STROMGREN, Sven G. (*M* 1938) Metallurgical Engr., Svenska Flaktfabriken Kungsgatan 18, Stockholm, Sweden.

STROTHER, William E. (*A* 1941) Construction Engr., Barge-Thompson Co., and • 997 Burns Dr., Atlanta, Ga.

STROUSE, Sherman W. (*A* 1934) Sales Mgr., Trane Co., 493 Franklin St., Buffalo, and • 95 Mayville Ave., Kenmore, N. Y.

STROUSE, Sidney B. (*M* 1921) Member of Firm, • S. B. & B. H. Strouse, 500-529 Guarantee Trust Bldg., and 22 S. Illinois Ave., Atlantic City, N. J.

STRUNIN, Jay (*A* 1939; *J* 1933) Engr., and Contractor, • Strunin Plumbing & Heating Co., 408 Second Ave., New York, and 217 Ocean Ave., Brooklyn, N. Y.

STUART, Milton C.* (*M* 1935) Prof. of Mech. Engrg. • Lehigh University, and 1828 Jennings St., Bethlehem, Pa.

STUART, W. W. (*A* 1940) Owner, • W. W. Stuart Co., 417 Ninth St., and 1920 Pleasant St., Des Moines, Iowa.

STUBBS, W. C. (*M* 1934) Supvr., Ventilation Section, • Norfolk Navy Yard, and 30 Channing Ave., Portsmouth, Va.

STURDY, Oswald C. (*M* 1938) Sales Engr., • Foster Wheeler, Ltd., Commerce and Transportation Bldg., and 139 Indian Rd., Toronto, Ont., Canada.

STURM, William (*J* 1937; *S* 1936) Draftsman, Gausman & Moore, Consulting Engrs., 1026 First National Bank Bldg., and • 1822 Ashland, St. Paul, Minn.

SUDDERTH, Leo, Jr. (*A* 1942; *J* 1936) Branch Mgr., • Johnson Service Co., 311 Bona Allen Bldg., and 1115 Los Angeles Ave., N.E., Atlanta, Ga.

SULLIVAN, T. J. (*M* 1930) Pres., Sullivan Valve & Engineering Co., 910 S. Arizona St., and • 1205 W. Park St., Butte, Mont.

SUNDERLAND, Richard P. (*A* 1938) Pres., • General Meters-Controls Co., 205 W. Wacker Dr., Chicago, and 936 Judson Ave., Evanston, Ill.

SUPPLE, Graeme B. (*M* 1934) Dist. Sales Engr., • American Blower Corp., 625 Architects & Builders Bldg., and 420 East 55th St., Indianapolis, Ind.

SUTCH, H. C. (*A* 1940) Sales Engr., Bell & Gossett Co., 629 W. Washington, and • 1326 Ardmore Ave., Chicago, Ill.

SUTCLIFFE, A. G. (*M* 1922; *A* 1918) Chief Engr., Ilg Electric Ventilating Co., 2850 N. Crawford Ave., Chicago, and • 432 S. Delphia Ave., Park Ridge, Ill.

SUTFIN, George V. (*A* 1937) Sales Engr., • American Blower Corp., 1003-6 American Bldg., and 3270 Hildreth Ave., Cincinnati, Ohio.

SUTHERLAND, David L. (*A* 1934) Pres., • Sutherland Air Conditioning Co., 718 Washington Ave. N., and 1815 Colfax Ave. S., Minneapolis, Minn.

SUTTER, Edgar E. (*A* 1936) Sales Engr., Mueller Brass Co., Port Huron, Mich., and • 6705 Sixth St. N.W., Washington, D. C.

SWAIN, Douglas S. (*J* 1941) Sales Engr., Trane Company of Canada, Ltd., 365 Hargrave St., and • 27 Fawcett Ave., Winnipeg, Man., Canada.

SWAIN, William L. (*M* 1939) Dir., • Young, Austen & Young, Ltd., 35 Uphill Rd., Mill Hill, London, N.W. 7, and St. Catherine's, Sandy Lodge Rd., Moor Park, Hertfordshire, England.

SWANEY, Carroll R. (*M* 1929; *J* 1921) Co-Partner, • Gilbert Howe Gleason & Co., 28 St. Botolph St., Boston, and 43 Clyde St., Newtonville, Mass.

SWANSON, Donald F. (*J* 1938) Test Engr., Seeger Refrigerator Co., 850 Arcade St., St. Paul, and • 4316 Bloom Ave., Minneapolis, Minn.

SWANSON, Earl C. (*A* 1935) Vice-Pres., Andersen Corp., Bayport, Minn.

SWANSON, Nils W. (*A* 1936) Sales, McDonnell & Miller, 400 N. Michigan Ave., and • 2746 Morse Ave., Chicago, Ill.

SWEENEY, R. H. (*A* 1939) Sales Engr., Minneapolis-Honeywell Regulator Co., 4740 Baum Blvd., and • 1201 Hillsdale Ave., No. 16, Pittsburgh, Pa.

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SWENSON, J. E. (*A* 1930) Mgr., Htg. Div., • Minneapolis Gas Light Co., 739 Marquette Ave., and 4853 South 14th Ave., Minneapolis, Minn.

SWINGLE, Wayne T. (*A* 1938) Pres., Hastings Air Conditioning Co., Inc., 918 W. Seventh St., Hastings, Nebr.

SWISHER, Stephen G., Jr. (*M* 1936; *A* 1934) Mgr., • The Trane Co., 1835 N. Third St., and 1711 E. Dean Rd., Milwaukee, Wis.

SYMONDS, Edward S. (*M* 1939) Dir., • Abair Engineering, 1 Devonshire Sq., London, E.C. 2, and 84, The Ridgeway, Chingford, Essex, England.

SYSKA, Adolph G. (*M* 1933) Partner, • Syska & Hennessy, 144 East 39th St., New York, and 3 Alden Pl., Bronxville, N. Y.

SZEKELY, Ernest (*M* 1920) Pres., • Bayley Blower Co., 1817 South 66th St., Milwaukee, and 6026 W. Washington Blvd., Wauwatosa, Wis.

SZOMBATHY, L. R. (*A* 1930) Pres., • Ferguson Sheet Metal Works, Inc., 34 N. Florissant Blvd., Ferguson, and 3125 Hawthorne Blvd., St. Louis, Mo.

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TAGGART, Ralph C.* (*M* 1912) Div. of Architecture, N. Y. State Dept. of Public Works, Albany, and • 14 Lyon Ave., Menands, Albany, N. Y.

TAHRY, Mahmoud El (*M* 1939) Engr., • Carrier Egypt, S.A.E., 37 Sharia Kasr El Nil, Cairo, and 30, Hishmat Pacha St., Zamalek, Cairo, Egypt.

TALIAFERRO, Robert R.* (*M* 1919) Application Engr., Carrier Corp., 302 S. Geddes St., and • 101 Comstock Ave., Syracuse, N. Y.

TALLIANOS, Peter C. (*A* 1938) Mgr., The Egyptian Wireless Co., 36 Nebi Daniel St., Alexandria, Egypt.

TALLMADGE, Webster (*M* 1924) Pres., • Webster Tallmadge & Co., Inc., 304 Glenwood Ave., East Orange, and 56 Cambridge Rd., Montclair, N. J.

TANGER, Othon C. F. (*A* 1937) Dir., • N. V. Technische Handelsmaatschappij "Renova," Rembrandtlaan 34, Arnhem, Netherlands.

TARR, Harold M. (*M* 1931) Htg., Vtg. and Air Cond. Engr., 21 Montague St., Arlington Heights, Mass.

TASKER, Cyril* (*M* 1935), (Council 1941) Sr. Research Fellow • Ontario Research Foundation, 43 Queens Park, and 737 Avenue Rd., Toronto, Ont., Canada.

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TAYLOR, Fielding, Jr. (*J* 1938) Sales Engr., • American Machine & Metals, Inc., DeBothezat Ventilating Div., 100 Sixth Ave., New York, and 83-02 Cornish Ave., Elmhurst, L. I., N. Y.

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TAYLOR, R. F. (*M* 1915) Consulting Engr., • 910 Bankers Mortgage Bldg., and 2332 Watts Rd., Houston, Tex.

TAYLOR, Robert B. (*J* 1938) Repr., • Buffalo Forge Co., 112 State St., Albany, and R. F. D., Slingerlands, N. Y.

TAYLOR, Thomas E. (*A* 1942; *J* 1937) Consulting Engr., • 307 Postal Bldg., and 7307 N. Wall, Portland, Ore.

TAZE, D. L. (*M* 1931) Mgr., • American Blower Corp., 1302 Swetland Bldg., Cleveland, and 19412 Winslow Rd., Shaker Heights, Ohio.

TAZE, Edwin H. (*M* 1937) Branch Mgr., • American Blower Corp., 620 Court Sq. Bldg., Baltimore, and 28 Normal Terrace, Towson, Md.

TEASDALE, Lawrence A. (*M* 1926) Asst. to Gen. Mgr., • Yale University Service Bureaus, 20 Ashmun St., and 262 W. Rock Ave., New Haven, Conn.

TEELING, George A. (*M* 1930) Consulting Engr., • 1 Columbia Pl., Albany, and Clarksville, N. Y.

TEMPLE, W. J. (*M* 1931) Mgr. and Engr., J. A. Temple Co., 108 Parkway, and • 1215 Reed Ave., Kalamazoo, Mich.

TEMPLIN, Charles L. (*M* 1921) Pres., • Carrier Atlanta Corp., 348 Peachtree St., and 781 Sherwood Rd. N.E., Atlanta, Ga.

TENKONOHY, Rudolph J. (*M* 1923) 3650 Shaw Blvd., St. Louis, Mo.

TENNANT, Raymond J. J. (*A* 1929) Engr., • Pittsburgh Business Properties, Inc., 2237 Oliver Bldg., Pittsburgh, and 752 N. Meadowcroft Ave., Mt. Lebanon, Pittsburgh, Pa.

TENNEY, Dwight (*M* 1932) Pres., • Tenney Engineering, Inc., Bloomfield, and 33 Summit Rd., Verona, N. J.

TERHUNE, Ralph D. (*A* 1936) Mgr., • Bryant Heater Co., 249 High St., Newark, and 114 Highland Ave., Ridgewood, N. J.

TERRY, Matson C. (*M* 1936) Pres., Certified Products Co., 2014 North 14th St., and • 1429 Potomac Dr., Toledo, Ohio.

TERRY, Samuel W. (*M* 1941) Pres., • Aladdin Heating Corp., 2222 San Pablo Ave., Oakland, and 2820 Oak Knoll Terrace, Berkeley, Calif.

ter WEEME, Albert (*A* 1938) Sales Engr., N. V. Radiatoren, Singel 206-208, Amsterdam C, and • Archimedesweg 13, Amsterdam O, Holland.

THEISS, Ernest S. (*A* 1941; *J* 1940) Instructor, Duke University, College Station, and • 910 Fifth St., Durham, N. C.

THEOBALD, Art (*A* 1937) Engr., • Payne Furnace & Supply Co., Inc., 336 N. Foothill Rd., Beverly Hills, and 116½ S. Kings Rd., Los Angeles, Calif.

THEORELL, Axel T. (*M* 1939) Civil Engr., Theorells Ingeniorsbyra, Skoldungagatan 4, and • Lokattsvagen 41, Appelviken, Stockholm, Sweden.

THEORELL, Hugo G. T.* (*Life Member*; *M* 1902) Consulting Engr., • Hugo Theorells Ingeniorsbyra, Skoldungagatan 4, Stockholm, Sweden.

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THOMAN, Estell O. (*A* 1938) Htg. and Air Cond. Engr., Boot & Co., 115 Fulton St. W., and • 714 Fulton St. E., Grand Rapids, Mich.

THOMAS, Arthur E. (*J* 1938) Contracts Mgr., Young, Austen & Young, Ltd., Mayfield Lodge, Dudlow Lane, Calderstones, Liverpool, 18, and • "Arlen" 56, Thingwall Rd., Wavertree, Liverpool, 15, England.

THOMAS, Bernard A. (*A* 1937; *J* 1923) Mgr., Htg. Dept., Crane Co., and • 405 E. Idlewild Ave., Tampa, Fla.

THOMAS, Glegge (*M* 1923) Office Mgr., • Clarage Fan Co., 723 Albee Bldg., Washington, D. C., and 7 W. Leland St., Chevy Chase, Md.

THOMAS, L. G. L. (*M* 1934) Vice-Pres., • Economy Pumps, Inc., 1000 Weller Ave., and 403 South D St., Hamilton, Ohio.

THOMAS, Melvern F. (*M* 1909) Supvr. of Htg. & Elec. Work, No. 1 Training Command, R. C. A. F., 55 York St., and • 74 Rivercrest Rd., Toronto, Ont., Canada.

THOMAS, Norman A. (*M* 1928) Pres., Thomas Heating Co., 142 South 14th St., LaCrosse, Wis.

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THOMAS, Richard H. (*Life Member*; *M* 1920) Pres. and Treas., • Economy Pumps, Inc., and 403 South D St., Hamilton, Ohio.

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THOMPSON, Frank (*M* 1935) Mgr., The Canadian Fairbanks-Morse Co., Ltd., Belvidere St. S., and • 107 Quebec St., Sherbrooke, Que., Canada.

THOMPSON, Nelson S.* (*Life Member*; *M* 1917; *J* 1897) Chief Mech. Engr., U. S. Government, and • 1615 Hobart St. N.W., Washington, D. C.

THOMSEN, Nis B. (*M* 1938) 168 Fallingbrook Rd., Toronto, Ont., Canada.

THOMSON, Thomas N.* (*Life Member*; *M* 1899) Htg. and Plbg. Consultant, 37 Irwin Pl., Huntington, L. I., N. Y.

THORNBURG, Harold A. (*M* 1932; *J* 1929) Consulting and Chief Engr., N. V. Industrieele Mij. Gebr. van Swaay, 154 Nassau St., New York, N. Y., and Soerabaya, Java, N.E.I., and • 1911 Dorchester Rd., Apt. 6G, Brooklyn, N. Y.

THORNTON, Thaddeus L. (*M* 1937) Maintenance Engr., Prudential Insurance, 96 Barclay St., Newark, and • 37 Perry St., Belleville, N. J.

THORNTON, William Barry* (*M* 1931) Sales Engr., • Carrier Corp., Merchandise Mart, and 8314 Indiana Ave., Chicago, Ill.

THRUSH, H. A. (*M* 1918) Pres., H. A. Thrush & Co., Peru, Ind.

THULMAN, Robert Kelley* (*M* 1938) Mech. Engr., Federal Housing Administration, Vermont and K Sts. N.W., Washington, D. C., and • 6505 Ridgewood Ave., Chevy Chase, Md.

THUNEY, Francis M. (*A* 1939; *J* 1936) Application Engr., • Wm. E. Kingswell, Inc., 3707 Georgia Ave. N.W., Washington, D. C., and 202 Glenwood Rd., Bethesda, Md.

TIDMARSH, P. M. (*M* 1938) Vice-Pres. and Gen. Mgr., Tidmarsh Engineering Co., P. O. Box 2425, Tucson, Ariz.

TILFORD, Leo A. (*M* 1941) Owner and Mgr., • Leo A. Tilford Temperature Control Co., 295 W. Pearl St., and 1224 Loeser Ave., Jackson, Mich.

TILLER, Louin (*A* 1935; *J* 1935; *S* 1933) Air Cond. Engr., Oklahoma Gas & Electric Co., 321 N. Harvey, and • 2712 Northwest 15th St., Oklahoma City, Okla.

TILLINGHAST, Harry S. (*J* 1941; *S* 1940) Chief Time Study, • Aluminum Company of America, 64 Lincoln Blvd., Kenmore, N. Y., and 60 W. Euclid, Detroit, Mich.

TILTZ, Bernard E. (*M* 1930) Pres., Tiltz Air Conditioning Corp., 230 Park Ave., New York, and • 25 Lookout Circle, Larchmont, N. Y.

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TIMMIS, William Walter (*M* 1933; *A* 1925) Pres., • Au-Temp-Co. Corp., 33 West 60th St., New York, and Woodland Dr., Pleasantville, N. Y.

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WEBB, Ernest C.* (*M* 1935) Engrg. Service Mgr., • Iron Fireman Manufacturing Co., 3170 West 106th St., Cleveland, and 24721 West Lake Rd., Bay Village, Ohio.

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WEBBER, Charles H. (*A* 1940) Sales Engr., • Pacific Scientific Co., 1430 Grande Vista Ave., Los Angeles, and 1176 Mt. Lowe Drive, Altadena, Calif.

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WEBER, Eugene F., Jr. (*A* 1940; *J* 1937) Sales Engr., York Ice Machinery Corp., 117 South 11th St., St. Louis, and • 607 Forest Court, Clayton, Mo.

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WEST, C. H., Jr. (*A* 1941) Vice-Pres., • Massey, Wood & West, Inc., Lombardy Underpass, and 1602 Confederate Ave., Richmond, Va.

WEST, Perry* (*M* 1911) (Treas., 1924-25; Council, 1920-25) Prof. of Steam Power Engrg., Head of Mech. Engrg. Dept., • College of Engineering, University of Kentucky, and 185 E. Maxwell St., Lexington, Ky.

WESTENDARP, Francisco G. (*M* 1939) Engr., • Guinard Freres Sucs. Dept., Tecnico Apartado 668, Caracas, Venezuela.

WESTOVER, Wendell (*M* 1936) Pres., • Westover-Wolfe, Inc., 170 Washington Ave., and 254 Lenox Ave., Albany, N. Y.

WESTPHAL, N. E. (*J* 1940; *S* 1937) Ensign, U. S. N. R., • "B" Div. Officer, U.S.S. Helena, c/o Postmaster, San Pedro, Calif., and Long Beach, Michigan City, Ind.

WETHERED, Woodworth (*M* 1938) The Bohemian Club, San Francisco, Calif.

WETZELL, Horace E. (*M* 1934) Vice-Pres. and Chief Engr., • The Smith & Oby Co., 6107 Carnegie Ave., Cleveland, and 21144 Aberdeen Rd., Rocky River, Ohio.

WHEELER, Charles A. (*M* 1941) Branch Mgr., • Johnson Service Co., 511 Fifth Ave., and 345 49th St. Place, Des Moines, Iowa.

WHEELER, Joe, Jr. (*M* 1938) Sales Repr., • Johnson Service Co., 28 East 29th St., New York, and 261 Dogwood Lane, Manhasset, L. I., N. Y.

WHELAN, William J. (*M* 1923) Estimating, • Harrigan & Reid Co., 1365 Bagley Ave., and 3790 Seminole Ave., Detroit, Mich.

WHELLER, Harry S. (*M* 1916) Vice-Pres., • L. J. Wing Manufacturing Co., 154 West 14th St., New York, N. Y., and 725 Union Ave., Elizabeth, N. J.

WHITE, Elmer D. (*A* 1939; *J* 1937) 2705 Dar Hills Ave., Dayton, Ohio.

WHITE, Elwood S. (*M* 1921) Pres., • U. S. Radiator Corp., 1056 National Bank Bldg., Detroit, Mich., and Meadowbank Rd., Old Greenwich, Conn.

WHITE, Eugene B. (*M* 1934) Exec. Secy., • Architectural and Engineering Bureau, Young Men's Christian Assn., 19 S. LaSalle St., Chicago, and 309 N. Taylor Ave., Oak Park, Ill.

WHITE, Harry S. (*A* 1936) Mgr., • Acme Sheet Metal Co., 5301 E. Ninth, and 6805 Edgevale Rd., Kansas City, Mo.

WHITE, John C. (*M* 1932) State Power Plant Engr., • State Bureau of Engineering, 624 E. Main St., and 622 E. Main St., Madison, Wis.

WHITE, Taylor G., Jr. (*A* 1938) Branch Mgr., U.S. Radiator Corp., 709 Ashland, Louisville, Ky.

WHITE, Thomas J. (*A* 1941; *J* 1938) Sales Engr., • American Blower Corp., 625 Market St., San Francisco, and 2340 Pelham Place, Piedmont Pines, Oakland, Calif.

WHITE, W. Emery (*M* 1941) • The Smith-White Co., Commerce Bldg., and 1415 East 76th Terrace, Kansas City, Mo.

WHITE, W. R. (*M* 1938; *A* 1936) Industrial Engr., • Nebraska Power Co., 718 Electric Bldg., and 4916 Grand Ave., Omaha, Nebr.

WHITELAW, H. Leigh (*M* 1916) Jones & Laughlin Steel Corp., Third and Ross, and • Schenley Apts., Pittsburgh, Pa.

WHITMER, Robert P. (*M* 1935) Secy., • American Foundry & Furnace Co., and 1402 E. Washington St., Bloomington, Ill.

WHITNEY, C. W. (*M* 1935) Pres., • ABC Oil Burner & Engineering Co., 2012-14 Chestnut St., Philadelphia, and Sevilla Court, Apt. F-3, Bala-Cynwyd, Pa.

WHITT, Sidney A. (*A* 1938; *J* 1937) Sr. Design Engr., Fedders Manufacturing Co., Inc., 57 Tonawanda St., and • 12 Inwood Place, Buffalo, N. Y.

WHITTAKER, Wayne K. (*A* 1935) Engr., Irving Trust Co. Bldg., 1 Wall St., New York, and • 119-23 226th St., St. Albans, L. I., N. Y.

WHITTEN, H. E. (*M* 1924) Pres.-Treas., H. E. Whitten Co., 9 Federal Court, Boston, and • 56 Highland Rd., Somerville, Mass.

WHITTLESEY, Welsh C. (*M* 1941) Engr. (Naval Archt.), Navy Dept., Washington, D. C., and • 1500 S. Barton St., Arlington, Va.

WIDDOWFIELD, A. S. (*A* 1941; *J* 1937) Sales Engr., The Mercoid Corp., 4201 Belmont Ave., Chicago, Ill., and • 535 Hanna St., Birmingham, Mich.

WIDMER, Walter J. (*A* 1939) Secy.-Treas., • Widmer Plumbing & Heating Co., Inc., 34 N.E. Seventh Ave., and 1565 N. Shaver St., Portland, Ore.

WIEGNER, Henry B. (*M* 1919) Office Mgr., Johnson Service Co., 20 Winchester St., Boston, and • 143 Standish Rd., Watertown, Mass.

WIESNER, Blaine K. (*S* 1939) Student, Purdue University, and • 931 Sixth St., West Lafayette, Ind.

WIGGS, G. Lorne (*M* 1936; *A* 1932; *J* 1924) (Council, 1938-40) Consulting Engr., • 714 University Tower Bldg., and 4797 Grosvenor Ave., Montreal, Que., Canada.

WILCOX, Chester M. (*A* 1939) Engr., 3118 Mt. Pleasant St. N.W., Washington, D. C.

WILDE, Ray S. M. (*M* 1916) Consulting Engr., 18286 Griggs, Detroit, Mich.

WILDER, Edward L. (*M* 1915) Industrial Promotion Engr., • Rochester Gas & Electric Corp., 89 East Ave., and 369 Bonnie Brae Ave., Rochester, N. Y.

WILDER, Herbert P. (*M* 1938) Sales Engr., • Patterson-Kelley Co., 101 Park Ave., New York, and Ardsley Rd., Scarsdale, N. Y.

WILDMAN, Eugene L. (*A* 1942; *J* 1939) N. Y. Repr., Stewart A. Jellett Co., 1328 Broadway, New York, N. Y., and • 122 N. Walnut St., East Orange, N. J.

WILEY, Donald C.* (*A* 1939; *J* 1936) Production Mgr., • John J. Nesbitt, Inc., State Rd. and Rhawn St., Philadelphia, and Kelvin Ave., Somerton, Pa.

WILHELM, Joseph E. (*J* 1936; *S* 1934) Purch. Agent, Avery Engineering Co., 1906 Euclid Ave., Cleveland, and • 294 East 195th St., Euclid, Ohio.

WILKES, Gordon B.* (*M* 1937) Prof. of Heat Engrg., • Massachusetts Institute of Technology, Cambridge, and 51 Everett St., Newton Centre, Mass.

WILKINSON, Arthur (*A* 1936) Wilkinson Engineering Agencies, 1253 McGill College Ave., Montreal, Que., Canada.

WILKINSON, F. J. (*M* 1933) Mgr., Central Engrg. Service, Montgomery Ward & Co., Chicago Ave. and Larabee St., Chicago, and • 18257 Martin Ave., Homewood, Ill.

WILLARD, Arthur C.* (*M* 1914) (*Presidential Member*) (Pres., 1928; 1st Vice-Pres., 1927; 2nd Vice-Pres., 1926; Council, 1925-29) Pres., • University of Illinois, 355 Administration Bldg., and 711 Florida Ave., Urbana, Ill.

WILLEY, Earl C. (*M* 1934) Asst. Prof. of Mech. Engrg., Oregon State College, and • 121 North 29th St., Corvallis, Ore.

WILLIAMS, Allen W. (*Life Member*; *A* 1915) Secy.-Treas., • National Warm Air Register Manufacturers Institute, Room 808, 5 E. Long St., Columbus, and 51 Meadow Park, Bexley, Ohio.

WILLIAMS, Chester D. (*M* 1938) Mgr., • General Air Conditioning & Heating Co., 4001 Piedmont Ave., Oakland, and 2709 College Ave., Berkeley, Calif.

WILLIAMS, Donald D. (*A* 1940; *J* 1938) Industrial Gas Engr., • Iowa-Nebraska Light & Power Co., 1401 O St., and 2236 A St., Lincoln, Nebr.

WILLIAMS, Donald L. (*M* 1941) Sales Engr., • General Air Conditioning & Heating Co., 4001 Piedmont Ave., and 67 Glen Ave., Oakland, Calif.

WILLIAMS, Elwin C. (*A* 1939) Sales Repr., 4028 Egbert Ave., Cincinnati, Ohio.

WILLIAMS, Frank H. (*A* 1940; *J* 1934) Plant Engr., Delco-Remy Div., General Motors Co., and • 916 W. Seventh St., Anderson, Ind.

WILLIAMS, Gordon S. (*A* 1941; *J* 1937; *S* 1936) Partner, Augur-Williams Co., 15 Chapel St., Woodmont, Conn.

WILLIAMS, H. Edmund (*J* 1939) Engr., Justin C. O'Brien Co., Inc., 734 Lexington Ave., and • 14 West 103rd St., New York, N. Y.

WILLIAMS, J. Walter (*M* 1915) Pres.-Treas., • Forest City Plumbing Co., 332 E. State St., and 923 E. State St., Ithaca, N. Y.

WILLIAMS, Lyle G. (*M* 1939) Plbg. and Htg., • Crane Co., and 5555 33rd Ave. N.E., Seattle, Wash.

WILLIS, Leonard L. (*J* 1936; *S* 1935) Vice-Pres., H. Conrad Manufacturing Co., 509 First Ave. N.E., and • 4525 Bryant Ave. S., Minneapolis, Minn.

WILLNER, Ira (*M* 1937) Pres., • Willner Heating Co., Inc., 415 Lexington Ave., and 125 East 93rd St., New York, N. Y.

WILLS, Fred W. (*J* 1938) Sales Engr., Tuttle & Bailey, Inc., 61 W. Kinzie St., and • 2257 W. Addison St., Chicago, Ill.

WILLSON, Frank J. (*M* 1941) Vice-Pres., • Staynew Filter Corp., 25 Leighton Ave., and 2219 Westfall Rd., Rochester, N. Y.

WILMOT, Charles S. (*M* 1919) 436 Haverford Ave., Narberth, Pa.

WILSON, Alexander M. (*J* 1942; *S* 1939) Engr., Andrew Wilson Co., 616 Essex St., Lawrence, and • 13 Third St., North Andover, Mass.

WILSON, Eric D. (*M* 1936) Sales Engr., • Carrier Engineering Co., Ltd., 24 Buckingham Gate, London, S.W.1, and Ways End, St. Albans Rd., Reigate, Surrey, England.

WILSON, George T. (*M* 1925) Sales Engr., Gurney Foundry Co., Ltd., 4 Junction Rd., Toronto, and • 25 Tyre Ave., Islington, Ont., Canada.

WILSON, Raymond W. (*M* 1934) Member of Firm, Wilson-Brinker Co., 412 Pythian Bldg., and • 429 Creston Ave., Kalamazoo, Mich.

WILSON, Robert A. (*M* 1936) Sales Engr., Minneapolis-Honeywell Regulator Co., 4501 Prospect Ave., Cleveland, and • Cor. Olive St. and Robens Court, Chagrin Falls, Ohio.

WILSON, Victor H. (*A* 1938) Engr.-Distributor and Contractor in Refractories, Plibrico Jointless Firebrick Co., 403 Hitchcock Bldg., Nashville, and • "The Thistle-Patch," Donelson, Tenn.

WILSON, Westray E. (*A* 1939) Major, QMC, U. S. Army, • Reception Center, Fort Benning, Ga., and Owner and Mgr., Wilson Plumbing Co., 227 Haywood Rd., Asheville, N. C.

WILSON, W. H. (*A* 1932) Chief Power Plant Engr., Pullman-Standard Car Manufacturing Co., 11001 Cottage Grove Ave., and • 22 West 110th Place, Chicago, Ill.

WILTBERGER, Constant F. (*M* 1935) Partner, Consulting Engrg., Pennell & Wiltberger, Land Title Bldg., and • 2650 N. Ninth St., Philadelphia, Pa.

WINANS, G. D. (*M* 1929) Engr. of Steam Distribution, • The Detroit Edison Co., 2000 Second Ave., and 16183 Wisconsin, Detroit, Mich.

WINER, Bernard B. (*S* 1940) Student, • Carnegie Institute of Technology, 4921 Forbes St., and 3138 Avalon St., Pittsburgh, Pa.

WINKLER, Ralph A. (*A* 1940; *J* 1937) Sales Engr., Alfred C. Goethel Co., 2337 North 31st St., Milwaukee, and • P.O. Box 179, Elm Grove, Wis.

WINSLOW, C.-E. A.* (*M* 1932) (Council, 1940-41) Prof. of Public Health, Yale University School of Medicine, and Dir., John B. Pierce Laboratory of Hygiene, • 310 Cedar St., and 314 Prospect St., New Haven, Conn.

WINTERBOTTOM, Ralph F. (*M* 1923) Engr., Winterbottom Supply Co., and • 720 Moir, Waterloo, Iowa.

WINTERER, Frank C. (*M* 1920) Branch Sales Mgr., • American Radiator & Standard Sanitary Corp., 300 Broadway, and 836 Juno St., St. Paul, Minn.

WISE, Mason W. (*M* 1923) Owner, M. W. Wise Co., "Lakewood," and • 1656 Melrose Drive S.W., Atlanta, Ga.

WISER, C. E. (*A* 1941) Dist. Mgr., • Minneapolis-Honeywell Regulator Co., 3023 Farnam St., and 6312 Florence Blvd., Omaha, Nebr.

WITHERIDGE, David E. (*J* 1936) Sales Engr., W. A. Witheridge Co., 2340 Mershon St., Saginaw, Mich.

WITMER, Howard S. (*A* 1937) Design Engr., • City Hall, and 600 Elm St., Bay City, Mich.

WOESE, Carl F. (*M* 1934) Consulting Engr., • Robson & Woese, Inc., 1001 Burnet Ave., and 256 Robineau Rd., Syracuse, N. Y.

WOLFE, John S. (*M* 1941) Mech. Engr., Board of School Directors, 1012 W. Highland Ave., and • 2004 N. Bartlett Ave., Milwaukee, Wis.

WOLFF, Peter P. (*M* 1935) Engr., Bell & Gossett Co., 3000 Wallace St., and • 7509 Ridgeland Ave., Chicago, Ill.

WOLIN, Milton W. (*J* 1938; *S* 1937) Engr., Typhoon Air Conditioning Co., Inc., 252 West 26th St., New York, N. Y., and • R.F.D. 2, Box 73-D, New Brunswick, N. J.

WOLL, Willard M. (*M* 1938) Engr.-Industrial Htg. and Refrig., • Commonwealth Edison Co., 72 W. Adams St., and 9320 S. Throop St., Chicago, Ill.

WOLLENBERGER, Louis (*M* 1938) Industrial Gas Engr., • Coast Counties Gas & Electric Co., 22 Pacific Ave., and 122 Davies St., Santa Cruz, Calif.

WONG, Wilfred S. B. (*M* 1938) • American Engineering Corp., 989 Bubbling Well Rd., and 669 Hart Rd., Shanghai, China.

WONSON, Arthur S., Jr. (*J* 1941; *S* 1938) Navy Yard, Boston, and • Walnut Park Ave., Essex, Mass.

WOOD, Alfred W. (*A* 1941; *J* 1938) Sales Engr., Clare Brothers & Co., Ltd., and • 451 Margaret St., Preston, Ont., Canada.

WOOD, Charles F. (*M* 1937) Air Cond. Mgr., Prod. Development and Application Dept., Frigidaire Div., General Motors Sales Corp., 300 Taylor St., Dayton, and • R.R.1, Spring Valley, Ohio.

WOODGER, Herbert W. (*M* 1939) Htg. and Vtg. Engr., • General Electric Co., 100 Woodlawn Ave., Pittsfield, and "Pineacres," East St., Lenox, Mass.

WOODHOUSE, Graham D. (*A* 1938) General Supt., Dowagiac Steel Furnace Co., and • 304 West St., Dowagiac, Mich.

WOODMAN, Lawrence E. (*M* 1934) Pres., • Woodman Engineering Corp., 203 E. Capitol, and 925 Adams, Jefferson City, Mo.

WOODS, Baldwin M. (*M* 1937) Prof. Mech. Engrg., • University of California, Engrg. Bldg., and 249 The Uplands, Berkeley, Calif.

WOODS, Charles F. (*A* 1940) Town Plant Mgr., • Texas Southwestern Gas Co., and S. Baylor St., Brenham, Tex.

WOODS, Edward H. (*M* 1934) Engr., • Higgins & Zabriskie, 134-136 S. Aurora St., and Hook Place, Ithaca, N. Y.

WOODWARD, Rothwell (*M* 1938) Air Cond. Sales Engr., Frigidaire Div., General Motors Sales Corp., 300 Taylor St., and • 1527 Benson Drive, Dayton, Ohio.

WOOLCOCK, Edwin (*A* 1938) Mgr., • Woolcock Plumbing & Heating Co., 2217 15th St., and 440 Memorial Pkwy., Niagara Falls, N. Y.

WOOLLARD, Mason S. (*M* 1934) Htg. Engr., Harry H. Angus, Consulting Engr., 1221 Bay St., Toronto, and • 31 Hillcrest Park Ave., Toronto 5, Ont., Canada.

WOOLSTON, A. H. (*M* 1919) Chief Engr., • Woolston-Woods Co., 2132 Cherry St., and 4815 North 12th St., Philadelphia, Pa.

WOOLSTON, Robert H. (*J* 1941) Htg.-Vtg. Engr., • Woolston Woods Co., 2132 Cherry St., and 358 W. Mt. Airy Ave., Philadelphia, Pa.

WOOTEN, M. Frank, Jr. (*M* 1941; *A* 1940) Consulting Engr., • 104 Latta Arcade, and 400 Cherokee Rd., Charlotte, N. C.

WORKMAN, Albert E. (*A* 1941) Air Cond. Sales Engr., Houston Div., Dept. Mgr., • United Gas Corp., United Gas Bldg., and Apt. 3, 4707 Fannin, Houston, Tex.

WORMLEY, Robert F. (*A* 1938) Branch Mgr., • Grinnell Co. of Canada, Ltd., 700 Beaumont St., and 6092 Terrebonne Ave., Montreal, Que., Canada.

WORSHAM, Herman (*M* 1925; *J* 1918) Mgr. for G.M. Accts., • Frigidaire Div., General Motors Corp., 300 Taylor St., and 524 Daytona Pkwy., Dayton, Ohio.

WORTHINGTON, Thomas H. (*M* 1937) Mgr., Eastern Htg. Sales, • Standard Sanitary & Dominion Radiators, Ltd., 405 Beaubien St. W., Montreal, Que., Canada.

WORTON, William (*M* 1937) Branch Mgr., • C. A. Dunham Co., Ltd., 504 Scott Block, and 292 Lansdowne Ave., Winnipeg, Man., Canada.

WRIGHT, Clarence E. (*A* 1940; *J* 1935; *S* 1933) Mgr., Htg. and Vtg. Dept., Fairmont Wall Plaster Co., Tenth St., and • 303 Nuzum Place, Fairmont, W. Va.

WRIGHT, Daniel K., Jr.* (*J* 1938) Instructor in Mech. Engrg., Case School of Applied Science, 10900 Euclid Ave., Cleveland, and • 3618 Lindholm Rd., Shaker Heights, Ohio.

WRIGHT, Harris H. (*M* 1917) Owner, • H. H. Wright Co., 1322 Walnut St., and 808 Greenway Terrace, Kansas City, Mo.

WRIGHT, John B. (*M* 1940) Sales Engr., • Nash Engineering Co., South Norwalk, and Rowayton, Conn.

WRIGHT, K. A. (*M* 1921) Branch Mgr., • Johnson Service Co., 1905 Dunlap St., Cincinnati, Ohio, and 113 Orchard Rd., Fort Mitchell, Ky.

WRIGHT, Norman S., Jr. (*A* 1942; *J* 1941) Sales, Norman S. Wright & Co., 250 Perry St., San Francisco, Calif.

WRIGHTSON, Wilbor T. (*M* 1937) Eastern Mgr., • Garden City Fan Co., 55 West 42nd St., New York, and 22 Sagamore Rd., Bronxville, N. Y.

WUNDERLICH, Milton S.* (*M* 1925) Chief of Research, Minnesota & Ontario Paper Co., 1100 Builders Exchange, Minneapolis, and • 545 Mt. Curve Blvd., St. Paul, Minn.

WYATT, DeWitt H. (*M* 1936) Mech. Engr., Havens & Emerson, Fort Knox, and • Taylor Hotel, Elizabethtown, Ky., and 123 Acton Rd., Columbus, Ohio.

WYLD, Reginald G. (*M* 1937) Vice-Pres. in Charge of Engrg., • Chrysler Corp., Airtemp Div., 1119 Leo St., and 736 Devonshire Rd., Dayton, Ohio.

WYLIE, Howard M. (*M* 1925; *J* 1917) Vice-Pres. in Charge of Sales, The Nash Engineering Co., and • 51 Elmwood Ave., South Norwalk, Conn.

Y

YAGER, John J. (*M* 1921) Pres., Goergen-Mackwirth Co., Inc., 817 Sycamore St., and • 425 Woodbridge Ave., Buffalo, N. Y.

YAGLOU, Constantin P.* (*M* 1923) Assoc. Prof. Industrial Hygiene, • Harvard School of Public Health, 55 Shattuck St., Boston, and 10 Vernon Rd., Belmont, Mass.

YARBOROUGH, T. R. (*M* 1938) Mgr., Commercial, Air Cond. & Htg., Charles S. Martin Co., 1041 N. Highland Ave. N.E., and • 1266 Stillwood Drive, N.E., Atlanta, Ga.

YATES, James E. (*M* 1934) Mgr., • Yates, Neale & Co., 231 Tenth St., and 431–16th St., Brandon, Man., Canada.

YATES, Joseph E. (*M* 1939) Asst. Engr., • Pacific Power & Light Co., 405 Public Service Bldg., and 1820 Northeast 57th Ave., Portland, Ore.

YATES, Robert A. (*J* 1939) Steamfitter, • Yates, Neale & Co., 231 Tenth St., and 431–16th St., Brandon, Man., Canada.

YATES, Walter (*Life Member*; *M* 1902) Governing Dir., • Matthews & Yates, Ltd., Cyclone Works, Swinton, and 4 Egerton Park, Worsley, Manchester, England.

YERKES, William L. (*M* 1941; *A* 1937) Engr., • Carrier Corp., 12 South 12th St., Philadelphia, and 156 Jericho Manor, Jenkintown, Pa.

YOUNG, Emil O. (*A* 1935) Owner, • Young Regulator Co., 4500 Euclid Ave., Cleveland, and 3628 Cummings Rd., Cleveland Heights, Ohio.

YOUNG, Forest H., Jr. (*A* 1936) Sales Engr., • Auburn Stoker Co., 512 N. LaSalle St., and 4310 Berteau Ave., Chicago, Ill.

YOUNG, Harold J. (*M* 1937) Sales Engr., Young Radiator Co., Occidental Hotel Bldg., and • 1364 Lakeshore Drive, Muskegon, Mich.

YOUNG, J. T., Jr. (*A* 1936) Mgr., • Crane Co., Box 709, 20th and Wall, and 508 Ogden Canyon, Ogden, Utah.

YOUNG, Robert W. (*A* 1941) Sales Repr., Mueller Brass Co., Port Huron, Mich., and • 700 West 47th St., Kansas City, Mo.

YOUNGER, John R. (*J* 1941) Htg. Engr., Automatic Heat Co., and • 2835 Third Ave. W., Hibbing, Minn.

Z

ZACK, H. J. (*M* 1928) Prop., • The Zack Co., 2311 Van Buren St., Chicago, Ill.

ZAKI, Hussein M. (*J* 1941; *S* 1940) Architectural Engr., 17 Cleopatra St., Heliopolis, Cairo, Egypt.

ZEMELMAN, Irving M. (*J* 1941; *S* 1939) Asst. Engr., Sears, Roebuck & Co., Port Newark, and • 87 Schuyler Ave., Newark, N. J.

ZIBOLD, Carl E. (*M* 1929) Mech. Engr., Htg. and Vtg., 13 Chadwick Rd., Westminster Ridge, White Plains, N. Y.

ZIEBER, W. E. (*M* 1935) Dir. of Research, • York Ice Machinery Corp., and 22 S. Keesey St., York, Pa.

ZIEL, Herbert E. (*M* 1924) Mech. Engr., • Albert Kahn, Associated Archts. and Engrs., 345 New Center Bldg., and 694 Glynn Court, Detroit, Mich.

ZIESSE, Karl L. (*A* 1931) Partner, • Phoenix Sprinkler & Heating Co., 115 Campau Ave. N.W., and 315 Hampton Ave. S.E., Grand Rapids, Mich.

ZIMMERMAN, A. H. (*M* 1939; *A* 1930) Ventilation Engr., Chicago Health Dept., 54 W. Hubbard St., and • 6259 N. Francisco Ave., Chicago, Ill.

ZINK, David D. (*M* 1931) Major, • Hq. 1st Arm'd. Corps (G3), Fort Knox, and 315 N. Mulberry St., Elizabethtown, Ky.

ZINTEL, George V. (*A* 1941) Sales Engr., Himelblau Byfield & Co., 36 S. Throop St., and • 1428 Summerdale, Chicago, Ill.

ZOKELT, Carl G. (*M* 1921) Consulting Engr., 3810–24th Ave. S., Seattle, Wash.

ZUBER, Otto C. (*A* 1938) Chief Engr., • Amana Society, Amana, and South Amana, Iowa.

ZUHLKE, W. R. (*M* 1028) Hoffman Specialty Co., 500 Fifth Ave., New York, N. Y.

ZUMWALT, Ross (*A* 1941; *J* 1938) Partner, Zumwalt & Vinther, 507 Thomas Bldg., Dallas, Tex.

ZUROW, William Allan (*J* 1037) Sales Engr., St. Joseph Railway Light, Heat & Power Co., 522 Francis St., and • 728 S. Tenth St., St. Joseph, Mo.

ZWALLY, August L. (*A* 1937) Chief Air Cond. Engr., Interstate Electric Co., and • 908 Elmwood St., Shreveport, La.

SUMMARY OF MEMBERSHIP

Honorary Member	1
Presidential Members	24
Life Members	60
Members	1702
Associate Members	975
Junior Members	317
Student Members	53
Total	3132

UNITED STATES AND POSSESSIONS

Alabama	12
Arizona	4
Arkansas	4
California	129
Colorado	11
Connecticut	41
Delaware	11
District of Columbia	87
Florida	11
Georgia	53
Idaho	1
Illinois	248
Indiana	37
Iowa	45
Kansas	12
Kentucky	17
Louisiana	39
Maine	4
Maryland	52
Massachusetts	101
Michigan	176
Minnesota	107
Mississippi	4
Missouri	131
Montana	3
Nebraska	21
Nevada	5
New Jersey	110
New York	416
North Carolina	43
Ohio	185
Oklahoma	17
Oregon	47
Pennsylvania	282
Rhode Island	6
South Carolina	11
South Dakota	2
Tennessee	16
Texas	98
Utah	3
Vermont	2
Virginia	45
Washington	40
West Virginia	11
Wisconsin	90
Philippine Islands	2
	2792

DOMINION OF CANADA 209

FOREIGN COUNTRIES

Australia	11
Belgium	1
Bermuda Islands	1
Brazil	2
Chile	1
China	8
Cuba	1
Denmark	2
Egypt	4
England	36
France	7
French North Africa	1
Germany	3
Holland	1
India	6
Ireland	2
Italy	6
Japan	4
Manchoukuo	1
Mexico	3
Netherlands	1
New Zealand	3
Norway	2
Palestine	1
Puerto Rico	1
Roumania	1
South Africa	6
Spain	1
Straits Settlements	2
Sweden	8
Turkey	2
Venezuela	2
	131

TOTAL MEMBERSHIP 3132

LIST OF MEMBERS

(Geographically Arranged)

UNITED STATES *and* POSSESSIONS

ALABAMA

Birmingham—
Gause, H. C.
Hardy, F. L.
Lichty, C. P.
Myer, H.
Richards, G. H.
Walden, H. K.

Fort McClellan—
Friedman, D. H., Jr.

Mobile—
Gray, W. C.

Montgomery—
Dowdy, R. B.
Drum, L. J., Jr.

Talladega—
Addington, H. M.

Tuscaloosa—
Murphree, R. L.

ARIZONA

Lowell—
Vinson, N. L.

Phoenix—
Genre, E. J.
Hummel, G. W.

Tucson—
Tidmarsh, P. M.

ARKANSAS

Hope—
Prawl, F. E.

Little Rock—
Cumnock, H.
Kellogg, W. T.
McCoy, C. E.

CALIFORNIA

Albany—
Kaup, E. O.

Bakersfield—
Baker, H. S.

Berkeley—
Atkins, G. E.
Brokaw, G. K.
Cockins, W. W.
Fluckey, K. N.
Peterson, C. L.
Porter, N. E.
Raber, B. F.
Woods, B. M.

Beverly Hills—
Theobald, A.

Burlingame—
Gee, W. W.
Hill, J. A.

Camp Haan—
Forderbruggen, K. J.

El Monte—
Hazlehurst, H. D.

Fresno—
Newman, H. E.

Fullerton—
McKinley, C. B.
Miles, C. N.

Glendale—
Eggleston, H. L.
Scofield, P. C.
Wells, E. P.

Hollywood—
Billingsley, O. F., II

Huntington Park—
Stanley, R. L.

Long Beach—
Barth, J. W.

Los Angeles—
Anderson, C. S.
Barnum, W. E., Jr.
Blumenthal, M. I.
Bullock, H. H.
Dillender, E. A.
Douglas, H. H.
Downes, A. H.
Ellingwood, E. L.
English, H.
Fabling, W. D.
Hendrickson, H. M.
Hess, A. J.
Hogue, W. M.
Hokanson, C. G.
Hungerford, L.
Kennedy, M.
Kilpatrick, W. S.
Lauer, H. B.
Lowe, R. A.
McKenzie, C. M., Jr.
Medow, J.
Moriarty, J. M.
Ness, W. H. C.
Ott, O. W.
Park, J. F.
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Robin, R. C.
Stetson, L. R.
Swaney, C. R.
Tuttle, J. F.
Waterman, J. H.
Yaglou, C. P.

Bridgewater—
Beaulieu, A. A.

Cambridge—
Flint, C. T.
Holt, J.
Peterson, C. M. F.
Saurwein, G. K.
Sheffield, R. A.
Staszesky, F. M.
Touloukian, Y. S.
Wilkes, G. B.

Chelmsford—
Ridley, W. H.

Dorchester—
Ehrenzeller, A.
Goodrich, C. F.
Hosterman, C. O.
Shaer, I. E.

Dover—
Lincoln, R. L.

East Lynn—
Lauckner, C. G., III

Essex—
Wonson, A. S., Jr.

Fitchburg—
Dolan, W. H.
Illig, E. E.
Illig, W. R.
Karlson, A. F.
McKittrick, P. A.

Harwich Port—
Maxwell, G. W.

Hyde Park—
Ellis, F. R.

Lawrence—
Bride, W. T.

Leominster—
Kern, R. T.

Lynn—
Farrow, H. L.
Feehan, J. B.
Fuller, R. A.

Malden—
Denham, H. S.
Hill, C. F.
Richards, L. V.

Melrose—
Franklin, R. S.
Gerrish, G. B.

Newton Center—
Lair, P. H.

Newtonville—
Emerson, R. R.
Jones, W. T.
Moore, H. C.

North Andover—
Wilson, A. M.

Pittsfield—
Nicholls, J. M.
Wagner, E. A.
Woodger, H. W.

Quincy—
Rodee, E. J.

Reading—
Ingalls, F. D. B.

Revere—
Brayman, A. I.

Roslindale—
Larson, C. W.

Shirley—
Boyden, D. S.

Somerville—
Scalingi, C. R.
Whitten, H. E.

South Hamilton—
Mandell, T. P.

Springfield—
Cooper, W. B.
Cross, R. E.
Holmes, R. E.
Huggins, L. G.
Malin, B. S.
Murphy, W. W.

Watertown—
Wiegner, H. B.

Wellesley Hills—
Barnes, W. E.

Westfield—
Leland, W. B.

West Newton—
Wahlin, B. J.

West Roxbury—
Krapohl, W. H.
McPherson, W. A.

Weymouth—
McCafferty, J. E.

Winchester—
Carrier, E. G.

Wollaston—
Blair, E. L.

Worcester—
Kolb, R. P.
Wechsberg, O.

MICHIGAN

Alpena—
Freestrom, A.

Ann Arbor—
Backus, T. H. L.
Kessler, C. F.
Marin, A.

Battle Creek—
Christenson, H.
Dempsey, S. J.
Dyer, W. S.

Bay City—
Gray, J. W.
Henry, E. C.
Witmer, H. S.

Birmingham—
Hadjisky, J. N.
Hyde, E. F.
Root, E. B.
Widdowfield, A. S.

Detroit—
Adam, R. W.
Anderson, E. J.
Baldwin, W. H.
Barth, H. E.
Barton, J.
Bassett, J. W.
Bay, C. H.
Beattie, J.
Berryman, R. H.
Biggers, R. H.
Bishop, F. R.
Blackmore, F. H.
Boales, W. G.
Bottum, E. W.
Busse, H.
Champlin, R. C.
Clar, R., Jr.
Clark, E. H.
Collins, L. F.
Connell, R. F.
Coon, T. E.
Cummins, G. H.
Darlington, A. P.
Dauch, E. O.
Deppmann, R. L.
Donohoe, C. F.
Dubry, E. E.
Elliott, N. B.
Falk, D. S.
Feinberg, E.
Feely, F. J.
Giguere, G. H.
Goss, M. H.
Harrigan, E. M.
Hesselschwerdt, A. L. Jr.
Heydon, C. G.
Hogan, E. L.
Hubbard, N. B.
Hughson, H. H.
Hutzel, H. F.
Johnson, F. W.
Kaufman, H. J.
Kilner, J. S.
Kincaide, M. C.
Kirkpatrick, A. H.
Knibb, A. E.
Lewis, K. C.
Linsenmeyer, F. J.
Livermore, J. N.
Luty, D. J.
Mabley, T. H.
MacMillan, A. R.
Mally, C. F.
Marzolf, F. X.
McConachie, L. L.
McCrea, J. B.
McGeorge, R. H.
McIntire, J. F.
McLean, D.
Metcalfe, C.
Milward, R. K.
Morse, C. T.
Morse, L. S., Jr.
Oberschulte, R. H.
O'Gorman, J. S., Jr.
Old, W. H.
O'Rourke, H. D., Jr.
Paetz, H. E.
Parrott, L. G.
Partlan, R. L.
Pavey, C. A.
Purcell, F. C.
Randall, R. D.
Randall, W. C.
Reader, J. T.
Sanford, S. S.
Schmidt, K., Jr.
Shea, M. B.
Sheley, E. D.
Smith, W. O.
Snyder, J. W.
Soeters, M.
Spitzley, R. L.
Spurgeon, J. H.
Stites, R., Jr.
Strand, C. A.
Taylor, H. J.
Toonder, C. L.
Tuttle, G. H.
Waid, G. H.
Walker, J. H.
Weinert, F. C.
Whelan, W. J.
White, E. S.
Wilde, R. S. M.
Winans, G. D.
Ziel, H. E.

Dowagiac—
Cunningham, J. S
Deming, R. E.
Harden, J. C.
Torr, T. W.
Woodhouse, G. D.

East Lansing—
Ely, R. S.
Kelly, O. A.
Merz, R. A.
Miller, L. G.
Pesterfield, C. H.

Ferndale—
Gair, K. B.

Flint—
Hendriksen, L.

Grand Rapids—
Boot, A.
Bradfield, W. W.
Bratt, H. D.
Hall, T. A.
Marshall, O. D.
Morton, C. H.
Stafford, T. D.
Thoman, E. O.
Todd, S. W., Jr.
Warren, F. C.
Ziesse, K. L.

Grosse Pointe—
Buckeridge, V. L.
Kaiser, F.
Jordan, F. O.
Snyder, E. F., Jr.

Grosse Pointe Park—
Davis, G. L., Jr.

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Voorhees, G. A.

Highland Park—
Harrower, W. C.

Hillsdale—
Oberlin, J. A.

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DeRoo, W. C.
Harbin, F., Jr.

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Seeber, R. R.

Jackson—
Link, C. H.
Tilford, L. A.

Kalamazoo—
Brinker, H. A.
Brundage, F. W.
Downs, S. H.
McConnor, C. R.
Metzger, H. J.
Schlichting, W. G.
Temple, W. J.
Wilson, R. W.

Lansing—
Clemens, J. D.
Distel, R. E.
Hill, V. H.
McLouth, B. F.
Miller, J. W.
Parsons, R. A.
Robinson, K. E.

Lawrence—
Morton, P. S.

Marquette—
Bernhard, G.

Mt. Clemens—
Bailey, E. P.

Muskegon—
Young, H. J.

Muskegon Heights—
Reid, H. F.

Okemos—
Foote, J. H., Jr.

Plymouth—
Lee, J. A.

Pontiac—
Trzos, O. A.

Royal Oak—
Burch, L. A.
Keyser, H. M.

Saginaw—
Godfrey, J. E.
Witheridge, D. E.

Three Rivers—
Hall, C. H.

Wayland—
Snook, A. H.

MINNESOTA

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Swanson, E. C.

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Paine, H. A.

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Pauley, R. D.

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Foster, C.

Hibbing—
Bispala, J. T.
Younger, J. R.

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Algren, A. B.
Bell, E. F.
Bensen, C. L.
Betts, H. M.
Bjerken, M. H.
Bredesen, B. P.
Burns, E. J.
Burritt, C. G.
Burritt, E. E., Jr.
Caple, I.
Carlson, C. O.
Chalmers, C. H.
Copperud, E. R.
Cottrell, W. H.
Cumming, F. J.
Dahlstrom, G. A.
Davidson, J. C.
Dever, H. F.
Evans, R. W.
Forfar, D. M.
Gausewitz, W. H.
Gerrish, H. E.
Gorgen, R. E.
Gross, L. C.
Haines, J. E.
Hanson, L. C.
Hanson, L. P.
Harris, J. B.
Hayes, O. J.
Helstrom, H. G.
Herman, N. B.
Hitchcock, P. C.
Huch, A. J.
Jordan, R. C.
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Knowles, E. L.
Lange, F. F.
Larson, C. P.
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Legler, F. W.
Lilja, O. L.
Lund, C. E.
Marshall, S. C.
McCormick, G. W., Jr.
McDonald, T.
Mills, H. C.
Mitchell, J. G.
Morgan, G. C.
Nessell, C. W.
Newton, A. B.
Nicols, J. A.
Orr, G. M.
Petersen, B. W.
Petersen, C. P.
Proebstle, L.
Reisberg, L. K.
Resch, R. J.
Roberts, H. P.
Rosas, M. L.
Rowley, F. B.
Rowley, R. K.
Schad, C. A.
Schultz, A. W.
Seelert, E. H.
Stafford, J. F.
Stiller, F. W.
Sutherland, D. L.
Swanson, D. F.
Swenson, J. E.
Tupper, E. B.
Uhl, E. J.
Uhl, W. F.
Willis, L. L.

Nicollet Island—
Schernbeck, F. H.

Owatonna—
Anderson, G. A. M.

Robbinsdale—
Hyde, L. L.

Rochester—
Adams, N. D.

St. Paul—
Anderson, D. B.
Backstrom, R. E.
Bauer, A. E.
Bean, G. S.
Colman, R. C.
Craig, J. A.
Diamond, D. D.
Estes, E. C.
Gausman, C. E.
Hickey, D. W.
Hyde, E. V.
Jones, E. F.
McNamara, W.
Oberg, H. C.
Otto, R. W.
Persson, N. B.
Peterson, J. R.
Ruff, D. C.
Sturm, W.
Winterer, F. C.
Wunderlich, M. S.

Spring Park—
Streater, E. C.

Wayzata—
Hughes, S.

White Bear Lake—
Sanford, A. L.

MISSISSIPPI

Camp Shelby—
Sterner, D. S.

Jackson—
Light, J. C.
Ludlow, H. M.

Tupelo—
Dabbs, J. T.

MISSOURI

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DuBois, L. J.
Weber, E. F., Jr.

Ferguson—
Szombathy, L. R.

Ft. Leonard Wood—
Cheeseman, E. W.

Glendale—
Nelson, C. L.

Independence—
Cook, B. F.

Jefferson City—
Woodman, L. E.

Joplin—
Jones, J. T.
McMullen, E. W.
Satterlee, H. A.

Kansas City—
Arthur, J. M., Jr.
Ball, W.
Banner, F. L. D.
Barnes, A. R.
Betz, H. D.
Boyd, L. E.
Caleb, D.
Campbell, E. K.
Campbell, R. P.
Cassell, W. L.
Clegg, C.
Dean, F. J., Jr.
Dean, M. H.
Dodds, F. F.
Downes, N. W.
Farber, L. M.
Fehlig, J. B.
Fehlig, J. B., Jr.
Flarsheim, C. A.
Forslund, O. A.
Gillham, W. E.
Harbordt, O. E.
Jones, T. S.
Kitchen, J. H.
Leffel, P. C.
Mahon, C. A.
Marchio, E., Jr.
Mart, L. T.
Mason, R. B.
Millis, L. W.
Nottberg, G.
Nottberg, H. J.
Nottberg, H., Jr.
O'Dower, H. J.
Parks, V. H., Jr.
Pellmounter, T.
Pexton, F. S.
Rivard, M. M.
Russell, W. A.
Ryan, J. B.
Selig, E. T., Jr.
Sheppard, F. A.
Smith, H. C.
Stephenson, L. A.
Stevens, K. M.
Tobin, T. A.
Weiss, C. A.
White, H. S.
White, W. E.
Wright, H. H.
Young, R. W.

Kirkwood—
Lautz, F. A.
Schwartz, N. E.

Normandy—
Dulle, W. L.

North Kansas City—
Nelson, A. A.

Overland—
Cain, W. J.

Richmond Heights—
Noord, D. F.
Siegel, D. E.
Spelbrink, R. G.

St. Joseph—
Harton, A. J.
Zurow, W. A.

St. Louis—
Ahrens, C. F.
Bayse, H. V.
Blackmore, J. J.
Boester, C. F.
Bradley, E. P.
Burnap, C. H.
Carlson, E. E.
Carter, J. H.
Clarkson, J. R.
Cooper, J. W.
Corrigan, J. A.
Davis, C. R.
Driemeyer, R. C.
Dreher, L. F.
Evans, B. L.
Fagin, D. J.
Falvey, J. D.
Gilmore, L. A.
Grossmann, H. A.
Grossenbacher, H. E.
Haller, A. L.
Hamig, L. L.
Hester, T. J.
Horch, G. E.
Kella, W. B.
Killoren, D. E.
Kuntz, E. C.
Laskaris, N. G.

Laufketter, F. C.
Malone, J. S.
McLarney, H. W.
McMahon, T. W.
Moon, L. W.
Norris, W. P.
Oonk, W. J.
Peiser, M. B.
Pellegrini, L. C.
Rodenheiser, G. B.
Rosebrough, J. S.
Saenger, L. W.
Sampson, W. D.
Sanders, C. M., Jr.
Scherrer, L. B.
Simons, B. C.
Sodemann, P.
Sodemann, W. C. B.
Stammer, E. L.
Steckhan, L.
Tenkonohy, R. J.
Toensfeldt, R.
Weiss, W. G.

University City—
Carlock, M. F.
Sharp, H. C.
Smith, R. C.

Webster Groves—
Hartwein, C. E.
Hyde, D. F.
Jones, E. A.
Myers, G. W. F.
Ronsick, E. H.
Rosebrough, R. M.

MONTANA

Billings—
Cohagen, C. C.

Butte—
Sullivan, T. J.

Great Falls—
Ginn, T. M.

NEBRASKA

Hastings—
Swingle, W. T.

Lincoln—
Hellmers, C. C., Jr.
Lehman, M. G.
Stanton, H. W.
Williams, D. D.

Omaha—
Adams, E. F.
Eaton, B. K.
Furber, S. L.
Green, E. W.
Kleinkauf, H.
Mathis, J.
McCulley, D. E.
Merwin, G. E.
Olson, M. J.
Peterson, B. G.
Reifschneider, J.
Saxon, R. B.
White, W. R.
Wiser, C. E.

Schuyler—
Matousek, A. G.

Scottsbluff—
Davis, O. E.

NEVADA

Reno—
Humes, W. E.
Mast, C. M.
Stevens, H. R.
Stevens, J. E.
Thompson, C.

NEW JERSEY

Asbury Park—
Strevell, R. P.

Atlantic City—
Strouse, S. B.

Bayonne—
Schwartz, J.

Belleville—
Thornton, T. L.

Blackwood—
Weid, H. L.

Bloomfield—
Albright, C. B.
Austin, W. H.
Berry, R. U.
Burges, J. H. M.
Faust, F. H.
Jackes, H. D.
Kelsey, H. D.
McLenegan, D. W.
Tenney, D.

Bogota—
Griess, P. G.
Mumford, A. R.

Camden—
Brown, W. M.
Coward, C. W.
Hynes, L. P.
Kappel, G. W. A.
Plum, L. H.
Webster, E. K.
Webster, W., Jr.

Chatham—
Maddux, O. L.

Cliffside Park—
Butler, P. D.

Closter—
Steinke, B. J.

Collingswood—
Mohrfeld, H. H.

Deal—
Fineran, E. V.

East Orange—
Ferguson, R. R.
Schroth, A. H.
Tallmadge, W.
Turno, W. G. W.
Wildman, E. L.

Elizabeth—
Anderson, J. W.
Bermel, A. H.
Cornwall, G. I.

Englewood—
Hendrickson, W. B.

Erlton—
Powers, E. C.

Essex Fells—
Ekings, R. M., Jr.
Soule, L. C.
Stacey, A. E., Jr.

Fort Dix—
King, J. S.
Sharp, J. R.
Sobel, F.

Freehold—
Buck, D. T.

Haddonfield—
Buzzard, F. H.
Matthews, J. E.

Hasbrouck Heights—
Goodwin, S. L.
Hoecker, G. F., Jr.

Hoboken—
Dick, H. S.

Irvington—
Feldermann, W.
Reinke, A. G.
Stengel, F. J.

Jersey City—
Jones, H. L.
Kunzog, T. W.
Ritchie, W.
Walterthum, J. J.

Kearney—
Shaffer, C. E.

Kenvil—
Johnston, R. M.

Leonia—
Close, R.

Lyndhurst—
Ehrlich, M. W.

Maplewood—
Settelmeyer, J. T.

Merchantville—
Binder, C. G.

Montclair—
Broome, J. H.
Jacobus, D. S.

Newark—
Bryant, P. J.
Carey, P. C.
Conklin, R. M.
Haitmanek, L. M.
Kruse, W. C., Jr.
Kucinski, W. V.
Leinroth, J. P.
Morehouse, H. P.
Ray, L. B.
Raymer, W. F., Jr.
Steinmetz, C. W. A.
Terhune, R. D.
Zemelman, I. M.

New Brunswick—
Wolin, M. W.

Nutley—
Morris, C. R.

Orange—
Crawford, J. H., Jr.
Moraweck, A. H., Jr.

Passaic—
Bichowsky, F. R.
Blazer, B. V.

Paterson—
Cox, H. F.
Frank, O. E.
McBride, J. N.

Perth Amboy—
Simkin, M.

Plainfield—
Pond, W. H.
Reis, R.

Red Bank—
Isaacs, H. A., Jr.
Miner, H. H.

Ridgewood—
Fitts, J. C.
Miller, A. T.
Walton, C. W., Jr.

South Orange—
Anderson, E. L.
Browne, A. L.

Summit—
Oaks, O. O.

Teaneck—
Heebner, W. M.
Roy, A. C.

Union—
Lyman, S. E.

Union City—
Taverna, F. F.

Verona—
Boxall, F.

Vineland—
Houck, R. C.

West Engleton—
Hutcheon, C. R.

Westfield—
Hansler, J. E.

Westmont—
Robinson, E. R.

West Orange—
Adlam, T. N.
Snavely, E. R.

Woodbury—
Schmieler, J. B.

NEW YORK

Albany—
Bond, H. A.
Murray, T. F.
Mytinger, K. L.
Taggart, R. C.
Taylor, R. B.
Teeling, G. A.
Westover, W.

Big Flats—
Davis, B. C.

Bronxville—
Bishop, C. R.
Dornheim, G. A.
Groves, S. A.

Brooklyn—
Balsam, C. P.
Bieringer, F. A.
Bigolet, L.
Brex, I. E.
Deier, B.
Dutcher, H. S.
Dwyer, T. F.
Fidelius, W. R.
Goldberg, M.
Heymsfield, H. R.
Hollister, N. A.
Jacobi, B. A.
Janet, H. L.
Josephson, S.
Kreiner, J.
Levine, C.
Levine, L. J.
London, M.
Phillips, F. W.
Ritchie, E. J.
Rodman, R. W.
Schechter, J. E.
Thornburg, H. A.
Tusch, W.
Vervoort, E. L.
Wachs, L. J.
Weissblatt, N.

Buffalo—
Adema, G. E.
Beman, M. C.
Berringer, S. H.
Booth, C. A.
Cherry, L. A.
Cheyney, C. C.
Constant, E. S.
Curry, R. F.
Davis, J.
Day, H. C.
Drake, G. M.
Evans, E. C.
Farnham, R.
Farrar, C. W.
Harding, L. A.
Hawk, J. K.
Heath, W. R.
Heckel, E. P., Jr.
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Hirschman, W. F.
Hurwich, S. B.
Jackson, M. S.
Kamman, A. R.
Landers, J. J.
Lenihan, W. O.
Lighthart, C. H.
Long, H. P.
Love, C. H.
Madison, R. D.
Mahoney, D. J.
Moesel, F. A.
Mosher, C. H.
Peterson, D. J.
Pleuthner, R. L.
Quackenbush, S. M.
Reif, A. F.
Reif, C. A.
Roebuck, W., Jr.
Schmidt, H.
Seelbach, H., Jr.
Seyfang, W. G.
Siegel, R. C.
Sommers, W. J.
Spencer, W. E.
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Walker, E. R.
Wendt, E. F.
Whitt, S. A.
Yager, J. J.

Croton—
Elliott, I.

Dewitt—
Keane, G F.

Dunkirk—
Mayne, W. L.

Eden—
Ensign, W. A.
Lewis, H. F.

Fayetteville—
Des Reis, J. F.

Hamberg—
Schafer, H. C.

Hastings-on-Hudson
Hewett, J. B.
Reynolds, T. W.

Irvington—
Bastedo, A. E.

Ithaca—
Elwood, W. H.
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Sawdon, W. M.
Williams, J. W.
Woods, E. H.

Kenmore—
Candee, B. C.
Criqui, A. A.
Mollenberg, H. J.
Mould, H. W., Jr.
Strouse, S. W.
Tillinghast, H. S.

Kodak Park—
Cook, R. P.

Larchmont—
Tiltz, B. E.

Lockport—
Saunders, L. P.

Long Island—
Alt, H. L. (St. Albans)
Apt, S. R. (Bayside)
Bachmann, A. J. (Ridgewood)
Bastedo, G. R. (Richmond Hill)
Birkett, H. S. (Roslyn Heights)
Blackburn, E. C., Jr. (Garden City)
Bloom, L. (Freeport)
Bonthron, R. C. (Hempstead)
Carbone, J. H. (St. Albans)
Carroll, D. E. (Woodside)
Chase, C. L. (Manhasset)
Crone, T. E. (Jamaica)
Eastwood, H. F. (Merrick)
Eschenbach, S. P. (Bayside)
Fisher, J. T. (Flushing)
Franck, P. (Jackson Heights)
Freeman, A. W. (Jackson Heights)
Fritz, C. V. (Freeport)
Fuller, E. W. (Woodhaven)
Gilman, F. W. (Long Island City)
Gornston, M. H. (Jamaica)
Graber, E. (Douglaston)
Hiers, C. R. (Great Neck)
Hirsch, M. H. (Forest Hills)
Holland, G. R. (Elmhurst)
Jalonack, I. G. (Hempstead)
Kadel, G. B. (Valley Stream)
Kahn, C. R., Jr. (Woodmere)
Keller, G. A. (Wantagh)
Kern, J. F., Jr. (Massapequa)
Lane, D. D. (Elmhurst)
Lang, J. (Richmond Hill)
Langberg, M. (Jackson Heights)
LaRaus, J. (Jackson Heights)
Lawrence, F. D. (Jackson Heights)
MacWatt, D. A. (Plandome)
Magnusson, N. (Jamaica)
Marino, F. A. (Corona)
Matzen, H. B. (Rockville Centre)
Meyer, C. L. (Hollis)
Olsen, G. E. (Arverne)
Pabst, C. S. (Woodhaven)
Potter, J. R. (Long Beach)
Quirk, C. H. (Hempstead)
Raynis, T. (Manhasset)
Rose, J. C. (Hollis)
Rudd, D. J. (Babylon)
Schwartz, M. (Far Rockaway)
Sherbrooke, W. A. (Kew Gardens)
Spoerr, F. F. (Jamaica)
Stangle, W. H. (Forest Hills)
Sterne, C. M. (Long Island City)
Thomson, T. N. (Huntington)
Tucker, F. N. (Freeport)
Wallace, G. J. (East Elmhurst)
Wheeler, J., Jr. (Manhasset)

Mt. Vernon—
Brod, B. M.
Cullin, W. W.
Freitag, F. G.
Herske, A. R.
Morse, F. W.

New Rochelle—
Abrams, A.
Farley, W. F.
Fitz, J. C.
Mank, M.
Rose, H. J.
Virrill, G. A.

New York—
Adams, E. E.
Addams, H.
Addington, H. B.
Akers, A. W.
Akers, G. W.
Ambrose, E. R.
Ashley, E. E.
Baker, R. H.
Baker, T.
Barbieri, P. J.
Baum, A. L.
Bearman, A. A.
Beebe, F. E. W.
Bennett, E. A.
Berman, L. K.
Bodinger, J. H.
Bolton, R. P.
Bond, H. H.
Borak, E.
Brown, D.
Buensod, A. C.
Carpenter, R. H.
Chester, T.
Citron, D. J.
Clay, W.
Cucci, J.
Darts, J. A.
Davis, C.
Davison, R. L.
Demarest, R. T.
Denny, H. R.
Deterling, W. C.
Dodge, H. A.
Donnelly, R.
Downs, C. R.
Eadie, J. G.
Engle, A.
Feldman, A. M.
Fiedler, H. W.
Firestone, M. T.
Fleisher, W. L.
Foster, J. G.
Friedman, F. J.
Friedman, M.
Friend, W. F.
Fuller, C. A.
Gezari, Z.
Giannini, M. C.
Goldsmith, E.
Gordon, P. B.
Greenburg, L.
Gumaer, P. W.
Hall, C. J.
Hanburger, F. W.
Harrigan, E. R.
Heckler, S.
Heibel, W. E.
Heller, J. A.
Henry, A. S., Jr.
Hering, A.
High, J. M.
Hinrichsen, A. F.
Hobbie, E. H.
Hoffman, C. S.
Honerkamp, F.
Hosking, H. L.
Hotchkiss, C. H. B.
Hyman, W. M.
Iverstrom, C.

James, J. W.
Janos, W. A.
Jarcho, M. D.
Johnson, H. S.
Kaczenski, C.
Kelly, C. J.
Kepler, D. A.
Kimball, D. D.
Kingsland, G. D.
Koehler, C. S.
Kuhlmann, R.
Kunen, H.
Kurth, F. J.
Lee, B. H.
Lucke, C. E.
Markush, E. U.
Martin, G. W.
Matheka, C. R.
McCann, F. D.
McClintock, W.
McKitrick, W. D.
Meinke, H. G.
Meyer, H. C., Jr.
Milener, E. D.
Miller, C. A.
Miller, J.
Milligan, D. G.
Mollander, E. D.
Morro, J. J.
Munier, L. L.
Munkelt, F. H.
Offner, A. J.
Oldes, W. E. A. E.
Olson, R. G.
Olvany, W. J.
Page, V. C.
Parker, D. F.
Patorno, S. A. S.
Place, C. R.
Pohle, K. F.
Pollak, R.
Pryke, J. K. M.
Purinton, D. J.
Rainson, S. J.
Raisler, R. K.
Rather, M. F.
Reid, H. P.
Reynolds, W. V.
Ritter, A.
Rosenberg, P.
Ross, J. O.
Roth, C. F.
Ryan, H. J.
Salter, E. H.
Samuels, S.
Satlow, A.
Sawhill, R. V.
Schoepflin, P. H.
Scott, G. M.
Seelig, A. E.
Self, V. F.
Sellman, N. T.
Senior, R. L.
Sklenarik, L.
Smith, M. S.
Stellwagen, F. G.
Sternberg, E.
Stevens, E. K.
Strock, C.
Strunin, J.
Syska, A. G.
Taylor, F., Jr.
Timmis, W. W.
Trambauer, C. W.
Tuthill A. F.
Tyler, R. D.
Vetlesen, G. U.
Vroome, A. E.
Wadsworth, R. H.
Walford, L. C. A.
Waring, J. M. S.
Wasson, R. A.
Webster, C. C.
Wheller, H. S.
Wilder, H. P.
Williams, H. E.
Willner, I.
Wrightson, W. T.
Zuhlke, W. R.

Niagara Falls—
Woolcock, E.

North Tarrytown—
Weiss, A. P.

North Tonawanda—
Conaty, B. M.

Ossining—
Dee, L. H.

Plattsburg Barracks—
Grabman, H. B.

Poughkeepsie—
Bertolette, C., Jr.

Rochester—
Baker, H. L., Jr.
Betlem, H. T.
Dickason, G. D.
Hakes, L. M.
Hutchins, W. H.
Kimmell, P. M.
King, R. L.
Leonhard, L. W.
Lewis, C. E.
Stacy, S. C.
Treadway, J. Q.
Vidale, R.
Wilder, E. L.
Whittaker, W. K.
Willson, F. J.

Rome—
Lynch, W. L.

Scarsdale—
Cumming, R. W.

Schenectady—
Harrington, E.
Hunziker, C. E.
Welch, L. A., Jr.

Snyder—
Gifford, C. A.

Staten Island—
Brown, H. J.
Callahan, P. J.
Frimet, M.
Johnson, E. B.
Pfuhler, J. L.
Vivarttas, E. A.
Volkhardt, A. N.
Waechter, H. P.

Syracuse—
Ashley, C. M.
Carrier, W. H.
Cherne, R. E.
Day, V. S.
Driscoll, W. H.
Dunne, R. V. D.
French, D.
Fritzberg, L. H.
Goldmann, P.
Graham, W. D.
Hockensmith, F. E.
Ingels, M.
Lewis, L. L.
Lyle, J. I.
Lyon, D. M.
Mohsin, S.
Murphy, E. T.
Noble, M.
Schoeffter, H. M.
Sheldon, N. E.
Silvera, A.
Snyman, G. C.
Taliaferro, R. R.
Traynor, H. S.
Woese, C. F.

Tarrytown—
d'Issertelle, H. G.

Tonawanda—
Karlsteen, G. H.

Tuckahoe—
Goulding, W.

Utica—
Randolph, H. F.
Steinhorst, T. F.

Valhalla—
Mehne, C. A.

Watervliet—
Schubert, A. G.

Westfield—
Van Alsburg, J. H.

White Plains—
Belsky, G. A.
Guler, G. D.
Zibold, C. E.

Williamsville—
Morgan, R. W.

Yonkers—
Eggers, W. K.
Flink, C. H.
Goerg, B.
Harmonay, W. L.
Ormiston, J. B.
Rainger, W. F.
Werker, H.

Yorktown—
Gitterman, H.

NORTH CAROLINA

Camp Davis—
Barnes, H. S.

Charlotte—
Baber, J. E.
Bailey, J. L.
Brandt, E. H., Jr.
Hill, H. H.
Hodge, W. B.
Lee, R. T.
Muirheid, J. G.
Petty, C. E.
Warren, R. M., Jr.
Wooten, M. F., Jr.

Cherry Point—
Rock, G. A.

Durham—
Altemueller, G. F.
Cooke, T. C.
Daddario, F. T.
Nicholson, S. J.
Reed, F. J.
Skinner, A., Jr.
Theiss, E. S.
Wallace, W. M., II

Fort Bragg—
Croley, J. G.
Kummer, C. J.

Greensboro—
Adams, C. Z.
Guest, P. L., Jr.
Harding, E. R.
Hoffman, H.
Klages, F. E. P.
Oertel, F. H. E.

High Point—
Gray, H. E.
Gray, W. E.
Lewis, W. W.

Morehead City—
Abramson, R. J.

Newport—
Long, E. J.

Raleigh—
Rice, R. B.
Vaughan, L. L.

Winston-Salem—
Andrews, W. G.
Bahnson, F. F.
Brown, M. D.
Cornwall, C. C.
Marshall, J.
McCallum, C. E.
Page, A.

Wrighsville Beach—
Burr, G. C.

OHIO

Akron—
Curl, R. S.

Amsterdam—
Allensworth, J. E.

Ashland—
Rybolt, A. L.

Chagrin Falls—
Guilbert, S. R.
Wilson, R. A.

Cincinnati—
Bechtol, J. J.
Bird, C.
Black, J. M.
Blum, R. J., Jr.
Boyd, T. D.
Buenger, A.
Coombe, J.
Cott, W. B.
Crombie, J.
Edwards, A. W.
Ellis, G. P.
Gannon, R. R.
Green, W. C.
Hard, A. L.
Helburn, I. B.
Houlis, L. D.
Hudepohl, L. F.
Huelsman, A. G.
Hust, C. E.
Hutchinson, B. L., Jr.
Juergens, W. A.
Junker, W. H.
Kiefer, C. J.
Killian, W. J.
Kinney, A. M.
Kramig, R. E., Jr.
Kuempel, L. L.
Leupold, G. L.
Lewis, H. K.
Mathewson, M. E.
McNamee, E. W.
Moore, H. W.
Motz, O. W.
Mursinna, G. P.
Pfriem, P. G.

Pillen, H. A.
Pistler, W. C.
Richard, E. J.
Royer, E. B.
Ruff, A. G.
Sigmund, R. W.
Silberstein, B. G.
Sproull, H. E.
Stevens, W. R.
Sutfin, G. V.
Thompson, E. B.
Ward, F. J.
Washington, L. W.
Williams, E. C.
Wright, K. A.

Cleveland—
Auer, G. G.
Baggaley, W.
Beach, W. R.
Berger, J. L.
Borkat, P.
Cohen, P.
Conner, R. M.
Cummings, R. J.
Curtis, H. F.
Cushing, C. F.
Downe, E. R.
Evans, W. A.
Eveleth, C. F.
Findley, H. N.
Fleming, P. B.
Friedman, A.
Gayman, P. D.
Geltz, R. W.
Gottwald, C.
Gray, E. W.
Hall, N. H.
Harvey, L. C.
Heisterkamp, H. W.
Jones, J. P.
Kaercher, C. M. H.
Kitchen, F. A.
Klie, W.
LaSalvia, J. J.
Levy, M. I.
Machen, J. T.
McKeeman, C. A.
McLeish, W. S.
Moore, W. R.
Nachman, G. P.
Nobis, H. M.
Novotney, T. A.
Platz, J. F.
Pogalies, L. H.
Powers, L. G.
Priester, G. B.
Quick, B. A.
Reich, J. G.
Repp, H. L.
Rowe, W. M.
Slawson, L. E.
Smith, W. D.
Southmayd, R. T.
Taze, D. L.
Tuve, G. L.
Vanderhoof, A. L.
Webb, E. C.
Weldy, L.
Wetzell, H. E.
Young, E. O.

Cleveland Heights—
Cady, E. F.
Clark, R. L.
Dickens, L. A.
Rhoton, W. R.

Columbus—
Allonier, H. R.
Armstrong, E. T.
Benson, M. L.
Breneman, R. B.
Brown, A. I.
Engdahl, R. B.
Gowdy, A. C.
Leiby, R. S.
Myler, W. M., Jr.
Noble, J. P.
Oelgoetz, J. F.
Sherman, R. A.
Slemmons, J. D.
Williams, A. W.

Cortland—
Swenehart, D. W.

Cuyahoga Falls—
Humphrey, D. E.
Ray, G. E.

Dayton—
Anoff, S. M.
Baker, I. C.
Brown, J. S., Jr.
Brown, T.
Chapman, W. A., Jr.
Gibbons, M. J.
Gonzalez, R. A.
Kucher, A. A.
Lindsay, G. W., Jr.
Livar, A. P.
Smith, N. J.
Weaver, J. O.
Woodward, R.
Worsham, H.
White, E. D.
Wyld, R. G.

East Cleveland—
Geiger, R. L.
Stark, W. E.
Steffner, E. F.

Elyria—
Maynard, J. E.

Euclid—
Reardon, J. F.
Wilhelm, J. E.

Fairfield—
Linebaugh, J. E.

Hamilton—
Thomas, L. G. L.
Thomas, R. H.

Kent—
Saginor, S. V.

Lakewood—
Longcoy, G. B.
Schurman, J. A., Jr.

Lima—
Hawisher, H. H.

Lorain—
Jackson, W. F.

Middletown—
Byrd, T. I.
Somers, W. S.
Stewart, R. S.

Newark—
Waggoner, J. H.

Norwood—
Braun, J. J.

Oberlin—
Ries, L. S.
Sable, E. J.

Painesville—
Hobbs, J. C.

Piqua—
Lange, R. T.

Shaker Heights—
Avery, L. T.
Foley, J. L.
Wright, D. K., Jr.

South Park—
Barney, W. E.

Springfield—
Hauck, E. L.

Spring Valley—
Wood, C. F.

Toledo—
Hall, J. R.
Jones, S.
Terry, M. C.
Watkins, G. B.

University Heights—
Mannen, D. E., Jr.

Waverly—
Armbruster, F. T. W.

Wilmington—
Marconett, V. G.
Sapp, C. L.

Youngstown—
Montgomery, J. R.
Sgambati, A. P.

OKLAHOMA

Lawton—
MacGregor, C. M.

Norman—
Dawson, E. F.

Oklahoma City—
Braniff, P. R.
Carnahan, J. H.
Carroll, W. M.
Dolan, R. G.
Gray, E. W.
Loeffler, F. X., Sr.
Mideke, J. M.
Morin, A. R.
Rolland, S. L.
Tiller, L.

Tulsa—
Dean, C. H.
Holmes, A. D.
Jones, E.
Meinholtz, H. W.
Mumford, W. W.

OREGON

Corvallis—
Willey, E. C.

Eugene—
Chase, P. S.

Portland—
Armstrong, C. E.
Banta, G. L.
Brissenden, C. W.
Burtchaell, J. T.
Burton, W. R.
Byrne, J. J.
Campau, W. R.
Carroll, E. E.
Chase, R. E., Jr.
Clark, A. C.
Enders, C. E.
Farnes, B. W.
Finnigan, W. T.
Fox, W. K.
Freeman, J. A.
Gehrs, W.
Goehler, E. E.
Gribbon, J. H.
Hanthorn, W.
Harrington, L. J.
Heinkel, C. E.
Hoey, J. K.
Kollas, W. J.
Kroeker, J. D.
Lynch, J. R.
McClung, T. H.
McIndoe, J. F.
Mead, H. K.
Moore, B. W.
Morrison, W. B.
Murhard, E. A.
Neubauer, E. W.
Nielsen, H. B.
Ponder, E.
Richardson, L. S.
Risley, G. H.
Taylor, T. E.
Turner, E. S.
Urban, F. F.
Watson, K. W.
Weller, A. K.
Widmer, W. J.
Yates, J. E.

Salem—
Cooper, D. E.
Van Wyngarden, J.E.

PENNSYLVANIA

Abington—
Bigelow, E. S.
Park, N. W.

Aldan—
Mulcey, P. A.

Allentown—
Goundie, J. K.
Hersh, F. C.
Hilder, F. L.
Korn, C. B.

Ambler—
McElgin, J. W.

Bala—
Stewart, J. P.
Wearanga, R. R.

Bethlehem—
Curley, E. I.
Murnin, E. A., Jr.
Stuart, M. C.
Warner, C. F.

Blairsville—
Mabon, J. E.

Bradford—
Paterson, F. C., Jr.

Brookline Park—
Call, J.

Bryn Mawr—
McIlvaine, J. H.

Dravosburg—
Marshall, A. W.

Drexel Hill—
Mather, H. H.
Matz, G. N.

Dunbar—
Sherwood, L. T.

East Pittsburgh—
Hazlett, T. L.
Penney, G. W.

Elizabethtown—
Dibble, S. E.

Erie—
Joyce, H. B.

Freeport—
McCullough, J. L.

Glenmoore—
Gant, H. P.

Glenside—
Tucker, L. A.

Harrisburg—
Eicher, H. C.
Geiger, I. H.
Herre, H. A.

Haverford—
Arnold, R. S.

Jenkintown—
Buck, L.

Johnstown—
Hunter, L. N.
Knowles, F. R.

Kennett Square—
Battan, S. W.

Kingston—
Macdonald, D. B.
McGown, F. H., Jr.

Lancaster—
Jones, A.
Lloyd, E. C.
Weitzell, P. H.

Lansdowne—
James, H. R.
Seltzer, P. A.

Manheim—
Weitzel, C. B.

McKeesport—
Dugan, T. M.

Merion—
Haynes, C. V.

Middletown—
Locke, R. A.

Milton—
Mohn, H. L.

Narberth—
Grant, W. A.
Searle, W. J., Jr.
Wilmot, C. S.

New Hope—
Davidson, P. L.

New Kensington—
Edwards, J. D.

Newtown—
Lewis, T.

North Irwin—
Cost, G. W.

Oakmont—
Lieberman, M. S.

Oxford—
Ware, J. H., III

Philadelphia—
Bachman, F.
Ballman, W. H.
Barnard, M. E.
Barr, G. W.
Bartlett, C. E.
Belford, L. duB.
Black, E. N., III
Black, H. G.
Blankin, M. F.
Bogaty, H. S.
Bornemann, W. A.
Caldwell, A. C.
Carney, E. J.
Cassell, J. D.
Childs, L. A.
Cody, H. C.
Crew, F. D.
Culbert, W. P.
Dafter, E. H.
D'Ambly, A. E.
Davidson, L. C.
Dietz, C. F.
Dome, A. G.
Donovan, W. J.
Eastman, C. B.
Elliot, E.
Erickson, H. H.
Essley, H. A.
Faltenbacher, H. J.
Farrington, S. E.
Galligan, A. B.
Goff, J. A.
Graham, J. J.
Hedges, H. B.
Hibbs, F. C.
Hucker, J. H.
Hunger, R. F.
Hutchison, J. E.
Ickeringill, J. C.
Jacobsen, K. C. S.
Jakoby, A. C.
Jardine, W. H.
Keeling, F. V.
Kelble, F. R.
Killough, R. E.
Kirkbride, J. O.
Kohler, J. C.
Kniebel, A. E.
Ladd, D.
Landau, M.
Lauer, R. F.
Lee, R. J.
Leonard, R. R.
Leopold, C. S.
Lyon, P. S.
Mack, L.
Macrow, L.
Makin, H. T., Jr.
Martocello, J. A.
McCullough, H. G.
Mellon, J. T. J.
Meloney, E. J.
Mensing, F. D.
Millham, F. B.
Moody, L. E.
Morgan, R. C.
Murdoch, J. P.
Nesbitt, A. J.
Nusbaum, L.
Nusbaum, S. R.
Parent, H. M.
Peller, L.
Pfeiffer, F. F.
Plewes, S. E.
Powell, G. W., Jr.
Powers, E. C.
Prewitt, H. B.
Prybil, P. L.
Redstone, A. L.
Rettew, H. F.
Roberts, H. L.
Rugart, K.
Sabins, E. R.
Semel, E.
Shanklin, A. P.
Sheffler, M.
Shore, D.
Smiles, R. H.
Smith, D. J.
Speckman, C. H.
Timmis, P.
Touton, R. D.
Traugott, M.
Tuckerman, G. E.
Wagner, E. K.
Wegmann, A.
Wells, W. F.
Whitney, C. W.
Wiley, D. C.
Wiltberger, C. F.
Woolston, A. H.
Woolston, R. H.
Yerkes, W. L

Pittsburgh—
Ambrose, A. H.
Baker, W. H., Jr.
Beatty, J. W.
Beighel, H. A.
Blackmore, G. C.
Borton, A. R.
Brauer, R.
Braun, C. R., Jr.
Breyer, F.
Bushnell, C. D.
Carr, M. L.
Collins, J. F. S., Jr.
Comstock, G. M.
Cooperman, E.
Cox, S. F.
Daly, R. E.
Dickinson, R. P., Jr.
Dorfan, M. I.
Edwards, P. A.
Everetts, J., Jr.
Ferderber, M. B.
Fullman, J. B.
Gallagher, F. H.
Griest, K. C.
Hach, E. C.
Hample, H. J.
Hebley, H. F.
Hecht, F. H.
Heilman, R. H.
Herre, H. M.
Houghten, F. C.
Humphreys, C. M.
Hyde, E. H.
Jones, L. K.
Kirkendall, H. J.
Lieblich, M.
Lifton, D.
Lige, W. W.
Loucks, D. W.
Lowe, W.
Maehling, L. S.
Manning, C. E.
Mahon, F. B.
Maier, G. M.
McGonagle, A.
McIntosh, F. C.
Metzger, A. F.
Miller, R. A.
Moore, H. L.
Mueller, J. E.
Nass, A. F.
Nicholls, P.
Park, H. E.
Peacock, G. S.
Powers, R. W.
Proie, J.
Reed, V. A., Jr.
Reed, W. H., III
Reilly, B. B.
Richfield, N. H.
Rieseck, W. L.
Riesmeyer, E. H., Jr.
Rittenhouse, O. R.
Rockwell, T. F.
Rose, H. J.
vonRosenberg, P. C.
Ross, D. S.
Scanlon, E. L.
Schneider, C. H.
Simpson, G. L.
Simpson, R. L.
Small, B. R.
Smith, C. F., Jr.
Smith, R. L.
Smyers, E. C.
Smith, W. F.
Snavely, A. B.
Speller, F. N.
Stanger, R. B.
Stauffer, J. E.
Steggall, H. B.
Stevenson, W. W.
Strauch, P. C.
Sweeney, R. H.
Tennant, R. J. J.
Tower, E. S.
Tumpane, J. P., Jr.
Waters, G. G.
Weddell, G. O.
Whitelaw, H. L.
Winer, B. B.

Pottstown—
Harberger, G. L.

Pottsville—
Marty, E. O.

Primos—
Johnson, A. J.

Reading—
Reese, H. L.

Reiffton—
Luck, A. W.

Ridley Park—
Mawby, P.

Scranton—
Gilboy, J. P.
Mahon, B. B.

Springdale—
Lynn, F. E.

Stroudsburg—
Kiefer, E. J.

Swarthmore—
Hobbs, W. S.
Robinson, A. S.
Thom, G. B.

Uniontown—
Marks, A. A.

Upper Darby—
Ahlff, A. A.
Bertrand, G. F.
Kipe, J. M.

Villa Nova—
Morehouse, J. S.

Washington—
Frazier, J. E.

Wilkinsburg—
Biber, H. A.
Campbell, T. F.
Good, C. S.
Graham, J. B.

Williamsport—
Axeman, J. E.

York—
Hertzler, J. R.
Kaitorie, V. T.
Mirabile, J. J.
Nicoll, S. F.
Walsh, E. R., Jr.
Zieber, W. E.

RHODE ISLAND

Pawtucket—
Kramer, C.

Providence—
Blanding, R. L.
Coleman, J. B.
Gibbs, E. W.
Hartwell, J. C.
McCarthy, J. J.

SOUTH CAROLINA

Charleston—
Bailey, F. A., Jr.
Burns, F. G.
Herty, F. B.

Clemson—
Shenk, D. H.

Columbia—
Hartin, W. R., Jr.
Kerr, W. E.
McDowell, H. L.
Sherman, W. P.

Fort Jackson—
Sloane, D. J.

Greenville—
Ramseur, V. D.
Waldrep, J. E.

SOUTH DAKOTA

Lead—
Pullen, R. R.

Sioux Falls—
Monick, F. R.

TENNESSEE

Donelson—
Wilson, V. H.

Elizabethtown—
Torok, E.

Kingsport—
Herbert, J. S.

Knoxville—
Cross, F. G.
Oakley, L. W.

Memphis—
Case, D. V.
Danielson, W. A.
Flinn, G. S.
Hoshall, R. H.
Shafer, W. P., Jr.

Nashville—
Armistead, W. C.
Baker, W. C.
Brown, F.
Campbell, G. S.
Crane, R. S.
Shapiro, C. A.

TEXAS

Amarillo—
Burnett, E. S.
Towne, C. O.

Baytown—
Kurtz, R. W.

Beaumont—
Shell, J.

Brenham—
Woods, C. F.

Brownwood—
Rhine, G. R.

Bryan—
Stiles, G. S.

College Station—
Badgett, W. H.
Giesecke, F. E.
Hines, G. M.
Hopper, J. S.
Long, W. E.
Smith, E. G.

Corpus Christi—
Holsworth, R. C.
Knepper, H. H.

Dallas—
Allison, R. E.
Anspacher, T. H.
Bishop, J. A.
Blum, H., Jr.
Brown, M. L.
Brown, M. W.
Cox, V. G.
DeVilbiss, P. T.
Dowdell, J. R.
Eutsler, E. E., Jr.
Farrow, E. E.
Gardner, C. R.
Gessell, E. T.
Gilbert, L. S.
Green, S. H.
Jelinek, F. R.
Kribs, C. L., Jr.
Landauer, L. L.
Lyford, R. G.
Martyn, H. J.
McClanahan, L. C.
Newby, I. P.
Perkins, R. C.
Pfeiffer, D. C.
Pines, S.
Renouf, E. P.
Rodgers, F. A.
Rogers, R. C.
Schimm, E. L.
Stern, E. J.
Ullrich, A. B., Jr.
Warren, H. P.
Zumwalt, R.

Denison—
Linskie, G. A.

Fort Worth—
Beals, D. E.
Harris, A. M.
MacEachin, G. C.
Miller, B. R.
Skinner, H. W.
Sprekelmeyer, J. M.
Werner, R. K.

Houston—
Andrews, W. M.
Banowsky, A. B.
Barnes, A. F.
Beaird, B. J.
Bible, H. U.
Boyd, R. L., Jr.
Chase, A. M., Jr.
Cooper, D. S.
Dieter, G. H.
Drescher, F. E.
Earl, W.
Howell, L.
Johnson, R. B.
Keeland, B. W.
Maves, G. D.
McKinney, C. A.
Mills, D. M.
Mitchell, A. J.
Morrow, J. D.
Naman, I. A.
Olson, G. E.
Owings, H. L.
Pettit, E. N., Jr.
Salinger, R. J.
Spencer, R. M.
Taylor, R. F.
Walsh, J. A.
Way, W. J., II
Workman, A. E.

Kingsville—
Richtmann, W. M.

Lubbock—
Ainsworth, S. E.

Sabine Pass—
Mabley, L. C.

San Antonio—
Barnes, R. W.
Diver, M. L.
Ebert, W. A.
Feldstein, H.
Herbert, R. M.
Pawkett, L. S.
Rummel, A. J.

Texas City—
Disney, M. A.

Vernon—
Weatherby, E. P., Jr.

Waco—
Benham, F. C., Jr.

UTAH

Ogden—
Young, J. T., Jr.

Salt Lake City—
Richardson, H. G.
Rumsey, J. L.

VERMONT

Burlington—
Lanou, J. E.
Raine, J. J.

VIRGINIA

Alexandria—
Gault, G. W.
Goergens, A. G.
Millard, J. W.
Norrington, W. L.
Roach, E. R.
Skagerberg, R.
Ward, H. H.

Arlington—
Clarke, J. H.
Fife, G. D.
Grimes, F. M.
Hackett, F. C.
Horne, H. F.
Marshall, W. D.
Martens, E. D.
Stokes, A.
Whittlesey, W. C.
Queer, E. R.

Falls Church—
Rogers, C. S.

Fort Belvoir—
McDermott, J. P.

Fredericksburg—
Delany, J. V.

Front Royal—
Hartsook, G. S., Jr.

Lawrenceville—
Jones, H. S.

Lynchburg—
Doering, F. L.
Franklin, S. H., Jr.

McLean—
Nye, L. B., Jr.

Norfolk—
Brewer, F. M.
King, B. A., Jr.
Nowitzky, H. S.
Thomas, R. C.

Petersburg—
Ibison, J. L.

Portsmouth—
Rosenberg, I.
Stubbs, W. C.

Richmond—
Bahlmann, W. F.
Bernard, E. L.
Carle, W. E.
Feder, N.
Johnston, J. A.
Peebles, J. K., Jr.
Schulz, H. I.
West, C. H., Jr.

Roanoke—
Bailey, A. E., Jr.
Bernert, L. A.
Nininger, C. H.

Williamsburg—
McGinnis, F. L.

Windsor—
Bailey, C. F.

WASHINGTON

Bremerton—
Bysom, L. L.
Naman, I. A.
Davis, R. J.

Kent—
Boyker, R. O.

Port Orchard—
Pratt, F. J.

Seattle—
Beggs, W. E.
Bouillon, L.
Clausen, A. H.
Cox, W. W.
Eastwood, E. O.
Faulkner, J. H.
Granston, R. O.
Griffith, H. T.
Hauan, M. J.
Langdon, E. H.
Leichnitz, R. W.
LeRiche, R. E.
Mallis, W.
Matthies, L. A.
May, C. W.
Mead, G. E.
Morse, R. D.
Musgrave, M. N.
Peterson, S. D.
Pollard, A. L.
Sparks, J. D.
Twist, C. F.
Wallis, W. M.
Watt, R. D.
Weber, E. L.
Wesley, R. O.
Williams, L. G.
Zokelt, C. G.

Spokane—
Russell, W. B.

Tacoma—
Chase, R. E.
Foote, E. E.
Norby, K. H.
Spofforth, W.

Vancouver—
Alben, E. A.

Yakima—
McCune, B. V.

WEST VIRGINIA

Charleston—
Rosenblatt, A. M.
Rothmann, S. C.
Shanklin, J. A.
Shearer, W. A., Jr.

Fairmont—
Tonry, R. C.
Wright, C. E.

Huntington—
Johnson, L. O.

Largent—
Donnelly, J. A.

Martinsburg—
Caskey, L. H., Jr.

South Charleston—
Pugh, D. C.

Wheeling—
Hitt, J. C.

WISCONSIN

Appleton—
Eisele, D. E.

Clintonville—
Quall, C. O.

Elm Grove—
Winkler, R. A.

Kohler—
Hvoslef, F. W.

La Crosse—
Anderegg, R. H.
Atherton, G. R.
Bowen, J. C.
Goodman, W.
Pellmounter, T. V.
Rowe, W. A.
Thomas, N. A.
Trane, R. N.

Lake Delton—
Page, H. W.

Madison—
Dean, C. L.
Feirn, W. H.
Hall, G.
Kliefoth, M. H.
Larson, G. L.
Nelson, D. W.
Seymour, J. E.
White, J. C.

Milwaukee—
Alfery, H. F.
Allan, W.
Becker, C. S.
Boden, W. F.
Bowers, A. F.
Buller, C. R.
Brown, W. H.
Cummiskey, J. F.
Cutler, J. A.
Davis, D. W., Jr.
Davis, K. T.
Ellis, H. W.
Frentzel, H. C.
Gerstenberger, E. J.
Goldsmith, F. W.
Griewisch, A. H.
Haerle, R. A.
Hamacher, K. F.
Hamilton, H. S.
Hanley, E. V.
Haus, I. J.
Hessler, L. W.
Hoffman, A.
Holland, W. T.
Hughey, T. M.
Jackson, C. H.
Jung, J. S.
Kluge, B. M.
Koch, R. G.
Krenz, A. S.
Lavorgna, M. L.
Lingen, R. A.
Mack, E. H.
McKee, J. W.
Miller, C. W.
Miller, L. B.
Mueller, H. P.
Noll, W. F.
Ouweneel, W. A.
Petersen, R. H.
Podolske, A. R.
Randolph, C. H.
Reinke, L. F.
Rice, C. J.
Runge, C. H.
Schreiber, H. W.
Spence, M. R.
Stevens, W. H.
Swisher, S. G., Jr.
Szekely, E.
Thom, A. J.
Tutsch, R. J.
Volk, G. H.
Volk, J. H.
Weil, F. H. E.
Weimer, F. G.
Werner, P. H.
Wolfe, J. S.

Neenah—
Angermeyer, A. H.
Eiss, R. M.
Harvey, A. D.

Racine—
Dixon, A. G.
Leitgabel, K. A.
May, M. F.
Menden, P. J.
Minkler, W. A.
Spieth, B.

Thiensville—
Trostel, O. A.

Wauwatosa—
Gifford, E. W.

PHILIPPINE ISLANDS

Manila—
Hausman, L. M.
Macrae, R. B.

DOMINION OF CANADA

Arvida, Que.—
Kent, A. D.

Brandon, Man.—
Yates, J. E.
Yates, R. A.

Calgary, Alta—
Deeves, E. W.
Jenkins, S. D.

Edmonton, Alta.—
Mould, D. E.

Flin Flon, Man.—
Foster, P. H.

Freeman, Ont.—
Goodram, W. E.

Galt, Ont.—
Libby, R. S.
Sheldon, W. D., Jr.

Halifax, Nova Scotia
Meagher, A. T.

Hamilton, Ont.—
Dickenson, M. E.
Moffat, O. G.

Hampstead, P. Q.—
Montgomery, E. G.

Islington, Ont.—
Wilson, G. T.

Kingston, Ont.—
Arkley, L. M.

Kitchener, Ont.—
Beavers, G. R.
Pollock, C. A.

Leaside, Ont.—
Norton, J. A.

London, Ont.—
Gilbert, T.
O'Flaherty, J. G.

Montreal, P.Q.—
Rogers, T. L. C.
Armstrong, W. J.
Ballantyne, G. L.
Baxter, W. E.
Berridge, W. W.
Boland, R. O.
Chenevert, J. G.
Colle, S. S.
Darling, A. B.
Dufault, F. H.
Dykes, J. B.
Ewens, F. G.
Flanagan, J. B.
Forrester, N. J.
Freeman, E. M.
Garneau, L.
Hamlet, F. A.
Hood, L. A.
Hughes, W. U.
Johnson, C. W.
Keith, J. P.
Kitchen, W. H. J.
LaMontagne, A. F.
LaRue, J. A. D.
Linton, J. P.
Madely, F. J.

Martin, R.
Milne, A. H.
Morris, J. A.
Nickle, A. J.
Noyes, R. R.
Osborne, G. H.
Peart, A. M.
Perras, G. E.
Phipps, F. G.
Robertson, J. A. M.
Ross, J. D.
Sampson, E. T.
Shaw, J. A.
Standring, R. A.
Ste-Marie, G. P.
Timmins, W. W.
Tolhurst, G. C.
Twizell, E. W.
Vollmann, C. W.
Watts, A. E.
Wells, E. E.
Wiggs, G. L.
Wilkinson, A.
Wormley, R. F.
Worthington, T. H.

Moosomin, Sask.—
Barton, E. H.

Newton, Ont.—
Barrett, C. M.

Noranda, P.Q.—
MacLean, H. A.

North Bay, Ont.—
Goodman, C. E.

Oakville, Ont.—
Stott, F. W.

Ottawa, Ont.—
Allen, A. W.
Colclough, O. T.
Gray, G. A.
Johns, C. F.
McGrail, T. E.
Pennock, W. B.

Outremont, P.Q.—
Gittleson, H.

Preston, Ont.—
Everest, R. H.
Wood, A. W.

Quebec, P.Q.—
Paquet, J.-M.
Roy, L.

Rivers, Man.—
Armour, E. G.

Sackville, N.B.—
Rand, F. R.

Sherbrooke, P.Q.—
Archambault, J. A.
Labonne, H.
Thompson, F.

Thorold, Ont.—
Calnan, E. J.

Toronto, Ont.—
Abbott, T. J.
Alexander, S. W.
Allcut, E. A.
Allsop, R. P.
Angus, H. H.
Anthes, L. L.
Arrowsmith, J. O.
Bayles, R. W.
Blackhall, W. R.
Bowerman, E. L.
Brittain, A., Jr.
Carter, A. W.
Chambers, F. W.
Church, H. J.
Clifton, J. A.
Cole, G. E.
Davis, E. J.
Dickey, A. J.
Dickson, G. P.
Dion, A. M.
Dowler, E. A.
Duncan, W. A.
Eaton, W. G. M.
Ellis, F. E.
Fear, S. L.
Ferris, A. L.
Fitzsimons, J. P.
Foley, J. J.
Fox, E.
Fox, J. H.
Gauley, E. R.
Givin, A. W.
Gordon, C. W.
Gurney, E. H.
Gurney, E. R.
Hatch, G.
Henion, H. D.
Hill, H. G.
Hills, A. H.
Hopper, G. H.
Hughes, L. K.
Jenkinson, V. J.
Jenney, H. B.
Jennings, S. A.
Jones, A. T.
Kelly, W. C.
Kemp, G. C.
Lawlor, J. J.
Leitch, A. S.
Lock, R. H.
MacDonald, D. J.
Marriner, J. M. S.
Mathison, R. St. C.
Maxwell, R. S.
McKerlie, J.
McLaren, T. H.
Moore, F. C.
Moore, H. S.
Morgan, A. S.
Murray, H. G. S.
Nearingburg, A.
O'Neill, J. W.
Paul, D. I.
Philip, W.
Price, D. O.
Ritchie, A. G.
Roth, H. R.
Shears, M. W.
Smith, G. E.
Smith, W. H.
Spall, E. G.
Stencel, R. A.
Stott, D. A.
Sturdy, O. C.
Tasker, C.
Thomas, M. F.
Thomsen, N. B.
Walden, C. D.
Wardell, A.
Woollard, M. S.

Trois-Rivieres, Que.—
Germain, O.

Vancouver, B.C.—
Hale, F. J.
Johnston, R. E.
Leek, C. W.
Leek, W.
Turland, C. H.
Vissac, G. A.

Victoria, B. C.—
Sheret, A.

West Hill, Ont.—
Playfair, G. A.

Westmount, P.Q.—
Colford, J.
Lefebvre, E. J.

Windsor, Ont.—
Hare, W. A.

Winnipeg, Man.—
Anderson, E.
Argue, E. J.
Ball, F. T.
Charles, P. L.
Chester, F. L.
Dahlgren, G. E.
Davis, G. C.
Glass, W.
Hatch, O. J.
Hepburn, E. M.
Jones, B. G.
Kent, R. L.
McDonald, I.
Michie, D. F.
Miller, E. R.
Munn, E. F.
Price, E. H.
Steele, J. B.
Stephenson, J. R.
Swain, D. S.
Worton, W.

Woodstock, Ont.—
Karges, A.

York Mills, Ont.—
Bishop, J. W.

AUSTRALIA

Darlinghurst—
Henderson, A. S.

Kew—
Davies, R. H.

Melbourne—
Atherton, A. E., Jr.
Bell, S. R.
Ross, R.

Rose Bay—
Robinson, J. A.

Sydney—
Davey, G. I.
Moloney, R. R.
Roseby, T. A.
Sands, C. C.

Watsons Bay—
Picot, J. W.

BELGIUM

Brussels—
Lebrun, P.

BERMUDA ISLANDS

Blackman, A. O.

BRAZIL

Rio de Janiero—
Botelho, N. J.
DeSales, M., Jr.

CHILE

Santiago—
Demeter.

CHINA

Shanghai—
Bradford, G. G.
Doughty, C. J.
Hutchinson, F. A.
Kwan, I. K.
Loh, N.-S.
Loo, P. Y.
Muckle, J.
Wong, W. S. B.

CUBA

Havana—
Colmenares, G. V.

DENMARK

Copenhagen—
Reck, W. E.
Schulein, E. H.

ENGLAND

Birmingham—
Richardson, R. D.

Buckinghamshire—
Russell, J. N.

Cheshire—
Kennett, V. A.

Kent—
Figgis, T. G.
Lipscombe, H. W. J.

Lancaster—
Bartley, H. E.

Leeds—
Jennins, H. H.

Liverpool—
Thomas, A. E.

London—
Bailey, W. M.
Benham, C. S. K.
Bird, G. L. H.
Butt, R. E. W.
Daniel, W. E.
Haden, G. N.
Herring, E.
Jackson, G. R.
Morrison, C. B.
Nobbs, W. W.
Swain, W. L.
Symonds, E. S.
Troup, J. D.
Wilson, E. D.

Manchester—
Pougher, B. R. E.
Pougher, E. W.
Webb, J. W.
Yates, W.

Middlesex—
Case, W. G.
Fraser, J. J.
MacLachlan, V. D.
Smith, G. E.

Nottingham—
Norfolk, L. W.

Surrey—
Faber, O.

Sussex—
Carter, D.

Trentham—
Greenland, S. F.

Trowbridge—
Haden, W. N.

Warwickshire—
Mann, W. N.

EGYPT

Alexandria—
Tallianos, P. C.

Cairo—
Ezz-El-Din, K.
Tahry, M. E.
Zaki, H. M.

FRANCE

Boulogne—
Waudby, W.

Lyon—
Goenaga, R. C.

Paris—
Bodmer, E.
Modiano, R.
Nessi, A.

Sucy en Brie—
Beaurrienne, A.

Vanves—
Ghilardi, F.

FRENCH NORTH AFRICA

Macherel, F.

GERMANY

Berlin-Charlottenburg—
Brust, O.

Berlin-Wilmersdorf—
Schmidt, E. G.

Stuttgart—
Klein, A. R.

HOLLAND

Amsterdam—
ter Weeme, A.

INDIA

Bombay—
Badhe, J. M.
Duplant, J. L.

Calcutta—
Ghose, K. N.
Ghosh, B. B.
Rachal, J. M.
Stirling, W. N.

IRELAND

Cork—
Barry, P. I.

Dublin—
Leonard, L. C. G.

ITALY

Florence—
Scotti, F. D.

Milan—
Dell'Orto, L.
Gini, A.
Marzorati, G.
Parrilli, R.

Torino—
Baldi, G.

JAPAN

Osaka—
Fukui, K.

Tokyo—
Kitaura, S.
Saito, S.
Sekido, K.

MANCHOUKUO

Yamatoku—
Kawase, S.

MEXICO

Mexico, D.F.—
Gilfrin, G. F.
Huber, E.
Ramoneda, E.

NETHERLANDS

Arnhem—
Tanger, O. C. F.

NEW ZEALAND

Christchurch—
Taylor, E. M.
Vale, H. A. L.

Dunedin—
Davies, G. W.

NORWAY

Oslo—
Tjersland, A.

Stabekk—
Alfsen, N.

PALESTINE

Tel Aviv—
Petigrow, B. N.

PUERTO RICO

Fort Buchanan—
Ruemmele, A. M.

ROUMANIA

Bucharest—
Wasser, M.

SOUTH AFRICA

Durban—
Christierson, C. A.

Johannesburg—
Caldwell, R. J. S.
Daitsh, A.
Ehlers, J.
Overton, S. H.
Rabe, A. E.

SPAIN

Madrid—
Alfageme, B.

STRAITS SETTLEMENTS

Singapore—
Avery, I.
Faxon, H. C.

SWEDEN

Nockeby—
Erikson, H. A.

Stockholm—
Eklund, K. G.
Gille, H. B.
Ostrom, E. W.
Rosell, A. F.
Stromgren, S. G.
Theorell, A. T.
Theorell, H. G. T.

TURKEY

Istanbul—
Karakash, T. J.
Veglery, A.

VENEZUELA

Caracas—
Blas, R. J.
Westendarp, F. G.

PAST OFFICERS

AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS

1894

President..........Edward P. Bates
1st Vice-President..........Wm. M. Mackay
2nd Vice-President..........Wiltsie F. Wolfe
3rd Vice-President..........Chas. S. Onderdonk
Treasurer..........Judson A. Goodrich
Secretary..........L. H. Hart

Board of Managers

Chairman, Fred P. Smith

Henry Adams	A. A. Cary
Hugh J. Barron	James A. Harding
Edward P. Bates, *Pres.*	L. H. Hart, *Secy.*

Council

Chairman, R. C. Carpenter

Albert A. Cryer	Chas. W. Newton
F. W. Foster	Ulysses G. Scollay, *Secy.*

1895

President..........Stewart A. Jellett
1st Vice-President..........Wm. M. Mackay
2nd Vice-President..........Chas. S. Onderdonk
3rd Vice-President..........D. M. Quay
Treasurer..........Judson A. Goodrich
Secretary..........L. H. Hart

Board of Managers

Chairman, James A. Harding

Geo. B. Cobb	Ulysses G. Scollay
Wm. McMannis	B. F. Stangland
Stewart A. Jellett, *Pres.*	L. H. Hart, *Secy.*

Council

Chairman, R. C. Carpenter

Henry Adams	T. J. Waters
Edward P. Bates	Albert A. Cryer, *Secy.*

1896

President..........R. C. Carpenter
1st Vice-President..........D. M. Quay
2nd Vice-President..........Edward P. Bates
3rd Vice-President..........F. W. Foster
Treasurer..........Judson A. Goodrich
Secretary..........L. H. Hart

Board of Managers

Chairman, Wm. M. Mackay

Hugh J. Barron	Stewart A. Jellett
W. S. Hadaway, Jr.	Wiltsie F. Wolfe
R. C. Carpenter, *Pres.*	L. H. Hart, *Secy.*

Council

Chairman, A. A. Cary

Albert A. Cryer	B. F. Stangland
Wm. McMannis	J. J. Blackmore, *Secy.*

1897

President..........Wm. M. Mackay
1st Vice-President..........H. D. Crane
2nd Vice-President..........Henry Adams
3rd Vice-President..........A. E. Kenrick
Treasurer..........Judson A. Goodrich
Secretary..........H. M. Swetland

Board of Managers

Chairman, R. C. Carpenter

Edward P. Bates	Stewart A. Jellett
W. S. Hadaway, Jr.	Wiltsie F. Wolfe
Wm. M. Mackay, *Pres.*	H. M. Swetland, *Secy.*

Council

Chairman, Albert A. Cryer

John A. Fish	James Mackay
Wm. McMannis	B. F. Stangland

1898

President..........Wiltsie F. Wolfe
1st Vice-President..........J. H. Kinealy
2nd Vice-President..........A. E. Kenrick
3rd Vice-President..........John A. Fish
Treasurer..........Judson A. Goodrich
Secretary..........Stewart A. Jellett

Board of Managers

Chairman, Wm. M. Mackay

Thomas Barwick	A. C. Mott
John A. Connolly	Francis A. Williams
Wiltsie F. Wolfe, *Pres.*	Stewart A. Jellett, *Secy.*

Council

Chairman, R. C. Carpenter

Henry Adams	W. S. Hadaway, Jr.
Albert A. Cryer	Wm. McMannis
Wiltsie F. Wolfe, *Pres.*	Stewart A. Jellett, *Secy.*

1899

President..........Henry Adams
1st Vice-President..........D. M. Quay
2nd Vice-President..........A. E. Kenrick
3rd Vice-President..........Francis A Williams
Treasurer..........Judson A. Goodrich
Secretary..........Wm. M. Mackay

Board of Managers

Chairman, Stewart A. Jellett

B. H. Carpenter	Wm. Kent
A. A. Cary	Wiltsie F. Wolfe
Henry Adams, *Pres.*	Wm. M. Mackay, *Secy.*

Council

Chairman, R. C. Carpenter

John Gormly	Wm. McMannis
W. S. Hadaway, Jr.	B. F. Stangland
Henry Adams, *Pres.*	Wm. M. Mackay, *Secy.*

1900

President..........D. M. Quay
1st Vice-President..........A. E. Kenrick
2nd Vice-President..........Francis A. Williams
Treasurer..........Judson A. Goodrich
Secretary..........Wm. M. Mackay

Board of Governors

Chairman, D. M. Quay

Wm. Kent, *Vice-Chm.*	D. M. Nesbit
R. C. Carpenter	C. B. J. Snyder
John Gormly	Wm. M. Mackay, *Secy.*

1901

President..........J. H. Kinealy
1st Vice-President..........A. E. Kenrick
2nd Vice-President..........Andrew Harvey
Treasurer..........Judson A. Goodrich
Secretary..........Wm. M. Mackay

Board of Governors

Chairman, J. H. Kinealy

Wm. Kent, *Vice-Chm.*	John Gormly
R. C. Carpenter	C. B. J. Snyder
R. P. Bolton	Wm. M. Mackay, *Secy.*

1902

President .... A. E. Kenrick
1st Vice-President .... Andrew Harvey
2nd Vice-President .... Robert C. Clarkson
Treasurer .... Judson A. Goodrich
Secretary .... Wm. M. Mackay

Board of Governors

Chairman, A. E. Kenrick

John Gormly, *Vice-Chm.*	J. H. Kinealy
R. C. Carpenter	C. B. J. Snyder
Wm. Kent	Wm. M. Mackay, *Secy.*

1903

President .... H. D. Crane
1st Vice-President .... Wm. Kent
2nd Vice-President .... R. P. Bolton
Treasurer .... Judson A. Goodrich
Secretary .... Wm. M. Mackay

Board of Governors

Chairman, H. D. Crane

C. B. J. Snyder, *Vice-Chm.*	A. E. Kenrick
R. C. Carpenter	Geo. Mehring
John Gormly	Wm. M. Mackay, *Secy.*

1904

President .... Andrew Harvey
1st Vice-President .... John Gormly
2nd Vice-President .... Robert C. Clarkson
Treasurer .... Ulysses G. Scollay
Secretary .... Wm. M. Mackay

Board of Governors

Chairman, Andrew Harvey

John Gormly	H. D. Crane
Robert C. Clarkson	A. E. Kenrick
J. J. Blackmore	C. B. J. Snyder
R. C. Carpenter	Wm. M. Mackay, *Secy.*

1905

President .... Wm. Kent
1st Vice-President .... R. P. Bolton
2nd Vice-President .... C. B. J. Snyder
Treasurer .... Ulysses G. Scollay
Secretary .... Wm. M. Mackay

Board of Governors

Chairman, Wm. Kent

R. P. Bolton	James Mackay
C. B. J. Snyder	B. F. Stangland
B. H. Carpenter	J. C. F. Trachsel
A. B. Franklin	Wm. M. Mackay, *Secy.*

1906

President .... John Gormly
1st Vice-President .... C. B. J. Snyder
2nd Vice-President .... T. J. Waters
Treasurer .... Ulysses G. Scollay
Secretary .... Wm. M. Mackay

Board of Governors

Chairman, John Gormly

C. B. J. Snyder, *Vice-Chm.*	James Mackay
R. C. Carpenter	B. F. Stangland
Frank K. Chew	T. J. Waters
A. B. Franklin	Wm. M. Mackay, *Secy.*

1907

President .... C. B. J. Snyder
1st Vice-President .... James Mackay
2nd Vice-President .... Wm. G. Snow
Treasurer .... Ulysses G. Scollay
Secretary .... Wm. M. Mackay

Board of Governors

Chairman, C. B. J. Snyder

James Mackay, *Vice-Chm.*	Frank K. Chew
R. E. Atkinson	A. B. Franklin
R. C. Carpenter	Wm. G. Snow
Edmund F. Capron	Wm. M. Mackay, *Secy.*

1908

President .... James Mackay
1st Vice-President .... Jas. D. Hoffman
2nd Vice-President .... B. F. Stangland
Treasurer .... Ulysses G. Scollay
Secretary .... Wm. M. Mackay

Board of Governors

Chairman, James Mackay

Jas. D. Hoffman, *Vice-Chm.*	John F. Hale
B. F. Stangland	August Kehm
R. C. Carpenter	C. B. J. Snyder
Frank K. Chew	Wm. M. Mackay, *Secy.*

1909

President .... Wm. G. Snow
1st Vice-President .... August Kehm
2nd Vice-President .... B. S. Harrison
Treasurer .... Ulysses G. Scollay
Secretary .... Wm. M. Mackay

Board of Governors

Chairman, Wm. G. Snow

August Kehm, *Vice-Chm.*	Samuel R. Lewis
John R. Allen	James Mackay
R. C. Carpenter	B. F. Stangland
B. S. Harrison	Wm. M. Mackay, *Secy.*

1910

President .... Jas. D. Hoffman
1st Vice-President .... R. P. Bolton
2nd Vice-President .... Samuel R. Lewis
Treasurer .... Ulysses G. Scollay
Secretary .... Wm. M. Mackay

Board of Governors

Chairman, Jas. D. Hoffman

R. P. Bolton, *Vice-Chm.*	John F. Hale
Geo. W. Barr	Samuel R. Lewis
R. C. Carpenter	James Mackay
Judson A. Goodrich	Wm. M. Mackay, *Secy.*

1911

President .... R. P. Bolton
1st Vice-President .... John R. Allen
2nd Vice-President .... A. B. Franklin
Treasurer .... Ulysses G. Scollay
Secretary .... Wm. W. Macon

Board of Governors

Chairman, R. P. Bolton

John R. Allen, *Vice-Chm.*	A. B. Franklin
John T. Bradley	Jas. D. Hoffman
R. C. Carpenter	August Kehm
James H. Davis	Wm. W. Macon, *Secy.*

1912

President....John R. Allen
1st Vice-President....John F. Hale
2nd Vice-President....Edmund F. Capron
Treasurer....James A. Donnelly
Secretary....Wm. W. Macon

Board of Governors

Chairman, John R. Allen

John F. Hale, *Vice-Chm.*
Edmund F. Capron
R. P. Bolton
Jas. D. Hoffman
Dwight D. Kimball
Samuel R. Lewis
Wm. M. Mackay
Wm. W. Macon, *Secy.*

1913

President....John F. Hale
1st Vice-President....A. B. Franklin
2nd Vice-President....Edmund F. Capron
Treasurer....James A. Donnelly
Secretary....Edwin A. Scott

Board of Governors

Chairman, John F. Hale

A. B. Franklin, *Vice-Chm.*
John R. Allen
Edmund F. Capron
R. P. Bolton
Frank T. Chapman
Ralph Collamore
James A. Donnelly
Dwight D. Kimball
Wm. W. Macon
James M. Stannard
Theodore Weinshank
Edwin A. Scott, *Secy.*

1914

President....Samuel R. Lewis
1st Vice-President....Edmund F. Capron
2nd Vice-President....Dwight D. Kimbal
Treasurer....James A. Donnelly
Secretary....J. J. Blackmore

Council

Chairman, Samuel R. Lewis

E. F. Capron, *Vice-Chm.*
Dwight D. Kimball
John R. Allen
Frank T. Chapman
Frank I. Cooper
James A. Donnelly
John F. Hale
Harry M. Hart
Frank G. McCann
Wm. W. Macon
James M. Stannard
J. J. Blackmore, *Secy.*

1915

President....Dwight D. Kimball
1st Vice-President....Harry M. Hart
2nd Vice-President....Frank T. Chapman
Treasurer....Homer Addams
Secretary....J. J. Blackmore

Council

Chairman, Dwight D. Kimball

Harry M. Hart, *Vice-Chm.*
Homer Addams
Frank T. Chapman
Frank I. Cooper
E. Vernon Hill
Wm. M. Kingsbury
Samuel R. Lewis
Frank G. McCann
J. T. J. Mellon
Henry C. Meyer, Jr.
Arthur K. Ohmes
J. J. Blackmore, *Secy.*

1916

President....Harry M. Hart
1st Vice-President....Frank T. Chapman
2nd Vice-President....Arthur K. Ohmes
Treasurer....Homer Addams
Secretary....Casin W. Obert

Council

Chairman, Harry M. Hart

F. T. Chapman, *Vice-Chm.*
Homer Addams
Charles R. Bishop
Frank I. Cooper
Milton W. Franklin
E. Vernon Hill
Dwight D. Kimball
Henry C. Meyer, Jr.
Arthur K. Ohmes
Fred R. Still
Walter S. Timmis
Casin W. Obert, *Secy.*

1917

President....J. Irvine Lyle
1st Vice-President....Arthur K. Ohmes
2nd Vice-President....Fred R. Still
Treasurer....Homer Addams
Secretary....Casin W. Obert

Council

Chairman, J. Irvine Lyle

A. K. Ohmes, *Vice-Chm.*
Homer Addams
Davis S. Boyden
Bert C. Davis
Milton W. Franklin
Charles A. Fuller
Harry M. Hart
E. Vernon Hill
James M. Stannard
Fred R. Still
Walter S. Timmis
Casin W. Obert, *Secy.*

1918

President....Fred R. Still
1st Vice-President....Walter S. Timmis
2nd Vice-President....E. Vernon Hill
Treasurer....Homer Addams
Secretary....Casin W. Obert

Council

Chairman, Fred R. Still

W. S. Timmis, *Vice-Chm.*
Homer Addams
William H. Driscoll
Howard H. Fielding
H. P. Gant
C. W. Kimball
J. Irvine Lyle
E. Vernon Hill
Frank G. Phegley
Fred. W. Powers
Champlain L. Riley
Casin W. Obert, *Secy.*

1919

President....Walter S. Timmis
1st Vice-President....E. Vernon Hill
2nd Vice-President....Milton W. Franklin
Treasurer....Homer Addams
Secretary....Casin W. Obert

Council

Chairman, Walter S. Timmis

E. Vernon Hill, *Vice-Chm.*
Homer Addams
Howard H. Fielding
Milton W. Franklin
Harry E. Gerrish
George B. Nichols
Frank G. Phegley
Fred. W. Powers
Robt. W. Pryor, Jr.
Champlain L. Riley
Fred R. Still
Casin W. Obert, *Secy.*

1920

President....E. Vernon Hill
1st Vice-President....Champlain L. Riley
2nd Vice-President....Jay R. McColl
Treasurer....Homer Addams
Secretary....Casin W. Obert

Council

Chairman, E. Vernon Hill

C. L. Riley, *Vice-Chm.*
Homer Addams
Jos. A. Cutler
Wm. H. Driscoll
A. C. Edgar
Alfred Kellogg
Jay R. McColl
George B. Nichols
Robt. W. Pryor, Jr.
W. S. Timmis
Perry West
Casin W. Obert, *Secy.*

1921

President....Champlain L. Riley
1st Vice-President....Jay R. McColl
2nd Vice-President....H. P. Gant
Treasurer....Homer Addams
Secretary....Casin W. Obert

Council

Chairman, Champlain L. Riley

Jay R. McColl, *Vice-Chm.*
Homer Addams
Jos. A. Cutler
Samuel E. Dibble
Wm. H. Driscoll
H. P. Gant
E. S. Hallett
E. Vernon Hill
Alfred Kellogg
E. E. McNair
Perry West
Casin W. Obert, *Secy.*

1922

President......Jay R. McColl
1st Vice-President......H. P. Gant
2nd Vice-President......Samuel E. Dibble
Treasurer......Homer Addams
Secretary......Casin W. Obert

Council

Chairman, Jay R. McColl

H. P. Gant, *Vice-Chm.*
Homer Addams
Jos. A. Cutler
Samuel E. Dibble
Wm. H. Driscoll
E. S. Hallett
L. A. Harding
E. E. McNair
H. J. Meyer
C. L. Riley
Perry West
Casin W. Obert, *Secy.*

1923

President......H. P. Gant
1st Vice-President......Homer Addams
2nd Vice-President......E. E. McNair
Treasurer......Wm. H. Driscoll
Secretary......Casin W. Obert

Council

Chairman, H. P. Gant

Homer Addams, *Vice-Chm.*
W. H. Carrier
J. A. Cutler
S. E. Dibble
Wm. H. Driscoll
E. S. Hallett
Alfred Kellogg
Thornton Lewis
E. E. McNair
Perry West
Casin W. Obert, *Secy.*

1924

President......Homer Addams
1st Vice-President......S. E. Dibble
2nd Vice-President......William H. Driscoll
Treasurer......Perry West
Secretary......F. C. Houghten

Council

Chairman, Homer Addams

S. E. Dibble, *Vice-Chm.*
F. Paul Anderson
W H. Carrier
J. A. Cutler
William H. Driscoll
H. P. Gant
W. E. Gillham
L. A. Harding
Alfred Kellogg
Thornton Lewis
Perry West
F. C. Houghten, *Secy.*

1925

President......S. E. Dibble
1st Vice-President......Wm. H. Driscoll
2nd Vice-President......F. Paul Anderson
Treasurer......Perry West
Secretary......F. C. Houghten

Council

Chairman, S. E. Dibble

Wm. H. Driscoll, *Vice-Chm.*
Homer Addams
F. Paul Anderson
W. H. Carrier
J. A. Cutler
W. E. Gillham
W. T. Jones
Thornton Lewis
J. H. Walker
Perry West
A. C. Willard
F. C. Houghten, *Secy.*

1926

President......W. H. Driscoll
1st Vice-President......F. Paul Anderson
2nd Vice-President......A. C. Willard
Treasurer......W. E. Gillham
Secretary......A. V. Hutchinson

Council

Chairman, W. H. Driscoll

F. Paul Anderson, *Vice-Chm.*
W. H. Carrier
J. A. Cutler
S. E. Dibble
W. E. Gillham
C. V. Haynes
W. T. Jones
E. B. Langenberg
Thornton Lewis
J. F. McIntire
A. C. Willard

1927

President......F. Paul Anderson
1st Vice-President......A. C. Willard
2nd Vice-President......Thornton Lewis
Treasurer......W. E. Gillham
Secretary......A. V. Hutchinson

Council

Chairman, F. Paul Anderson

A. C. Willard, *Vice-Chm.*
H. H. Angus
W. H. Carrier
W. H. Driscoll
Roswell Farnham
H. H. Fielding
W. E. Gillham
C. V. Haynes
John Howatt
W. T. Jones
J. J. Kissick
E. B. Langenberg
Thornton Lewis
J. F. McIntire
H. Lee Moore
F. B. Rowley

1928

President......A. C. Willard
1st Vice-President......Thornton Lewis
2nd Vice-President......L. A. Harding
Treasurer......W. E. Gillham
Secretary......A. V. Hutchinson

Council

Chairman, A. C. Willard

Thornton Lewis, *Vice-Chm.*
F. Paul Anderson
H. H. Angus
W. H. Carrier
N. W. Downes
Roswell Farnham
W. E. Gillham
C. V. Haynes
John Howatt
W. T. Jones
J. J. Kissick
E. B. Langenberg
J. F. McIntire
H. Lee Moore
F. B. Rowley

1929

President......Thornton Lewis
1st Vice-President......L. A. Harding
2nd Vice-President......W. H. Carrier
Treasurer......W. E. Gillham
Secretary......A. V. Hutchinson
Technical Secretary......P. D. Close

Council

Chairman, Thornton Lewis

L. A. Harding, *Vice-Chm.*
H. H. Angus
W. H. Carrier
N. W. Downes
Roswell Farnham
W. E. Gillham
C. V. Haynes
A. C. Willard
John Howatt
W. T. Jones
E. B. Langenberg
G. L. Larson
F. C. McIntosh
W. A. Rowe
F. B. Rowley

1930

President......L. A. Harding
1st Vice-President......W. H. Carrier
2nd Vice-President......F. B. Rowley
Treasurer......C. W. Farrar
Secretary......A. V. Hutchinson
Technical Secretary......P. D. Close

Council

Chairman, L. A. Harding

W. H. Carrier, *Vice-Chm.*
H. H. Angus
D. S. Boyden
R. H. Carpenter
J. D. Cassell
N. W. Downes
Roswell Farnham
C. W. Farrar
John Howatt
W. T. Jones
E. B. Langenberg
G. L. Larson
Thornton Lewis
F. C. McIntosh
W. A. Rowe
F. B. Rowley

1931

President......W. H. Carrier
1st Vice-President......F. B. Rowley
2nd Vice-President......W. T. Jones
Treasurer......F. D. Mensing
Secretary......A. V. Hutchinson
Technical Secretary......P. D. Close

Council

Chairman, W. H. Carrier

F. B. Rowley, *Vice-Chm.*
D. S. Boyden
E. K. Campbell
R. H. Carpenter
J. D. Cassell
E. O. Eastwood
Roswell Farnham
E. H. Gurney
L. A. Harding
John Howatt
W. T. Jones
E. B. Langenberg
G. L. Larson
F. C. McIntosh
F. D. Mensing
W. A. Rowe

1932

President .... F. B. Rowley
1st Vice-President .... W. T. Jones
2nd Vice-President .... C. V. Haynes
Treasurer .... F. D. Mensing
Secretary .... A. V. Hutchinson
Technical Secretary .... P. D. Close

Council

Chairman, F. B. Rowley

W. T. Jones, *Vice-Chm.*	F. E. Giesecke
D. S. Boyden	E. H. Gurney
E. K. Campbell	C. V. Haynes
R. H. Carpenter	John Howatt
W. H. Carrier	G. L. Larson
John D. Cassell	J. F. McIntire
E. O. Eastwood	F. D. Mensing
Roswell Farnham	W. E. Stark

1933

President .... W. T. Jones
1st Vice-President .... C. V. Haynes
2nd Vice-President .... John Howatt
Treasurer .... D. S. Boyden
Secretary .... A. V. Hutchinson

Council

Chairman, W. T. Jones

C. V. Haynes, *Vice-Chm.*	E. H. Gurney
D. S. Boyden	John Howatt
E. K. Campbell	G. L. Larson
R. H. Carpenter	J. F. McIntire
J. D. Cassell	F. C. McIntosh
E. O. Eastwood	L. W. Moon
Roswell Farnham	F. B. Rowley
F. E. Giesecke	W. E. Stark

1934

President .... C. V. Haynes
1st Vice-President .... John Howatt
2nd Vice-President .... G. L. Larson
Treasurer .... D. S. Boyden
Secretary .... A. V. Hutchinson

Council

Chairman, C. V. Haynes

John Howatt, *Vice-Chm.*	W. T. Jones
M. C. Beman	G. L. Larson
D. S. Boyden	J. F. McIntire
Albert Buenger	F. C. McIntosh
R. H. Carpenter	L. Walter Moon
J. D. Cassell	O. W. Ott
F. E. Giesecke	W. A. Russell
E. H. Gurney	W. E. Stark

1935

President .... John Howatt
1st Vice-President .... G. L. Larson
2nd Vice-President .... D. S. Boyden
Treasurer .... A. J. Offner
Secretary .... A. V. Hutchinson

Council

Chairman, John Howatt

G. L. Larson, *Vice-Chm.*	C. V. Haynes
M. C. Beman	J. F. McIntire
D. S. Boyden	F. C. McIntosh
Albert Buenger	L. Walter Moon
R. H. Carpenter	A. J. Offner
J. D. Cassell	O. W. Ott
F. E. Giesecke	W. A. Russell
E. H. Gurney	W. E. Stark

1936

President .... G. L. Larson
1st Vice-President .... D. S. Boyden
2nd Vice-President .... E. H. Gurney
Treasurer .... A. J. Offner
Secretary .... A. V. Hutchinson

Council

Chairman, G. L. Larson

D. S. Boyden, *Vice-Chm.*	John Howatt
M. C. Beman	C. M. Humphreys
R. C. Bolsinger	L. Walter Moon
Albert Buenger	J. F. McIntire
S. H. Downs	A. J. Offner
W. L. Fleisher	O. W. Ott
F. E. Giesecke	W. A. Russell
E. H. Gurney	W. E. Stark

1937

President .... D. S. Boyden
1st Vice-President .... E. Holt Gurney
2nd Vice-President .... J. F. McIntire
Treasurer .... A. J. Offner
Secretary .... A. V. Hutchinson

Council

Chairman, D. S. Boyden

E. H. Gurney, *Vice-Chm.*		E. O. Eastwood
J. J. Aeberly		W. L. Fleisher
M. C. Beman		F. E. Giesecke
R. C. Bolsinger		C. M. Humphreys
Albert Buenger		G. L. Larson
S. H. Downs	W. E. Stark	W. A. Russell

1938

President .... E. Holt Gurney
1st Vice-President .... J. F. McIntire
2nd Vice-President .... F. E. Giesecke
Treasurer .... A. J. Offner
Secretary .... A. V. Hutchinson
Technical Secretary .... John James

Council

Chairman, E. Holt Gurney

J. F. McIntire, *Vice-Chm.*	W. L. Fleisher
N. D. Adams	F. E. Giesecke
J. J. Aeberly	C. M. Humphreys
M. C. Beman	A. P. Kratz
R. C. Bolsinger	A. J. Offner
D. S. Boyden	W. A. Russell
S. H. Downs	J. H. Walker
E. O. Eastwood	G. L. Wiggs

1939

President .... J. F. McIntire
1st Vice-President .... F. E. Giesecke
2nd Vice-President .... W. L. Fleisher
Treasurer .... M. F. Blankin
Secretary .... A. V. Hutchinson
Technical Secretary .... John James

Council

Chairman, J. F. McIntire

F. E. Giesecke, *Vice-Chm.*	W. L. Fleisher
N. D. Adams	E. H. Gurney
J. J. Aeberly	A. P. Kratz
M. C. Beman	A. J. Offner
M. F. Blankin	W. A. Russell
E. K. Campbell	G. L. Tuve
S. H. Downs	J. H. Walker
E. O. Eastwood	G. L. Wiggs

1940

President .... F. E. Giesecke
1st Vice-President .... W. L. Fleisher
2nd Vice-President .... E. O. Eastwood
Treasurer .... M. F. Blankin
Secretary .... A. V. Hutchinson
Technical Secretary .... John James

Council

Chairman, F. E. Giesecke

W. L. Fleisher, *Vice-Chm.*	J. F. McIntire
N. D. Adams	A. J. Offner
E. K. Campbell	G. L. Tuve
J. F. S. Collins, Jr.	T. H. Urdahl
S. H. Downs	J. H. Walker
A. P. Kratz	G. L. Wiggs
E. N. McDonnell	C.-E. A. Winslow

1941

President .... W. L. Fleisher
1st Vice-President .... E. O. Eastwood
2nd Vice-President .... J. H. Walker
Treasurer .... M. F. Blankin
Secretary .... A. V. Hutchinson
Technical Secretary .... John James

Council

Chairman, W. L. Fleisher

E. O. Eastwood, *Vice-Chm.*	A. J. Offner
M. F. Blankin	W. A. Russell
E. K. Campbell	L. P. Saunders
J. F. Collins, Jr.	C. Tasker
S. H. Downs	G. L. Tuve
F. E. Giesecke	T. H. Urdahl
A. P. Kratz	J. H. Walker
E. N. McDonnell	C.-E. A. Winslow

A. E. Stacey, Jr., *Ex-Officio*